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DEVELOPMENT OF VERTICAL SHAFT SKIP AND GUIDE DESIGN

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This report describes the development of skip and guide designs for vertical shafts in deep vein mines. Skip and guide systems for bottom dump skips are identified as being used in current practice. Mine visits were made to obtain current information on the needs and problems of operating mines. Methods of structural and mechanical analysis of skip bodies and bails are outlined. A computer model was written to find the dynamic interaction between skips and guides for guides with geometric misalinements. A materials evaluation section identifies current available and applicable materials that can be used effectively for new skip construction. Guide materials are also presented. Analysis includes the economies resulting from streamlined buntons and from low-skip payload ratios. Selection of materials is based on strength, abrasion, and corrosion resistance. Design requirements related to the shaft environment are presented. Consideration for the selection of guides and buntons is also given. Conceptual design includes the concept of cylindrical skips, center-seeking guide rollers, and asymmetric bail design to decrease dynamic interaction. An outline for field tests is presented.

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FOREWORD

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This report is a summary of the work recently completed as part of this contract during the period June 1976 to June 1977. This report was submitted by the authors on June 10, 1977.

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SUMMARY

GENERAL

During the course of this study visits and interviews were made at mine sites. In addition visits were made to consultants and contacts were made with many people engaged in mining or indirectly involved. When asked if a study of this kind was worth while, the universal response was "yes".

There is very little literature on this subject. The subject embraces a number of disciplines such as structures, materials, mechanical design, dynamics, systems, corrosion, fatigue, ventilation, economics, operations, and a host of other areas.

The mining industry has a well founded, deeply entrenched resistance to change. There is good basis for this attitude. Things have to work in an extremely bad environment performing giant tasks, and they have to do so repeatedly and reliably. Also they have to be economically feasible.

After a thing or scheme or system has operated and is understood, then people feel comfortable with it and will resist changing it. It is doing the job.

There is much work to be done in this field. A review of the research areas that follow will verify this.

RESEARCH AREAS

Several areas that require further research have become apparent during the course of this study. Of the many possible areas, five have consistently remained as the most important. These five are: testing, guide geometry measurement, advanced suspension systems, dynamics, and structural configuration and materials.

Testing

Improvement of designs can be accomplished more efficiently if the anticipated loads are determined during full scale tests. Also, areas of improvement can be identified more readily when specific tests evaluate the performance of a component in actual use. The items to be evaluated could range from the skip body stiffener cross section to the static spring tension of guide rollers. Performance data of a full scale skip-guide system can be used to modify components to improve dynamic response and validate dynamic computer models. These models can be used to determine responses to changes in various parameters more economically than performing full scale tests.

The determination of in-service loads in conjunction with an acceptable design analysis can provide more efficient skip and guide designs. The forces experienced on skip components during routine operation can be determined in a number of ways. The measurement of load, strain and/or deflection can provide the necessary information to determine the loads applied to the instrumented component.

Accepted methods of measuring load are with the use of direct measurement transducers such as load cells and calibrated strain gages and indirect measurement transducers such as linear and angular displacement transducers. The South Africans have utilized similar testing to identify the forces experienced in the buntons, guides, and guide rollers. The results of these tests have resulted in establishing design criteria.

The degree of dynamic stability that the skip demonstrates as it moves up and down the shaft has an effect upon the hoisting speed and the amount of guide and skip maintenance required. The interaction mechanism between the skip and guides is very complex. One way to evaluate the dynamic stability of a particular skip-guide system is to measure parameters which give an indication of stability. These parameters are acceleration, velocity, and displacement and can be measured using appropriate transducers during tests at varying hoisting speeds. Components such as guide roller springs can be changed and the test performed again to determine if the new condition is more stable and eventually determine the optimum component configuration for best dynamic stability. Another method of evaluating dynamic stability is with the dynamic computer model developed in Section V. In this method full scale data are required to validate the computer model.

Guide Geometry Measurement

In response to the questions directed at vertical shaft skip and guide systems in the United States, nearly half of the problems reported were excessive guide roller, guide shoe and guide wear. The effect of these problems is that hoisting speed is reduced and therefore production is reduced. It is logical that an effective maintenance tool for vertical shaft guides be developed. A system that surveys the guide faces can provide information that can be used to plan routine guide maintenance and point out critical areas that require immediate attention. A system of this type has been in use on the nation's railroads for several years and has proved most beneficial in reducing unnecessary maintenance and identifying critical locations where without immediate attention a derailment would be likely. The following describes a survey system for vertical shaft guides.

The purpose of the Improved Guide Utilization by Instrumented Dynamic Evaluation (I-GUIDE) System would be to provide vertical shaft guide geometry data that can be used to more efficiently plan routine guide maintenance and identify those defects to be repaired immediately. The guide geometry data could also be used as input to the skip-guide dynamic computer model.

The I-GUIDE System would determine the position of the six guide faces along the length of the shaft with respect to the center of the shaft at the collar. This method would determine the short wave length disturbances in the guides, such as joint discontinuity and guide beam shape between buntons, as well as the actual change in the position of the shaft from true vertical along its length.

The inertial position of two adjacent and perpindicular guide faces can be determined by double integrating the output of accelerometers mounted to the guide roller frames.

The rotation of the skip body can be measured by an angular rate gyroscope and the output integrated once to provide absolute angular position.

A tachometer mounted to one of the guide rollers can be used to measure skip speed and distance down the shaft. The signal conditioning and the recording system could be located in a detachable package mounted to the skip or man cage. After the survey has been completed, the data can be plotted as a function of distance down the shaft.

The I-GUIDE System would consist of four portable packages. Two of the packages can consist of a set of guide rollers and necessary transducers. These two packages would clamp to an existing skip and/or man cage. The guide rollers would be adjustable to locate to any size guide. The third package can contain the gyro, signal conditioning, recorder and power supply. This package would clamp to the skip. The fourth package would contain the playback electronics and the plotter. After the survey was completed, the data tape could be placed into the reduction package and plots of the survey generated on site.

Advanced Suspension Systems

Investigation and development of advanced shaft suspension systems will provide reliable new systems as they are needed and at the same time promote a better understanding of existing systems.

Most current systems having guide rollers have a constant spring rate suspension system. Variable spring rate suspension systems have been used in other industries to provide improved dynamic stability for all inputs. The variable spring rate suspension system has been in use in racing automobiles for several years. The initial spring deflection gives a low or soft spring rate and thereby provides a smoother response to transient inputs. As spring travel increases, so does the spring rate. The increased spring rate can provide a smoother response to slower inputs such as the guide deflection between buntons. The response of a loaded and empty skip on a suspension system with the same spring rate is vastly different. If the suspension system is tuned to give the loaded skip dynamic stability, then the empty skip might be unstable. This has been the case with piggyback type railroad cars. This problem has been reduced by automatically changing the spring rate with load. A system having the same effect could prove beneficial in skip suspension systems.

Existing rigid guide systems use conventional symmetric structural shapes. Investigation of the use of other types of structural or asymmetrical shapes could reveal advantages, especially in terms of dynamic stability. Tapered guides and/or rollers could be configured in such a way as to use the inertia of the skip as a restoring force. If a 45-degree taper were used, only two guide rollers would be needed rather than the usual three rollers at each corner. Development of these types of designs could lead to improvements which would result in more efficient and stable skip and guide systems.

Dynamics

Areas where further development of the dynamic computer model should be pursued include:

1. Further study of the literature.
2. Extensive parametric studies using the existing model.
3. Correlation of dynamic field and computer data.
4. Study of various types of contact forces.
5. Other skip geometries.
6. Improved formulation of damping effects.
7. Kinematic nonlinearities and x - z coupling.
8. Development of stability and resonance criteria.
9. Friction forces and their effect on rope tension dynamics.
10. Rope guide model.
11. Offset guide roller model.
12. Guide rollers with variable and different spring constants.
13. Input from I-GUIDE field data for real guide geometry.
14. Rope dynamics interaction with skip dynamics.

Structural Configuration and Materials

Traditionally skips have been made as rectangular boxes from mild steel. More recently some of the higher strength low alloy steels have been used, but generally skips have not changed. Two areas require additional research: one is the skip geometry such as cylindrical versus rectangular, and the other is new materials to be used with new configurations.

The basis for new configurations should be through the use of analysis. Many designs and ideas should be subjected to stress analysis to find the good and bad features. By establishing one or more permissible stress levels (depending on function and location) new concepts should be brought under close scrutiny.

A number of computer programs have been written for analysis of plates, plate-type structures, and shell structures as found in the skip body. Other programs have been written for frame structures such as found in the bail. The more advanced programs have plate, shell, and beam elements whereby a single program could be used to analyze the skip and bail assembled.

Initial study should be for static loads, followed by the dynamic loads during hoisting, filling and dumping. Although Section IV shows the means of providing and the reasons for using a variable stiffener spacing, very few skips are designed for hydrostatic type loads.

For any work of this kind to be successful, a set of goals should be established. Reduced weight, increased reliability, low cost, low or no maintenance are generally the designer's objectives when designing a new skip.

The sources of information from which engineers, designers, draftsmen, et al, can draw are quite limited. The two volumes of the new SME Mining Engineering Handbook cover skips, cages, shaft guides, and hoist accessories from page 15-63 to 15-69 and contain six figures, four of which are photos. Older references generally show types of construction no longer being used.

A handbook and manual of skip designs are needed.

Along with new structural design concepts the use of new materials should be studied. Materials for both the body and abrasion resistance require more investigation. Currently 80 percent of all shaft conveyances operating in Canada are made of aluminum. Foreign countries are using rubber-like abrasion resistant materials.

No one has seen a cylindrical skip in operation, let alone a fiberglass overwrapped cylinder.

A main problem, frequently observed during this study, is guide rollers requiring frequent replacing and excessive guide wear. Tires, bearings, and spring materials must be selected for the shaft environment.

A testing program, after adequate screening, is needed to use laboratory results from good simulation to select the best abrasion resistant material and guide roller components.

The proper coordination of new designs and new materials should improve the performance of deep vein skipping. This will happen from a research program considering the interrelationship of both structures and materials in the shaft environment.

CONCLUSIONS

Before significant changes will be made in today's skipping practices, the new skips will have to be demonstrated. Such demonstrations may not be possible at a production shaft where the uncertainty of new concepts may present problems in maintaining production.

Guide surveys conducted in a matter of a few hours will provide the industry with a much needed tool. A good deal of interest has been expressed for such a system.

New concepts in cylindrical skips and using other materials have the potential for a very low payload ratio and improved longtime performance.

Consistent design and analysis will produce skips and guides such that if changes are needed, there will be a basis to determine how much change and why.

By combining the knowledge from several individual sources such as the mine visits, the consultants, the literature, and from experience, an important subject, skip and guide design methods, has begun to evolve.

I. INTRODUCTION

Increased capacity and more economical skipping of ore and waste from deep shafts are described in this report, which is divided into two sections. The first portion, Sections II through VIII, provides the background and analysis, identifying the areas that need improvement. The second portion is a set of recommendations that provides design data for better skips and a rational approach to guide design and selection presented in Sections IX through XIII.

Well designed skips and guides are the backbone of successful longterm shaft operation. To date the subject has very little history of investigation and a small amount of documentation in the technical literature.

In the deep mines of several United States mining districts, large, well equipped shafts often cannot be justified. Production from deep levels must be maintained through the use of fast running skips. Nearly every shaft user has found it necessary to continually modify and redesign skip and guide practices to meet individual circumstances. Well designed skips and guides will do much toward diminishing overall shaft maintenance and repair. A wide variety of practices is acknowledged to exist, and this subject area has received research investigation.

Shaft conveyances and supporting structures affect the ventilation flow within the shaft.

This study does not examine the construction nor practices of wire rope usage nor the safety devices associated with skip and guide design.

A. HISTORY OF MINE SKIPS

In reviewing the history of mine skips we note that the word "skip" appears to be of Cornish invention and originally was applied to inclined shafts. The word has been spelled "skeet", "skep", or "skipp". Skip problems applicable to this contract can be dated from 1856 when the Academy of Sciences of Brussels proposed the following question, and a special prize for a satisfactory answer to the problem - "Indicate a complete practicable method for extending the exploitation of collieries to a depth of at least 1,000 meters?" One of the suggestions was hoisting by means of cables made of iron wires.

American skip practices can be traced to about 1860 when skips were recognized as superior to shoveling and chute arrangements. Dr. R. W. Raymond described such ore transfer at the Comstock Mines in 1870 as follows: "The mineral having been taken to the shaft is either dumped in a pile and then reshoveled into the bucket, or skip, or is dumped through a chute directly into the skip and the empty car returned to the face. This necessitates rehandling of the mineral, which, when it reaches the surface must be dumped again into a car or wagon. These, and other considerations have led to hoisting the car and load together. This effects a great saving of time and labor, and wear and tear of apparatus. It is the method adapted in the mines of the Comstock Lode, and in all well appointed vertical shafts of any considerable depth elsewhere."

A famous lithography print by T. L. Dawes in 1876 entitled "The Belcher Mine" not only shows the elaborate square set system of the Comstock but also illustrates these shaft operations. Note too, that several Comstock mines reached depths of 3,000 feet by the 1880's.

Finlay, in his book on mine costs (1910), pointed out that even with the low wages paid shovelers to lift ore a height of only 4 feet, the practice of shoveling was costing Comstock mines about 25 cents per ton. Actually Finlay's costs for some mines show that over half the mining cost was consumed by multiple shoveling and hoisting. Thus ore hoisting costs, as well as the mechanics of shaft conveyances, have been a long and continuing problem to mine management.

Skip practices in the United States for the next quarter of a century were perhaps little more than highly developed bucket hoisting, or were a system of cage hoisting with cars until the South African diamond mines at Kimberly introduced the famous Kimberly skip. But note that Ihlseng's book Manual of Mining (4th ed., 1906) continues to discuss multiple shoveling (pg 225) while elaborating on hoisting by cage with loaded cars.

Milo S. Ketchum stated in the preface to his book Design of Mine Structures (1912) that the purpose of his book was to present a systematic discussion of the design of mine structures. Material on cages and skips (pg 30-36) illustrates types used and Figure 38 shows a retangular Kimberly skip indistinguishable from many in present use. We conclude that by the first decade of the 20th century the "art" of skipping was developed much as we know it today. Thus we see in other mining texts, such as Young's Elements of Mining, (4th ed., 1946, pg 214) that "the vehicles used in hoisting are designed for the shaft compartments and to conform to operating conditions and the service to be rendered". That sentence was also included in the first edition of 1916.

Even in Peele's Mining Engineers Handbook, (3rd ed., 1941, Section 12, pg 110) the sentence "design of vertical skips is largely a matter of experience and good practice, loads and stresses being usually more than taken care of by proper allowance for heavy wear and tear of operation", is exactly the same as written both the first edition (1917) and the second edition (1927). There is little change in text material between the three editions.

The SME Mining Engineers Handbook of 1973, Section 15.4 (pg 15-63) discusses skips, cages, shaft guides, and hoist accessories. Here we see the first emphasis towards modern developments, although only six pages are devoted to the entire subject.

Earlier books by Staley, Mine Plant Design (1949), and Tillson, Mine Plant (1938, reprinted 1976), should be considered as the best information sources on skip design up to 1950. Staley's text, Chapter VI, brought out two examples which typically show the problems of skip design. In the Anaconda example, a novel sliding door was designed into the lower side of the skip to overcome the exceptionally long Kimberly skip that was necessary because of narrow shaft dimensions. In the Star Mine example, a novel overturning Kimberly skip design was developed which dumped during the lowering cycle rather than the lifting cycle. These are perhaps extreme examples, but nevertheless illustrate the difficulties often encountered in skip design.

Excellent references that Staley and Tillson are, they do not cover important skip developments since 1950. In the next decade several new bottom dump designs were patented and introduced to the market and which now represent most of the skips in service (outside of Kimberly skips still commonly used.) The leading manufacturers are as follows:

Card Corporation, Denver, CO.

Dorr-Oliver-Long, manufactured by Long Engineering Works, Ltd., Orilla, Ontario

Lakeshore Engineering Company, Iron Mountain, Michigan

Holmes Bros., Inc. Danville, Illinois

Sala Machine Works, Cooksville, Ontario

Saunders type skip, Toronto, Canada (no company assignee to patent)

Wabi Iron Works, New Liskeard, Ontario

Connellsville Corporation, Connellsville, PA

A great number of other companies also fabricate skips, cages, and other hoisting conveyances, such as:

Machinery Center, Salt Lake City, Utah

Cottonwood Steel Corp., Salt Lake City, Utah

Coeur d'Alenes Company, Wallace, Idaho

and several other companies primarily serving the coal industry. Many mining companies construct custom skips in their own mine shops.

Motivation for the introduction of these new skip designs was a general industry trend towards production from deeper mines which were being developed in the United States and Canada during the 1950's. Thus by increasing shaft depths from only 2500 feet to 3500 feet, some mines became uneconomical simply because of the length of the hoisting and dumping cycle time. Some advantages of bottom dump skips were reported as: 1) to reduce the height of mine headframes due to the Kimberly skip; 2) to eliminate the severe shock and impact stresses in the headframe during Kimberly skip dumping; 3) to decrease the creep time into the dumping scrolls; 4) to reduce the overall duty cycle time in hoisting; 5) to obtain winding characteristics favoring a more economical hoisting plant; 6) to adapt skips to hoisting on rope guides; and 7) in general mostly to simply reduce the time for skip dumping. In one example, cutting only a few seconds from the dump time increased hoisting production by 7 1/2 percent.

This brief history of mine skips revealed that the subject of skip and guide design has neither a history of investigation, little documentation in the technical literature, a small field of patents, and virtually no research and development efforts except perhaps for the South Africans.

B. THE SKIPPING OPERATION

Skips running in vertical shafts bring to the surface the ore and waste from deep underground mines. Production schedules are based on the skips remaining trouble-free. Most modern mines use bottom dump skips that can be as small a sixty cubic feet to as large as eight hundred cubic feet. Swing-out and fixed body designs are used.

Guides for skips are made of wood or steel or wire rope. High speed skipping requires well alined wood or steel guides and properly tensioned rope guides.

Shaft cross sections can be rectangular or circular or elliptical. Shaft walls may be supported by timber sets, steel sets, gunite, shotcrete poured concrete, or metal liner plates.

A skipping operation consists of filling, hoisting, dumping and returning down the shaft to be filled again. The filling operation can be volumetric or by weight, and the ore may or may not be crushed underground. High dynamic loads and rope stretch are encountered during filling. If rope stretch is sufficiently great, proper allowance must be made for the skip going downward during the fill. Chairing the skip during loading has proven to be an unsatisfactory procedure resulting in skip problems due to the higher impact loads. A skip at the loading station is shown in Figure I-1.

Hoisting requires the skip to be guided as it goes up the shaft. If guide problems occur, generally due to misalinement or becoming loose, then the hoisting speed is reduced and hourly production is decreased.

Dumping the skip may occur either underground or in the headframe, as shown in Figure I-2. The usual dumping arrangement is by means of scrolls for a swing-out body skip and by air cylinders for a fixed body skip, although the schemes have been interchanged.

One of the recurring problems is spills in the shaft. This can happen during loading, during hoisting, and while dumping. If the problem is persistent, special arrangements are made in the sump to remove the spilled muck and bring it to the surface.

C. SOURCES OF INFORMATION

Several sources of information have been used during this study. Early literature exists as text material published in the late 1910's to the 1950's. Technical papers and journal articles have appeared since the 1950's. South African literature began to appear in the 1960's on such subjects as air flow, rope guides, skip-guide dynamics, and experimental work in laboratories. German literature spans about the same time period, being concerned with analysis of skips and guides interacting dynamically. A heavier emphasis on skip guide dynamics appeared in the Russian Journals from about 1965. Streamlining buntons for better air flow has been in Canadian literature starting in the 1960's. Work on rope guides has been published in England since the 1960's.

More immediate sources of information have been the mine visits, visits with mining consultants, and general contacts with those directly engaged in or supporting mine activities. Many of the general observations are the result of having the same comments repeated a number of times. Experience with the overall design, analysis, detail drawings, and fabrication of skips, scrolls, and loading pockets, plus other vertical haulage equipment has also contributed to the sources of knowledge.

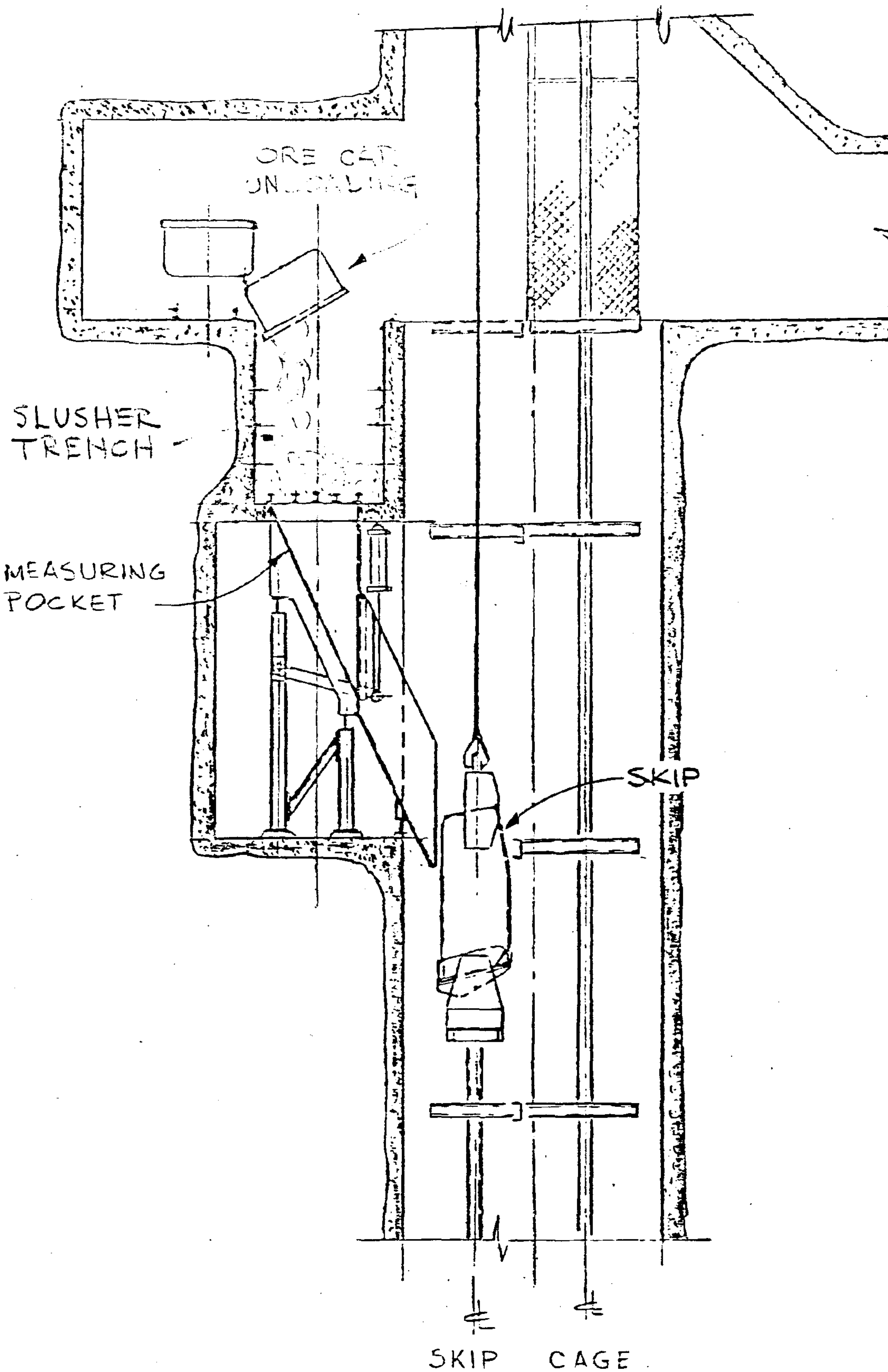


Figure I-1 Skip at Loading Station

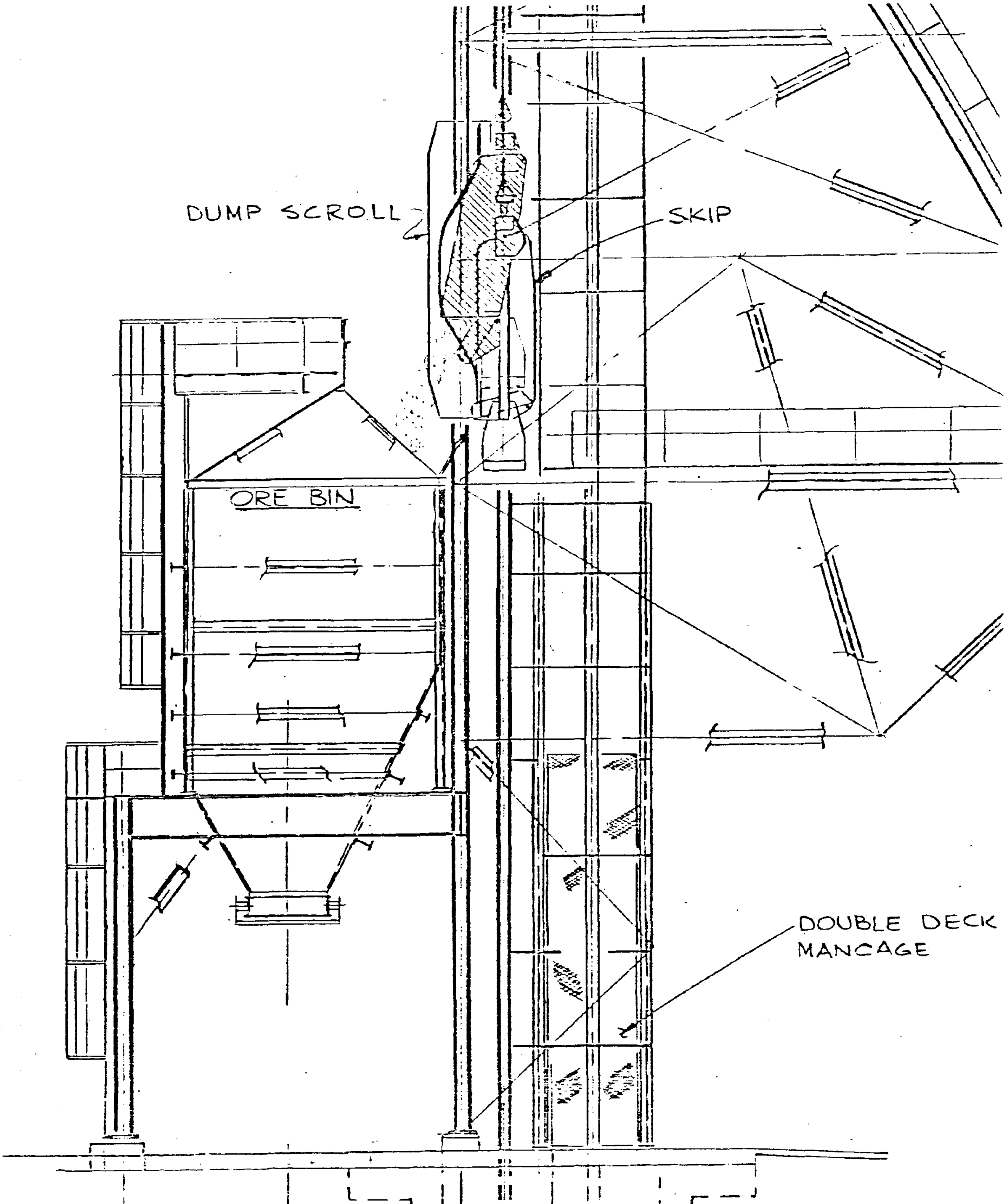


Figure I-2 Skip in Headframe

D. PAYLOAD RATIO

The ideal skip should weigh practically nothing compared to its haulage capacity. That ideal has not been realized; however, there are data to show how much skips weigh with respect to their payload. Three sources of information have provided weights for about 35 skips and their capacities. By using the payload ratio of skip weight divided by skip capacity (in the same units), we can arrive at a number to compare large and small skips on the same basis. This number multiplied by the capacity gives the skip weight.

Payload ratio as a function of skip capacity in tons is plotted in Figure I-3. Data were obtained from South African literature, results gathered during this program, and manufacturers' data. The spread of the payload ratios is from 0.4 to more than 1.9. Most of the points appear to be in a band between 0.6 and 1.0. If this number is a low number and the performance of the skip is satisfactory, then the skip is well designed because for the same rope pull the skip has a higher payload. Conversely, if the capacity remains the same, then the lighter skip (low payload ratio number) results in smaller rope pull, permitting perhaps a smaller rope, which again reduces rope pull and may result in a smaller head sheave and possibly a smaller hoist. The economies can be significant. It must be remembered that all weights are multiplied by factors of safety so that each actual pound represents several pounds of design load.

A recent paper has revealed that 80 percent of shaft conveyances in Canada are of aluminum. A payload ratio of one-half is commonly accepted. This study considers ways of achieving a low payload ratio. This study also considers the shaft environment and the effects of cycling on the structural and mechanical elements that make up a skip.

E. DYNAMIC COMPUTER PROGRAM "SKIP"

As part of this program a dynamic computer program has been written and run to study the interaction between skips and guides. The variables that affect the dynamic forces are skip weight, location of center of gravity, location of guide shoes and rollers, guide roller spring constant, damping, velocity, properties of the guides, and bunton spacing.

F. CONDITION OF SKIPS AFTER SHAFT USE

A skip after several years of service shows signs of wear. Several photos of a single skip prior to refurbishing illustrate some of the problem areas and also reinforce the need for conservative design.

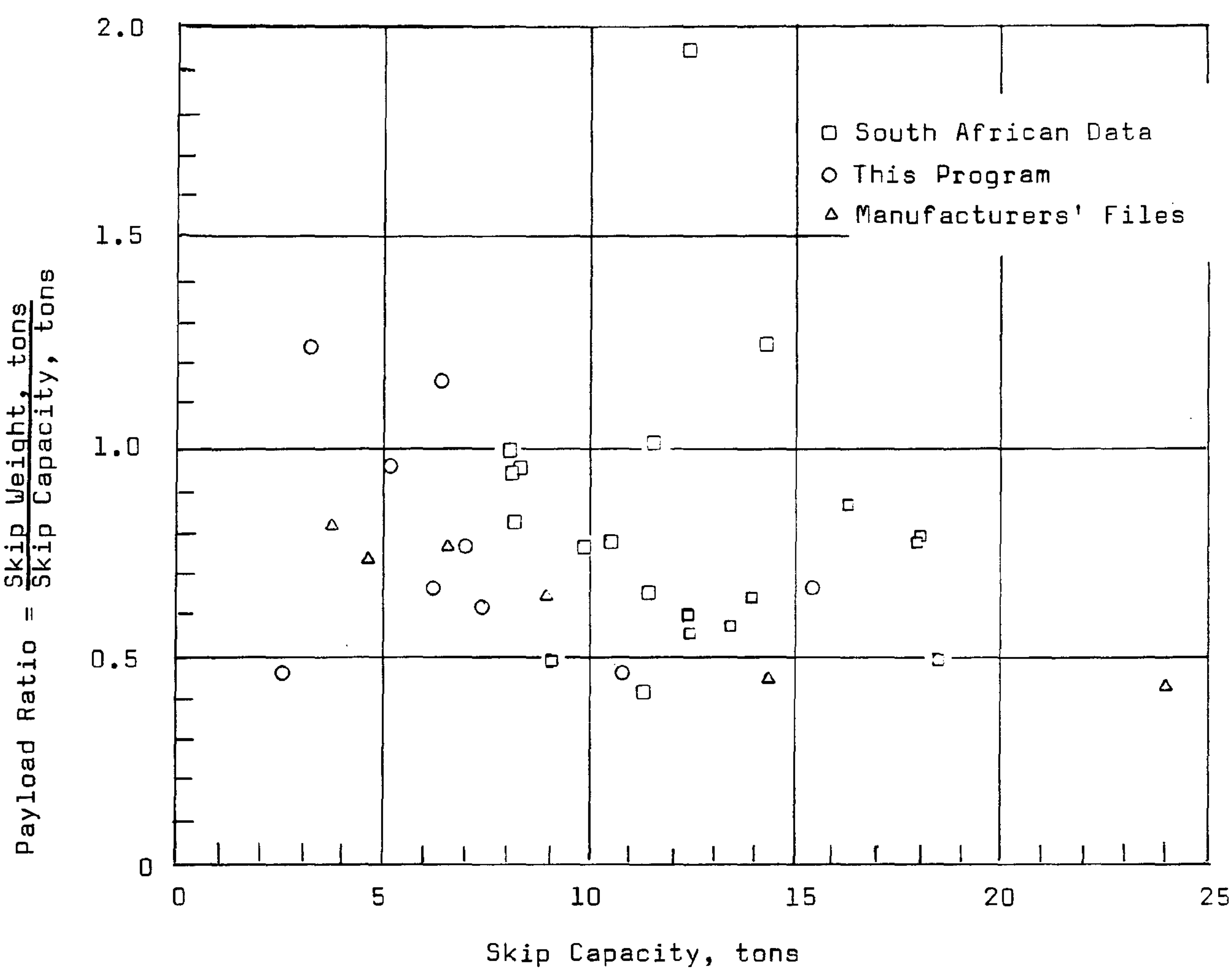


Figure I-3 Payload Ratio as a Function of Skip Capacity

Figure I-4 shows broken and bent door and bail details. The bail has been bent; welds are separated at the corner of the door; the door roller support bracket is bent and the cam plate on the bottom of the door has separated from the door.

In Figure I-5 the guide shoes are shown worn off, a box stiffener inside the body has separated, and large amounts of the crosshead have been worn away from loading.

The need to establish design methods, procedures, philosophies as related to skips and guides is apparent.

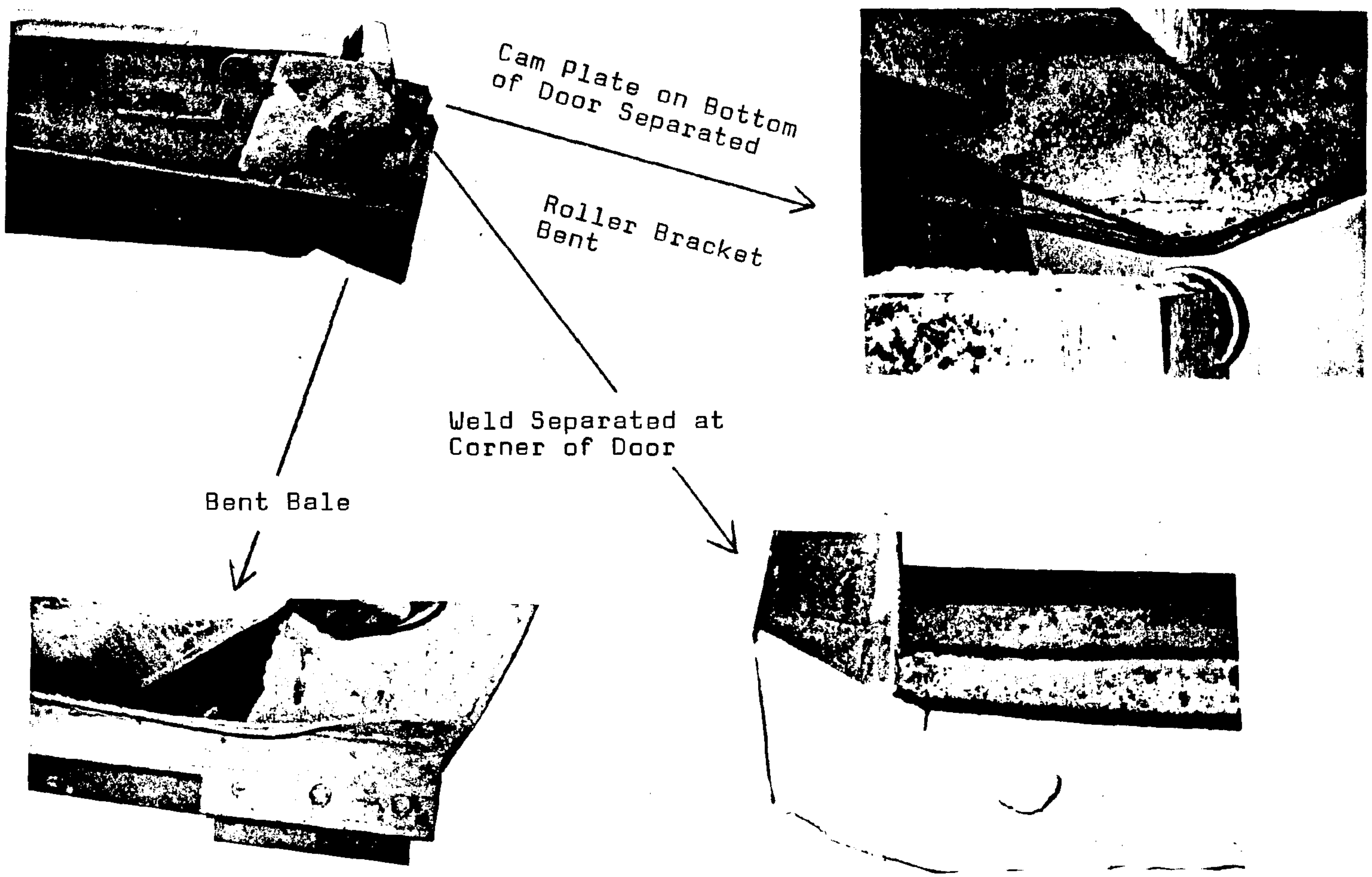


Figure I-4 Broken and Bent Door and Bail Details

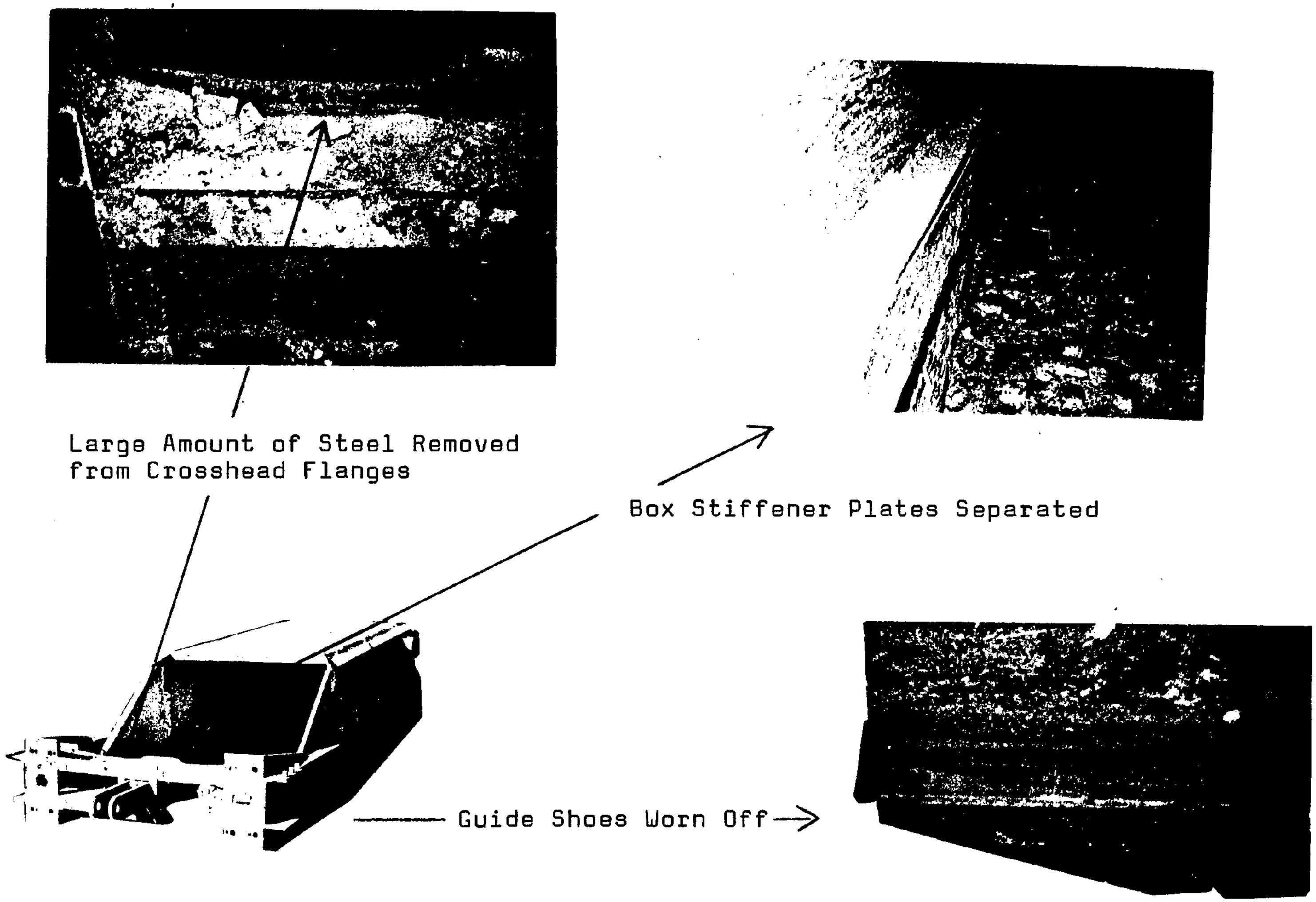


Figure I-5 Wear of Skip Crosshead, Stiffeners, and Guide Shoe after Service

II. SKIP AND GUIDE SYSTEMS

A. SWING-OUT BODY SKIPS

There are three types of swing-out body skips. They are the Saunders (Card Type "B" design), Jeto, and Rollamatic and are shown in Figures II-1, II-2, and II-3, respectively. The Jeto differs from both the Saunders and Rollamatic in that the dump door is attached to linkages that "over-center" between the closed and dump positions. The door on the Saunders and Rollamatic open while being supported by a roller. The door pivot on the Rollamatic is not located at the extreme back of the skip like the Sanders or Jeto, but more in toward the center. This concept was incorporated to eliminate the nut-cracker action when attempting to close the door that results from incomplete dumping (Ref. II-1).

B. FIXED BODY SKIPS

Three types of fixed body skips are shown in Figures II-4, II-5, and II-6. Figure II-4 is a Sala type skip, and the dump door is actuated by an air cylinder. The door design on Figure II-5 is a concave arc gage type, and in Figure II-6 is the Card Type A design with a flap gate. The method of door actuation of the latter two designs is not shown.

C. SKIP CAGE COMBINATIONS

It is common practice at many mines to attach a manned cage either to the top or bottom of a skip. This provides a conveyance for hoisting a few men without using the man-materials cage. An arrestment device must be designed into a skip fitted with a manned cage.

D. GUIDES

1. Alinement of Wood and Steel Guides

Good alinement of wood and steel guides is desirable to reduce large skip-guide interaction loads that result from poor guide alinement. An example of good guide alinement found during this study was 1/8 inch in 100 feet variation from true plumb while the entire shaft was within 1/4 inch from top to bottom. On the other hand, a shaft with poor alinement was misalined nine inches within a 25-foot length in one area of the shaft. Ground conditions can seriously affect shaft alinement.

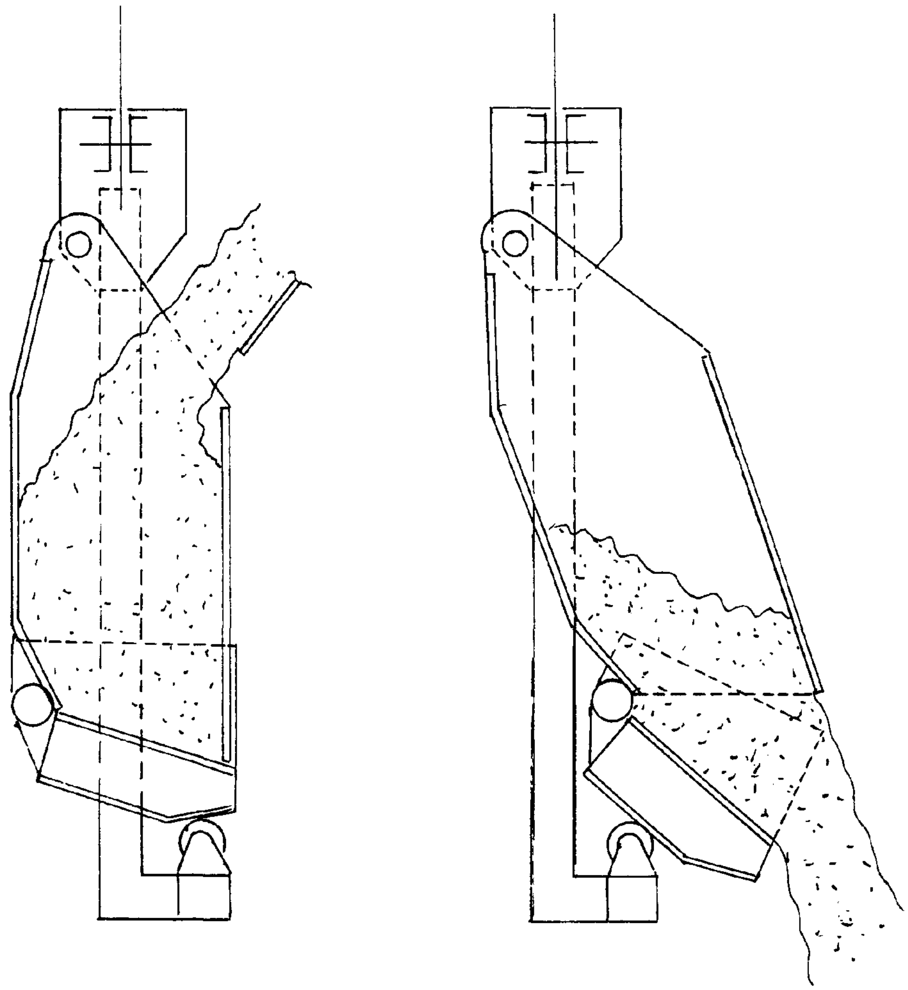


Figure II-1 Bottom Dump Skip with Swing-Out Body, Saunders Type, or Card Type B Design

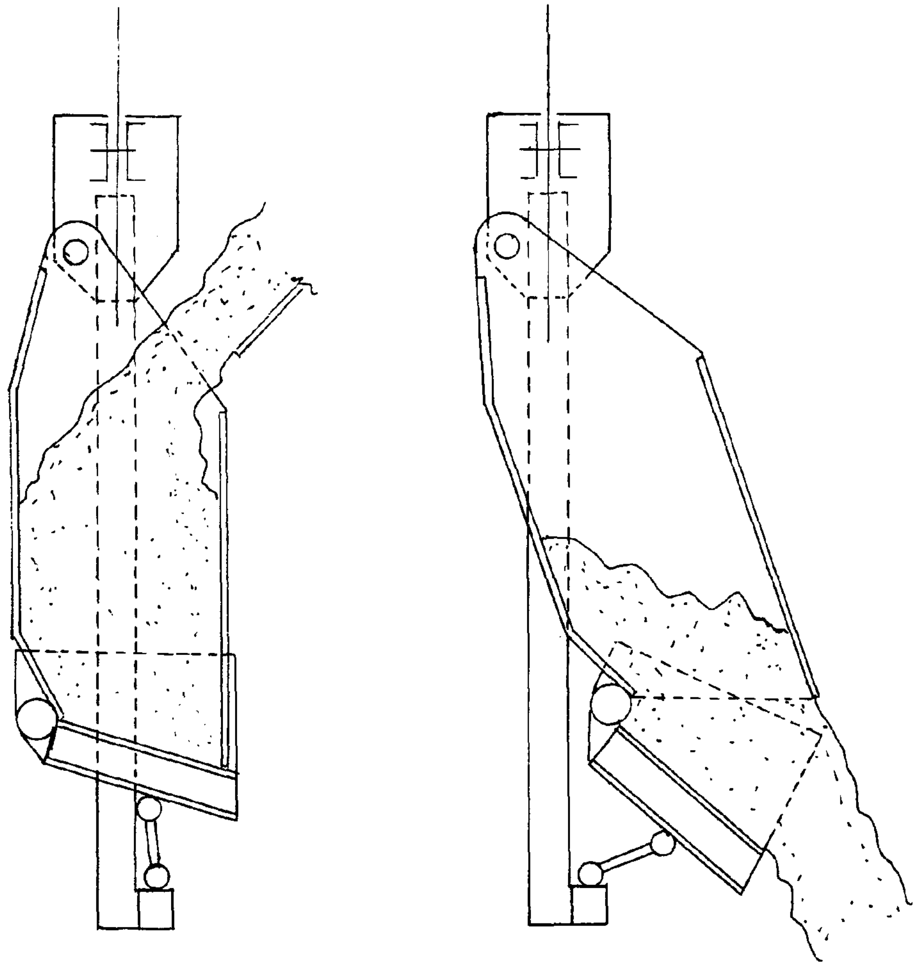


Figure II-2 Bottom Dump Skip with Swing-Out Body,
Jeto Type Design by Lake Shore.

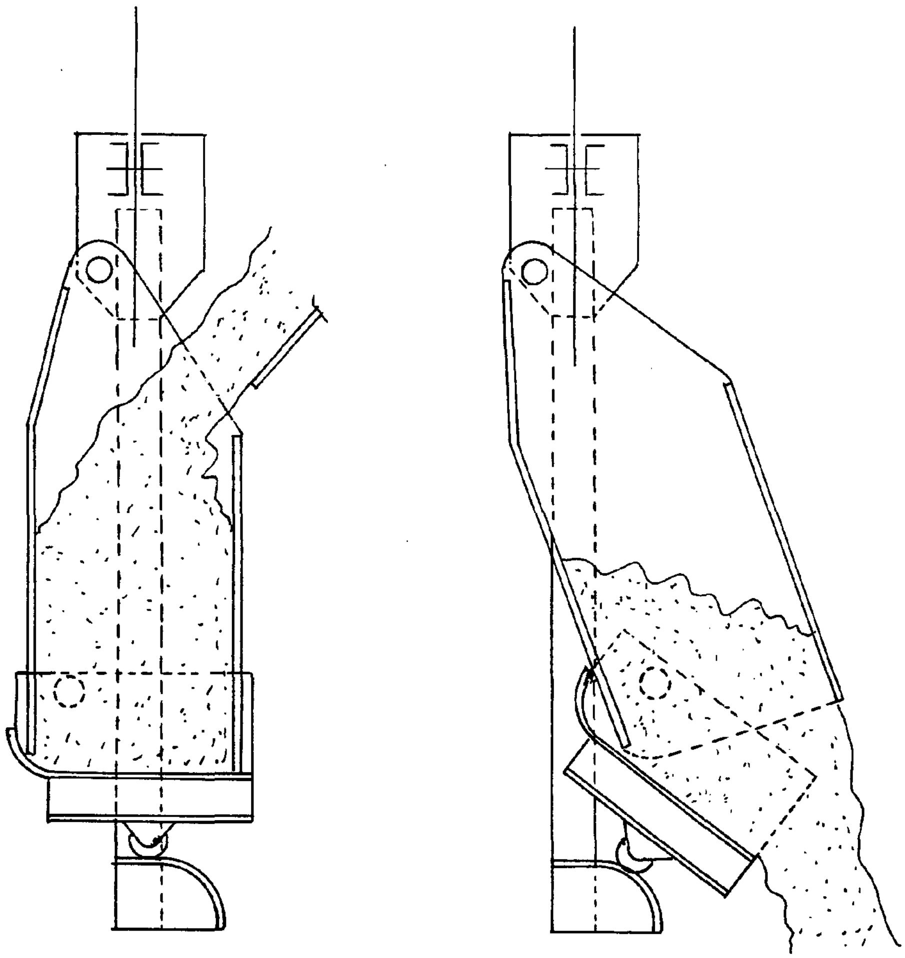


Figure II-3 Bottom Dump Skip with Swing-Out Body, Rollamatic Type Design.

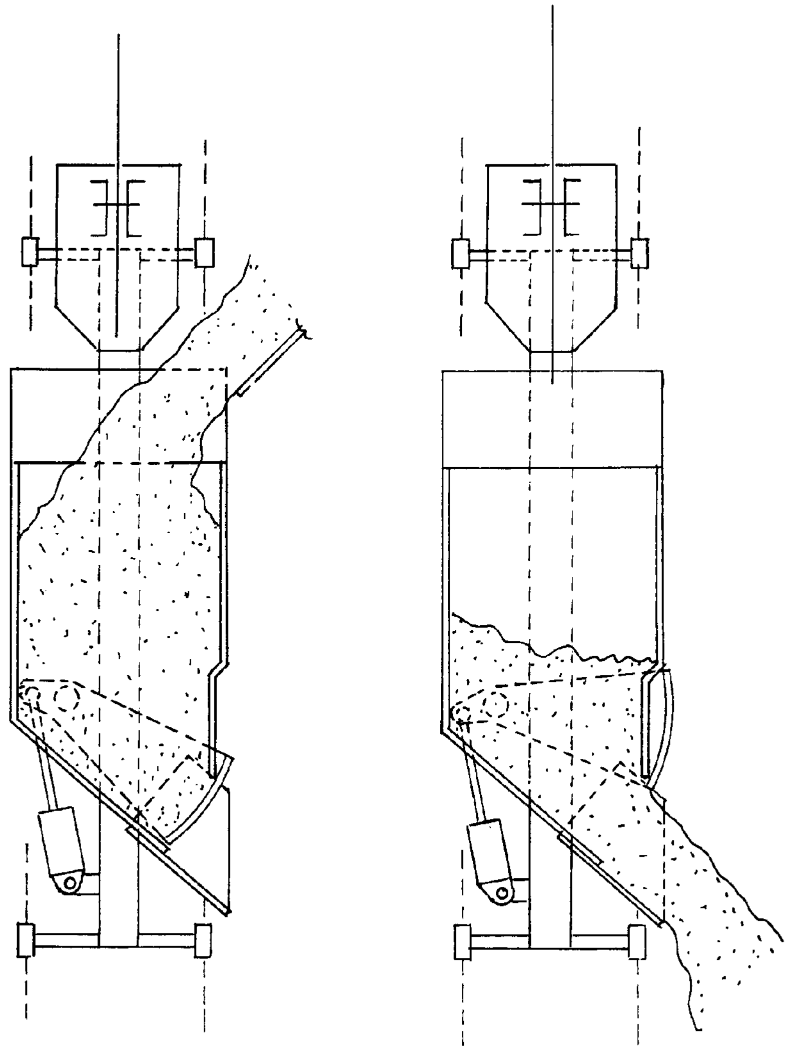


Figure II-4 Bottom Dump Skip with Fixed Body, Sala Type Design

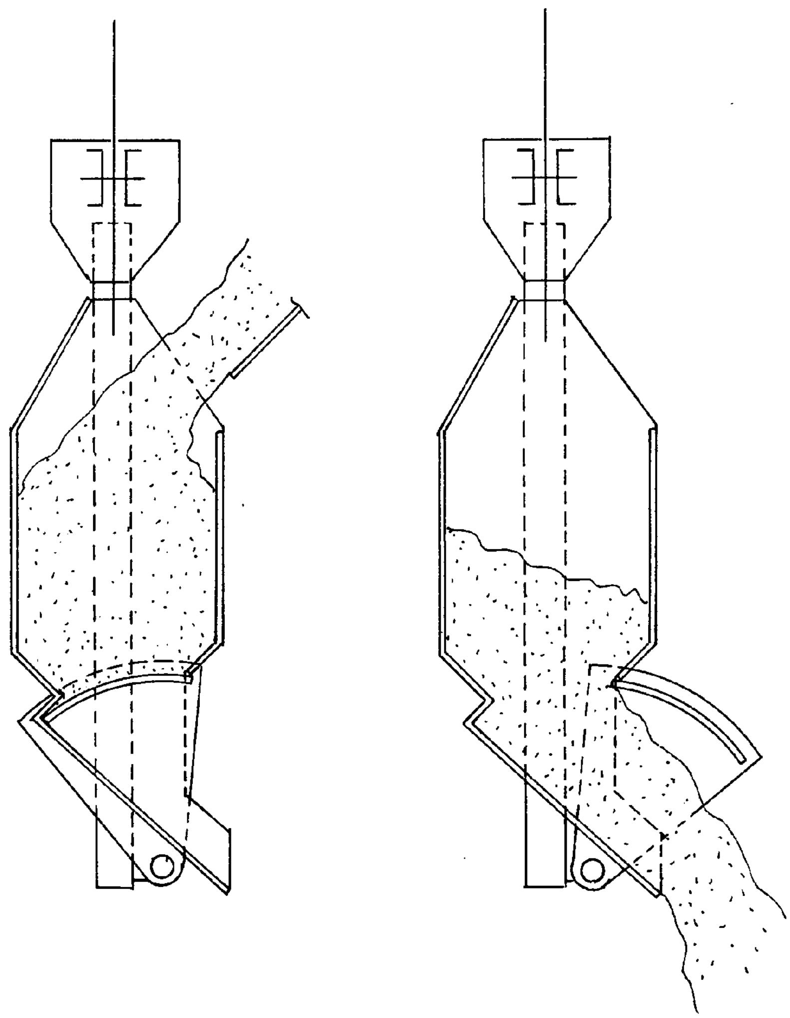


Figure II-5 Bottom Dump Skip with Fixed Body
Using Arc Gate Principle.

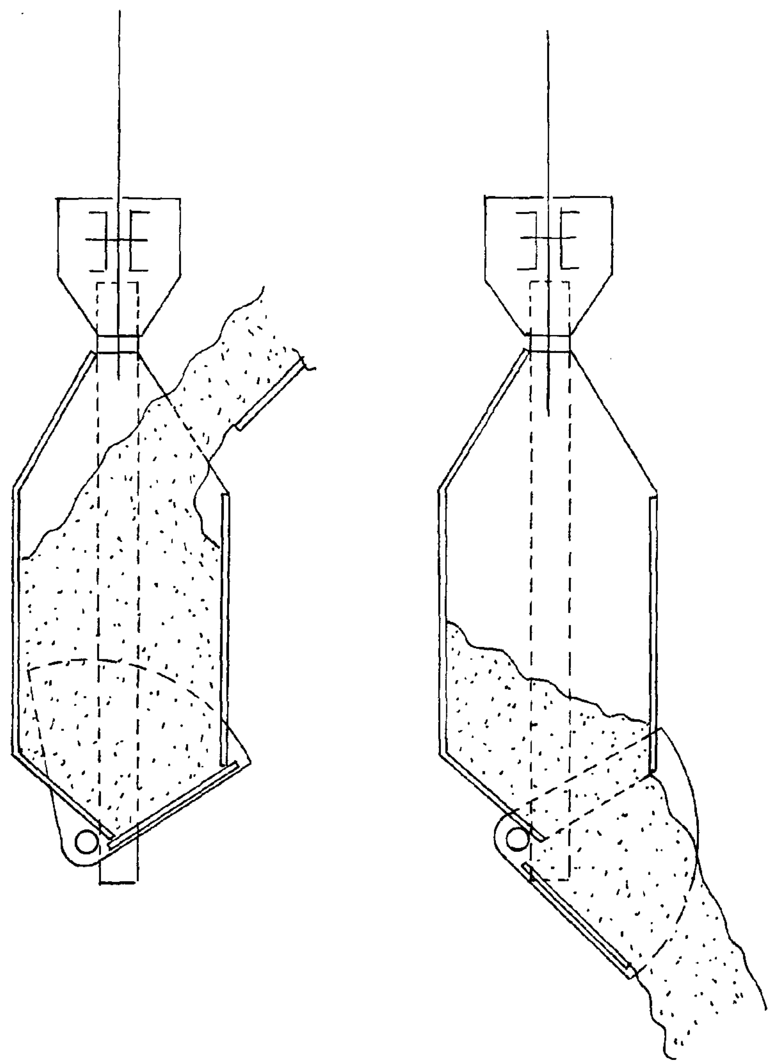


Figure II-6 Bottom Dump Skip with Fixed Body,
Card Type A Design.

2. Rope Guide Tension

Current Rope guide tension practice is to use end loads that give a minimum static safety factor of five (Ref. II-2). The curve in Figure II-7 illustrates the tension load as a function of rope length for this safety factor. The actual data points, also plotted on Figure II-7, give a sample of tension values being used (Ref. II-3). The actual tension of each rope is varied about the design value in the guide system by ten percent or half a ton to provide different natural oscillation frequencies.

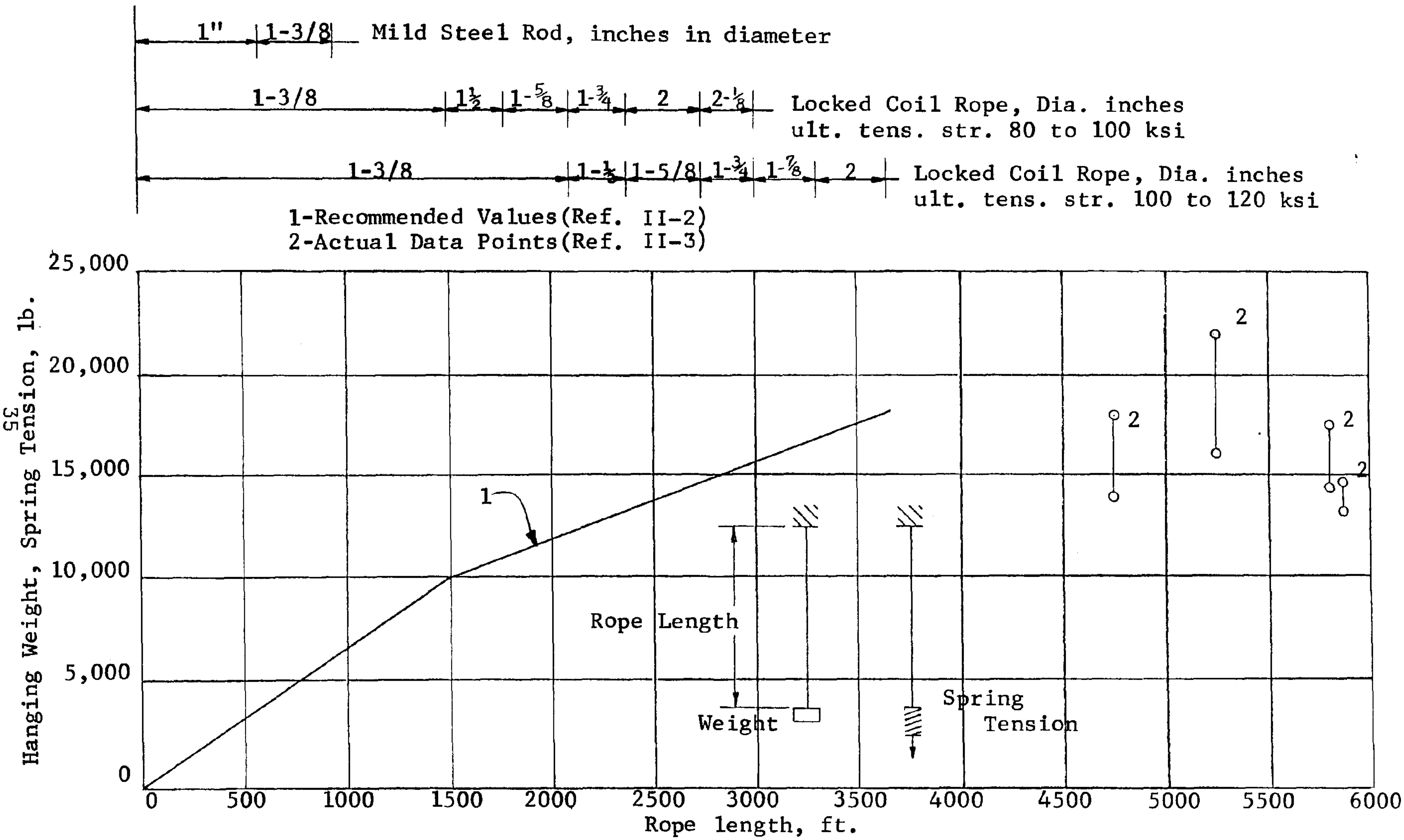


Figure II-7 Tensioning of Rope Guides and Rubbing Ropes

III. MINE VISITS AND OVERVIEW OF PRESENT SKIPPING PRACTICES

As part of this study, visits were made to nineteen mines in eight states. Some were selected for representative practice and others as outstanding examples. In addition, more information on current practice, problems, etc. was determined from literature reviews, prior experience, and contacts with mining companies. Visits were made to the first nineteen mines listed in Appendix C, Table C1. The personnel who participated are listed. Contacts were made with additional mines making a total of 72 mines that contributed to this study.

As shown in Figure III-1, information regarding rectangular shafts shows their average depth to be greater than circular shafts. The primary reason is there are several very deep rectangular shafts. These shafts also make the average depth of the wood guides with rollers type shaft to be the greatest. There was information for only one shaft with steel guides without guide rollers. The average depth for all shafts is 2060 feet.

The average maximum hoisting speed for all shafts was found to be 1380 feet per minute. Figure III-2 illustrates that wood guides without guide rollers have the lowest average at 830 feet per minute. Rope guides have the highest average hoisting speeds. Again, although plotted, the steel guides without guide rollers have only one data point, making this system difficult to compare to other systems.

The average skip payload tonnage for all shafts is 9.1 tons, as shown in Figure III-3; however, there is a large difference between rectangular and circular shaft figures, the latter having an average of 11.7 tons and the former 6.2 tons. The wood guide systems carry less tonnage than steel or rope. The addition of rollers in the wood guide system shows a higher payload.

Figure III-4 indicates that the only set materials used are wood and steel. The rectangular shafts showed almost even use of wood and steel. The circular shafts almost exclusively used steel. In general, circular shafts are newer than the rectangular shafts and steel is being used for new construction. The overall trend is toward the use of steel; 71 percent of the shafts use it. It should be noted that none of the steel used had aerodynamic shapes which have been shown to improve ventilation efficiency. The average bunton spacing for all shafts was 7.26 feet. Fifteen foot spacing has been found to improve ventilation as shown by the South Africans (Ref. VII-4). Only three shafts, all circular, had bunton spacing greater than ten feet.

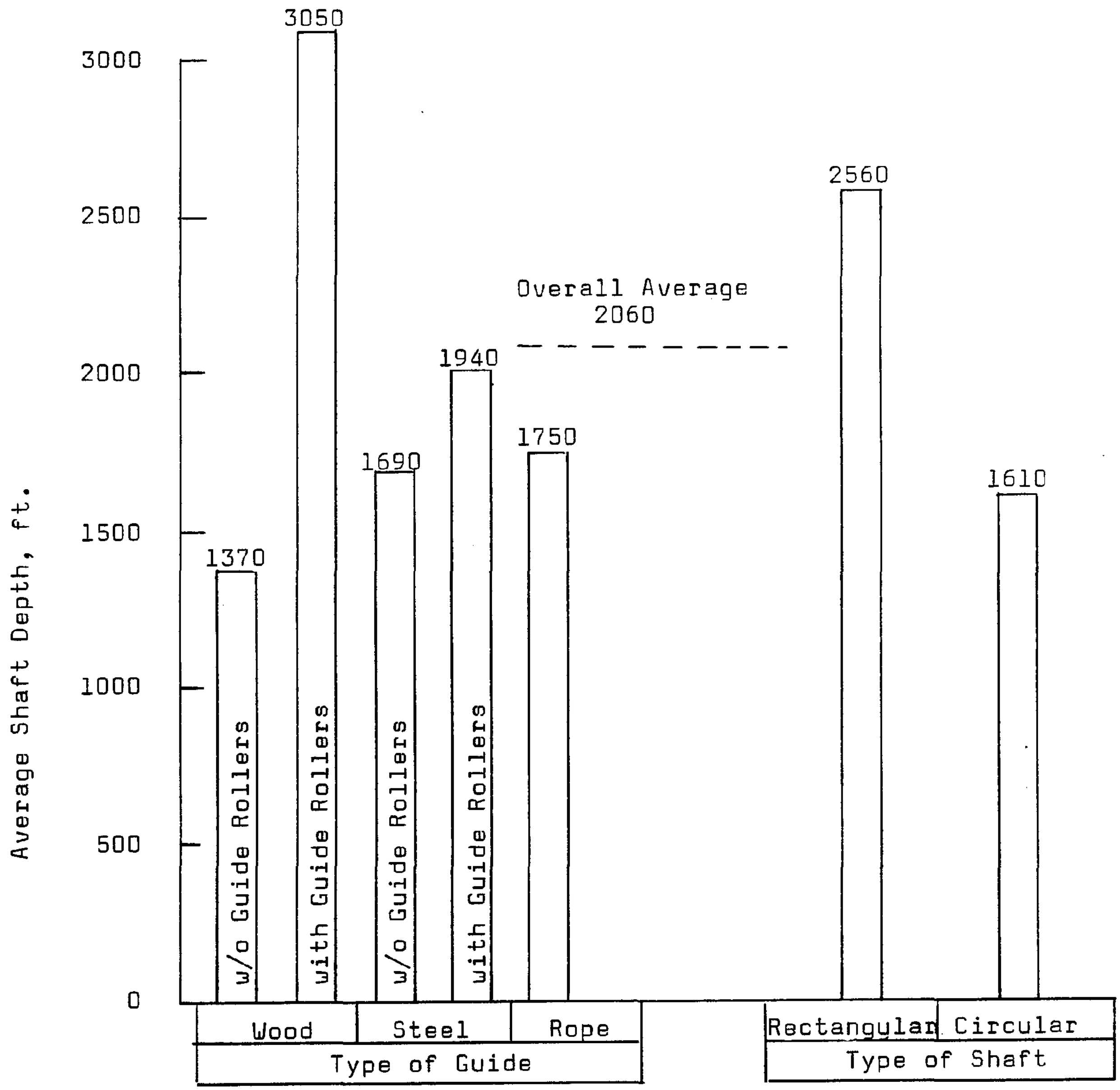


Figure III-1 Average Shaft Depth

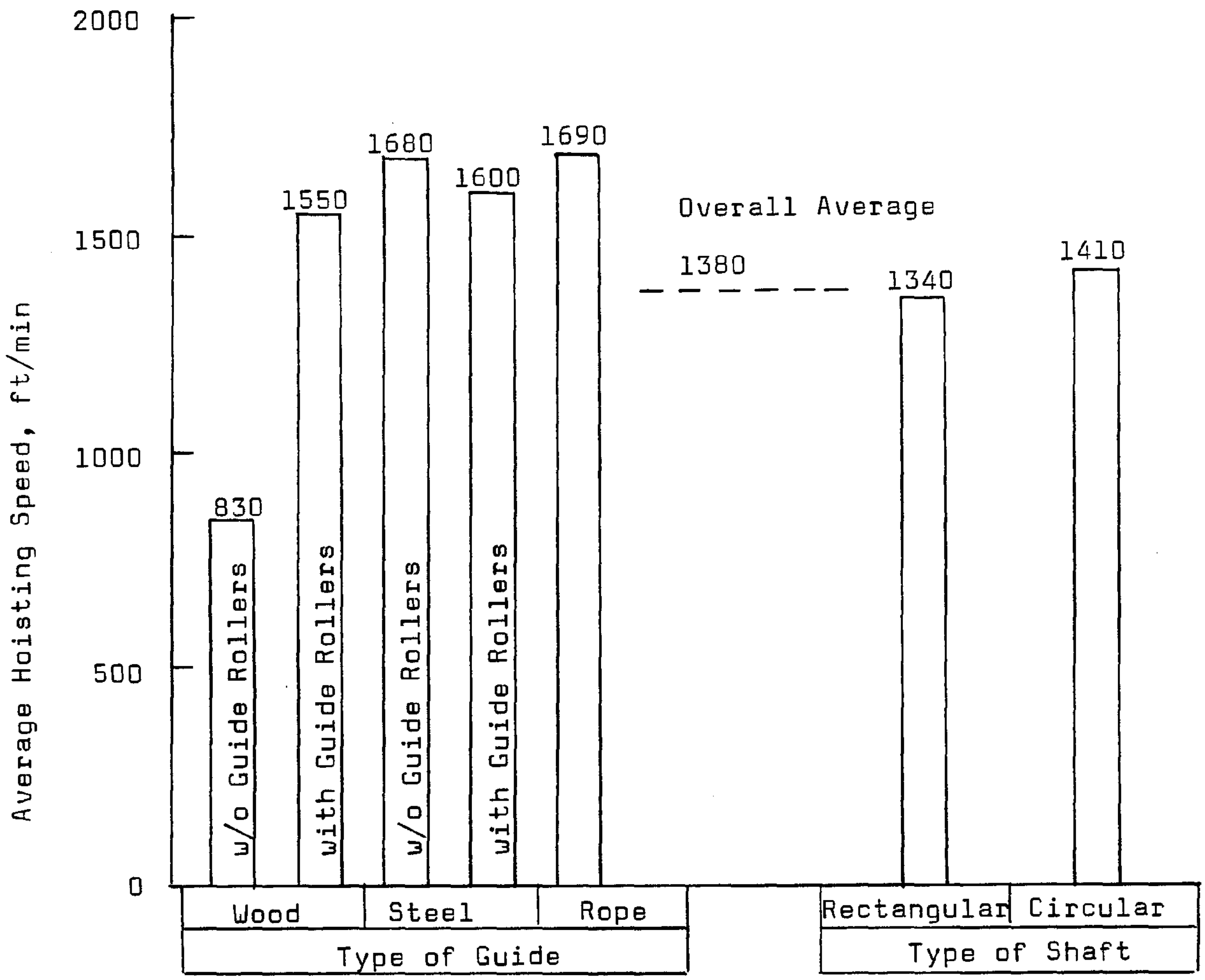


Figure III-2 Average Hoisting Speed

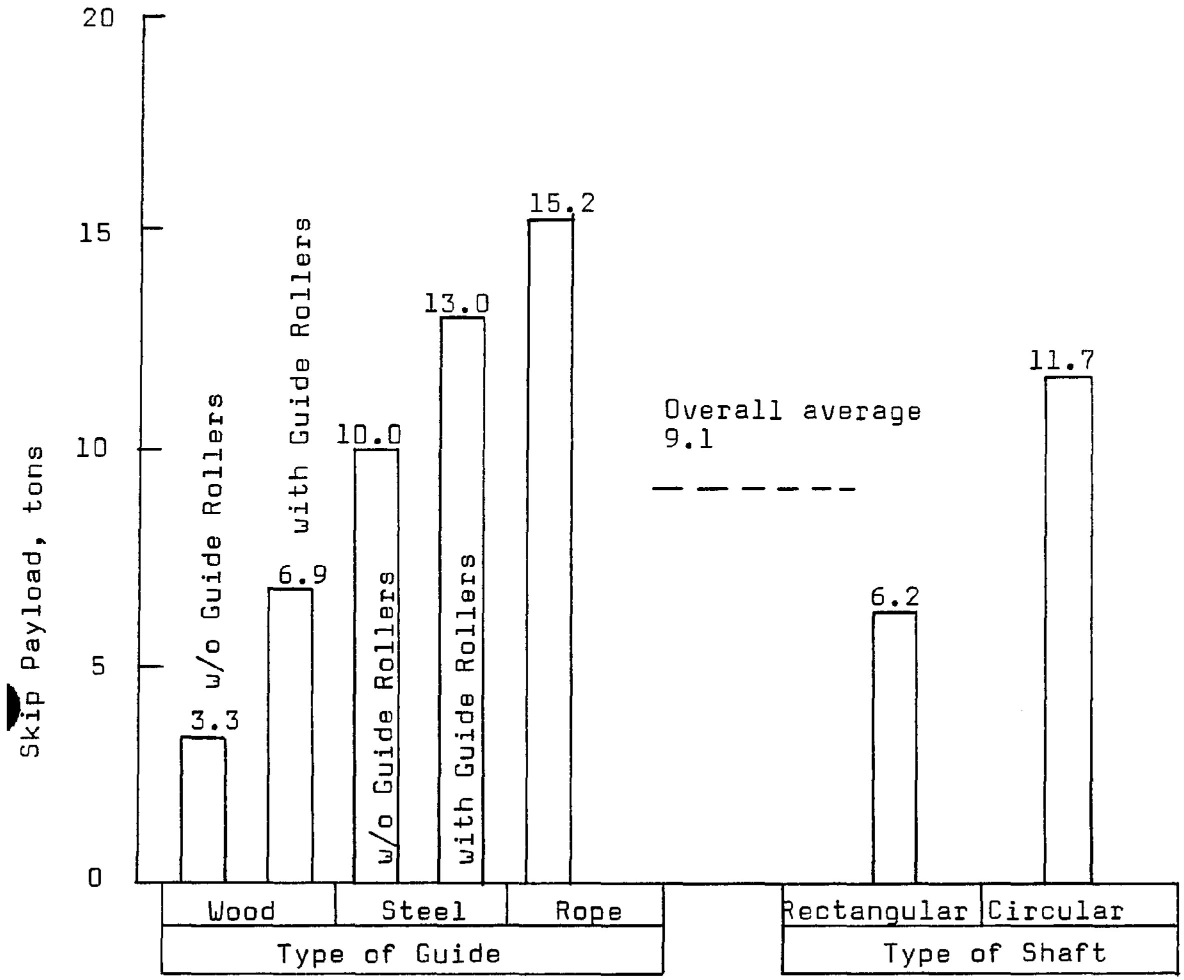


Figure II-3 Average Skip Tonnage

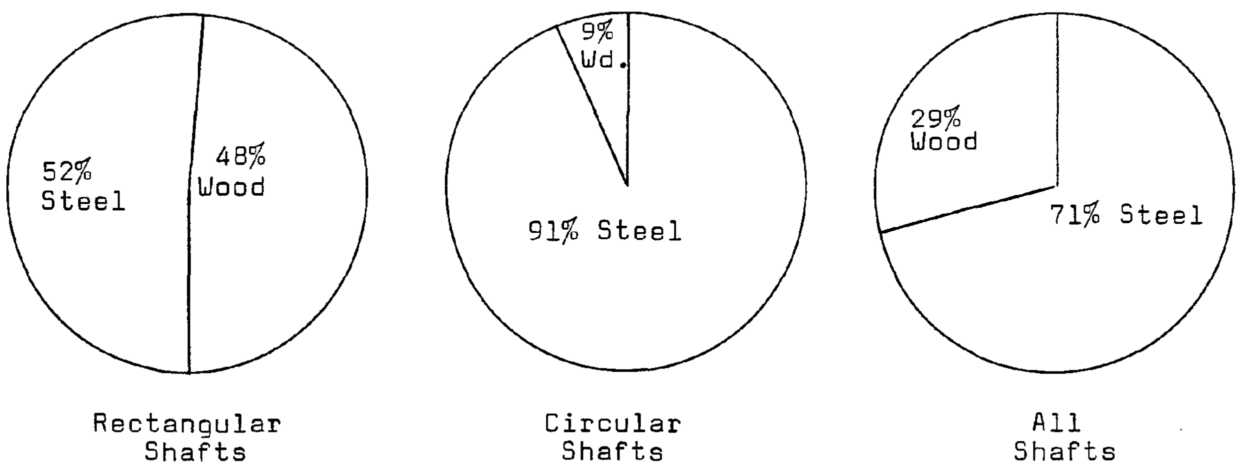


Figure III-4 Distribution of Set Materials

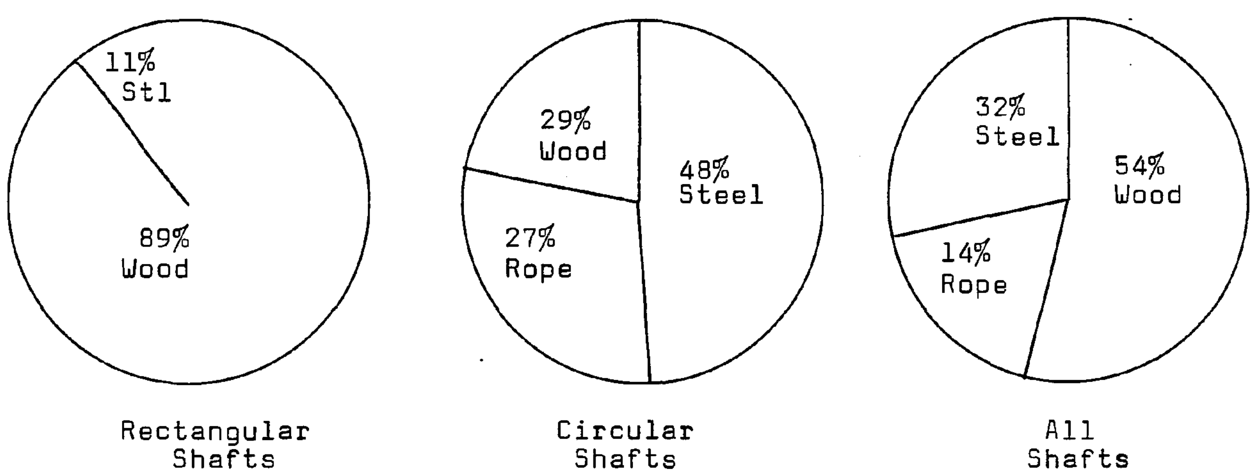


Figure III-5 Distribution of Guide Materials

While circular shafts had varying amounts of wood, steel and rope guides, rectangular shafts used only wood and steel. As shown in Figure III-5, wood is the most popular guide material used in rectangular shafts.

In circular shafts, steel is the primary guide material being used. Wood and rope are used in almost equal amounts for the remaining circular shafts. The overall distribution shows that wood is used most often, probably due to the requirement of man cages to have safety dogs.

The majority of skips used in rectangular shafts are the Kimberly, or overturning type. In circular shafts bottom dump skips are the predominant type used. The overall trend for all shafts is that 60 percent use bottom dump skips and 40 percent use the Kimberly type. Figure III-6 illustrates these distributions.

As shown in Figure III-7, a greater majority of circular shafts are lined than rectangular shafts. In fact, 95 percent of the rectangular shafts are unlined, whereas only 32 percent of the circular shafts are unlined.

Table III-1 gives a breakdown of the problems, by shaft shape. As shown, the major problem is with guides, guide rollers, and shoes. These problems appear to be due to higher tonnage and/or speeds being used to attain higher production.

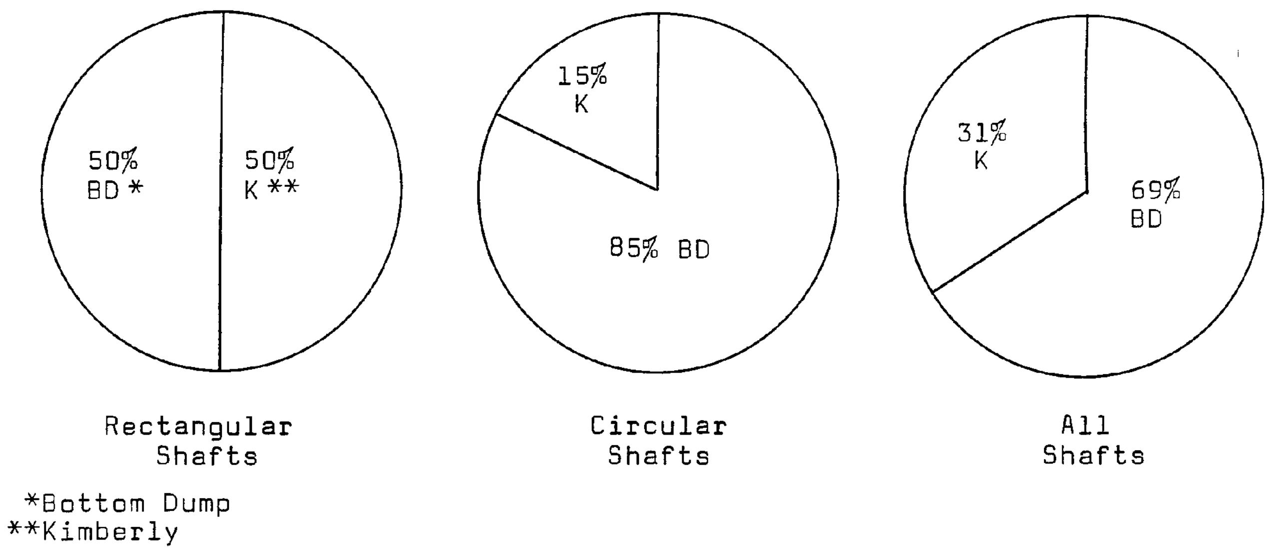


Figure III-6 Distribution of Skip Types

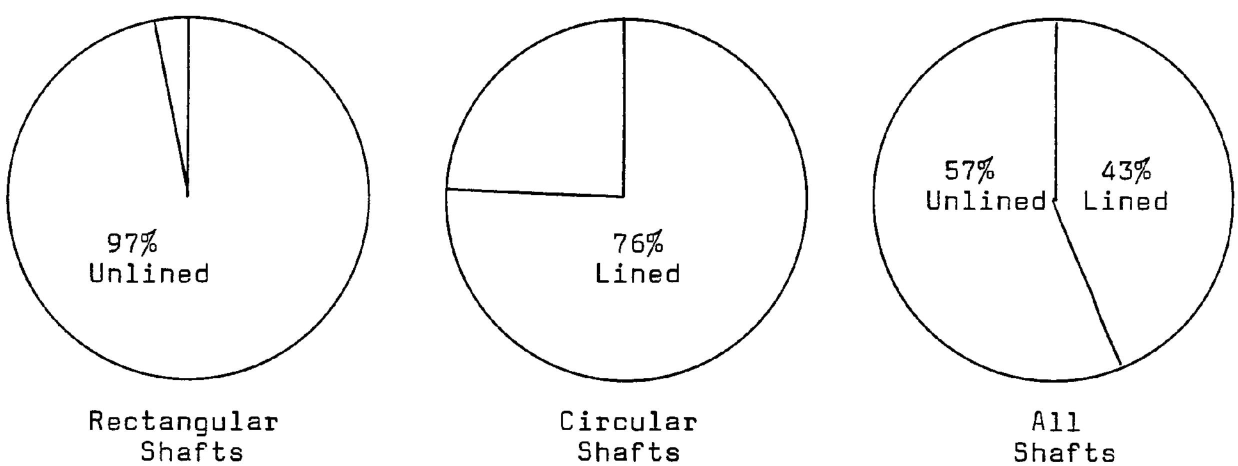


Figure III-7 Distribution of Shaft Lining

Table III-1 Problems With Skips

Shaft	Type of Problem								
Type	Rope	Spill	Unloading	Maint.	Sticky Muck	Structure	Roller/ Guide	Misaline	Corrosion
Circular	3	4	3	1	1	3	10	3	2
Rectangular	0	1	0	0	0	4	4	0	0
Total	3	5	3	1	1	7	14	3	2
Percent	3	13	8	2.5	2.5	18	36	8	5

IV. STRUCTURAL AND MECHANICAL ANALYSIS

Components of a skip include the body, door, crosshead, bail, guide rollers, guide shoes, shafts, bearings, wear plates, and dumping mechanism. Generally the body is a rectangular box made of stiffened flat plates. The dump door, closing the bottom of the body, may be a flat reinforced rectangular plate or it may be a gate design. See Figure IV-1 for details and location of skip components.

The crosshead fastens the bail to the rope, generally through a pin. The bail supports the body pivot for swing-out skips. It also takes the door loads, provides a body stop, fastens the guide shoes, and resists the bending moments during dumping. For a fixed-body bottom dump skip the design may or may not include a bail.

Guide rollers and guide shoes are used to prevent wear. Guide rollers are intended to prevent guide wear and guide shoes are used to prevent wear of the structural members.

Shafts, bearings, and dumping mechanism are designed to open the door for skip discharge. The dumping mechanism may be a scroll or an air cylinder to either swing out the body or to operate a gate-type door.

A. DESIGN METHODS

The individual parts of a skip can be selected by several methods. One way is a matter of experience and good practice whereby stresses are taken care of by proper allowances for heavy wear and tear, (Ref. IV-1). Another way is to consider the loads and the structural requirements to support those loads. A design and analysis approach that defines the structural requirements has several advantages. One is the matter of consistency. Member selection for a given set of loads is always the same process; and as loads and dimensions change, the selection is based on the same consistent set of design procedures. If field experience shows an overdesign or a marginal design, the design process can be modified to make adjustments.

Another advantage of designing and analyzing is being able to take full advantage of high strength materials. By establishing strength requirements for the parts of a skip, the use of materials with strengths greater than ordinary structural steel can produce a weight savings. This can be an important consideration when rope pull is limited. As an additional benefit, the high strength, low alloy steels are generally more corrosion resistant.

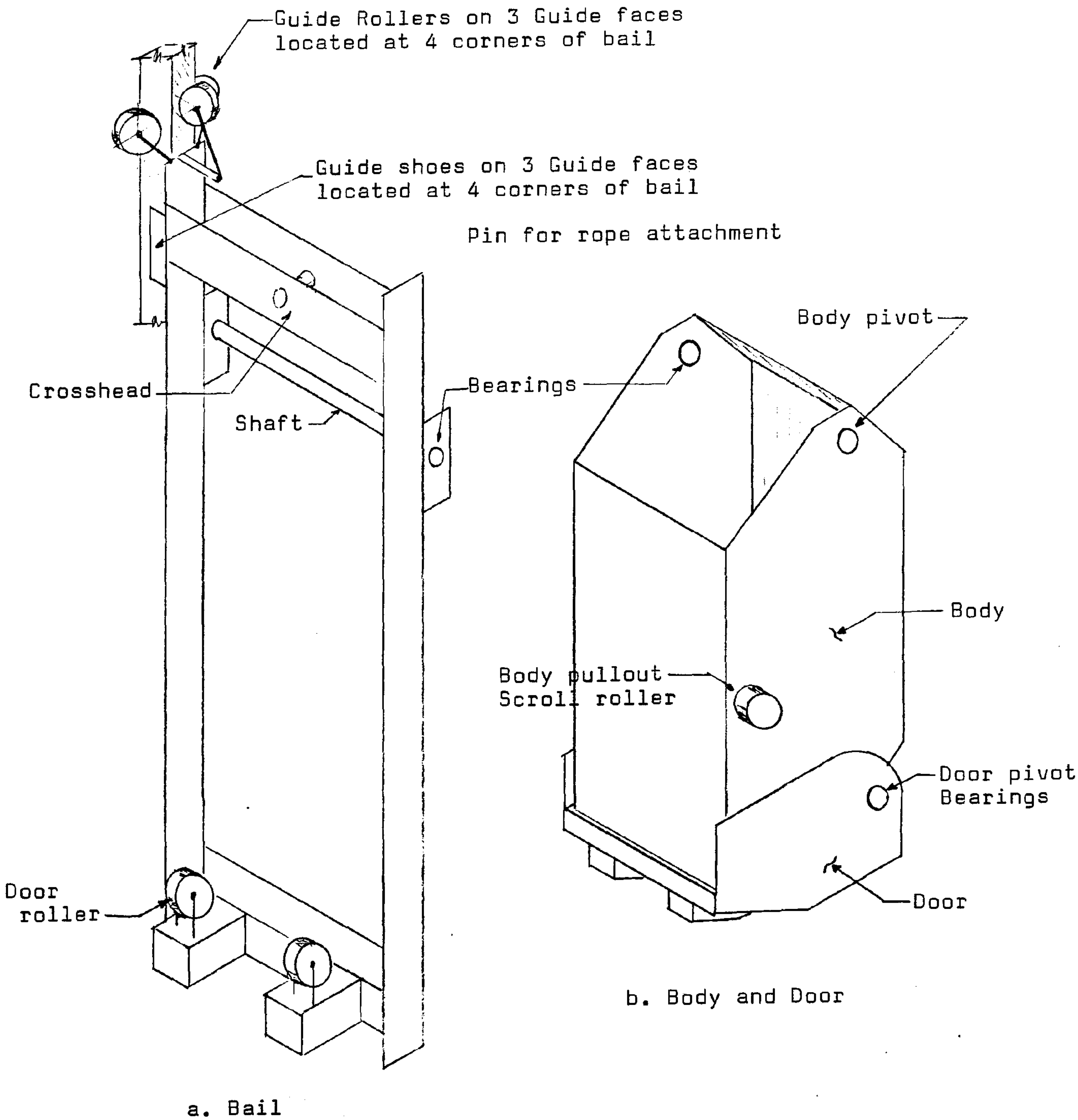


Figure IV-1 Skip Components

Yet another advantage of designing and analyzing to a set of loads is the capacity to produce new systems and concepts. By using engineering methods to size members, framing systems, and joints, a new system can be designed with reasonable assurances of its strength adequacy.

Both static and dynamic loads are imposed on a skip system. The design for static loads is rather well documented. Dynamic loads can be established in two ways. One common procedure in designing for dynamic loads is to select a load factor that converts the dynamic load to an equivalent static load (Ref. IV-2). This may be greater than one or less than one, depending on duration. The classic dynamic load factor of two is for a suddenly applied load not falling from any distance. For loads falling from some distance the following equation gives values of dynamic stress: (Ref. IV-3)

$$\text{Dynamic stress} = \text{Static stress} \left(1+ \sqrt{1 + 2\,\frac{\text{fall height (in.)}}{\text{static defl (in.)}}}\right) \qquad \text{(IV-1)}$$

Dynamic loads with impact factors less than one are not of interest for skip design.

Another method of designing for dynamic loads is to do a dynamic analysis. Except for simple structures that can be modeled as single degree of freedom systems (some beam problems), the use of computers is required for stress analysis. Section V covers the dynamic loads for a general six degree of freedom system representing skip-guide interaction.

B. FACTORS OF SAFETY

For hoisting conveyances Peele (Ref. IV-1) has specified a factor of safety of ten on ultimate strength. This produces a set of allowable stresses on the static loads that are one-tenth of the ultimate strength in tension and shear and on elastic stability for compression.

Another method is to increase the static loads to account for acceleration, impact, and service. Using this method the allowable stresses can be taken from a set of tables such as the AISC manual (Ref. IV-4). One such approach is to use an acceleration factor of 1.2, and impact factor of 1.5, and a service factor of 4.0. This amounts to a 7.2 factor on the static load. By using the ultimate and yield stresses for ASTM A36 steel, it can be shown this set of factors is equivalent to a factor of ten on the ultimate tensile stress. Other acceleration, impact, and service factors can be used for analysis and design based on the member and its service.

By factoring the static loads and using established allowable stresses, the selection of structural members made of the higher strength alloys is a straightforward process. Allowable stresses as a fraction of yield stress for tension, compression, and shear are tabulated (Ref. IV-4) for steels with a yield stress ranging from 36,000 to 100,000 psi.

C. BODY

A skip body is required to contain the muck load during filling, hoisting, and dumping. During filling the body is subjected to sliding and gouging abrasion and to impact loads. Hoisting imposes loads from acceleration and skip-guide dynamic interaction. Dumping loads come from swinging out the body to open the door or from opening the gate in a fixed body skip. Dumping also causes abrasive wear from sliding material.

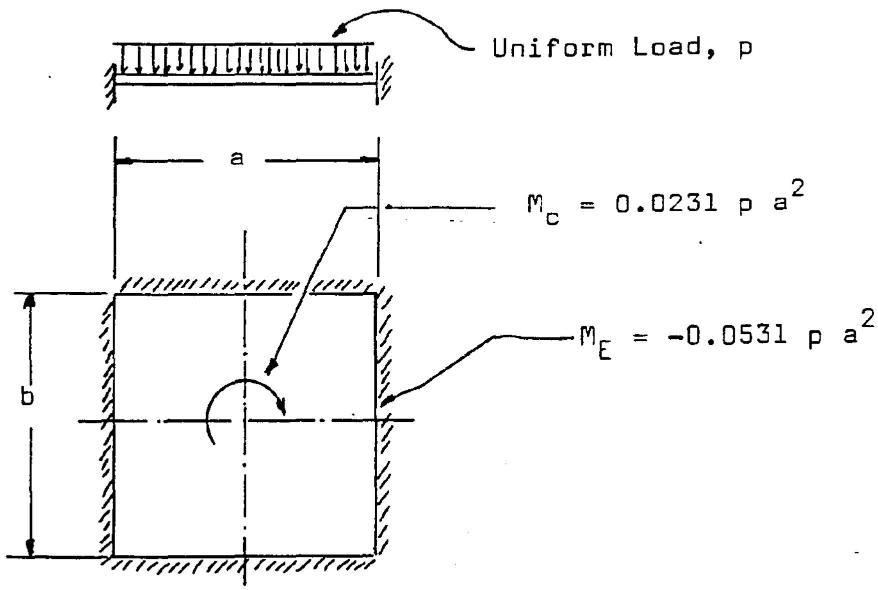
Design of the skip body is the design of a rectangular box with stiffeners. Plate material and thickness, stiffener size and spacing are the design variables. These design variables can be determined from the size of the skip and the amount of muck.

Flat plate design is illustrated in Figure IV-2. For a square plate with all edges fixed the maximum positive and negative moments for a uniform pressure loading are given for the center and the edges in Figure IV-2a. When the plate has an aspect ratio of 2 to 1, the moments are given by Figure IV-2b. These are practically the same as a unit width strip shown in Figure IV-2c. Therefore, for skip bodies the design of the body plates becomes the same as for a prismatic beam design with fixed ends. For skip bodies with continuously welded corners and stiffeners the assumption of end fixity is valid. Coefficients for the edge and center moments are from Ref. IV-5.

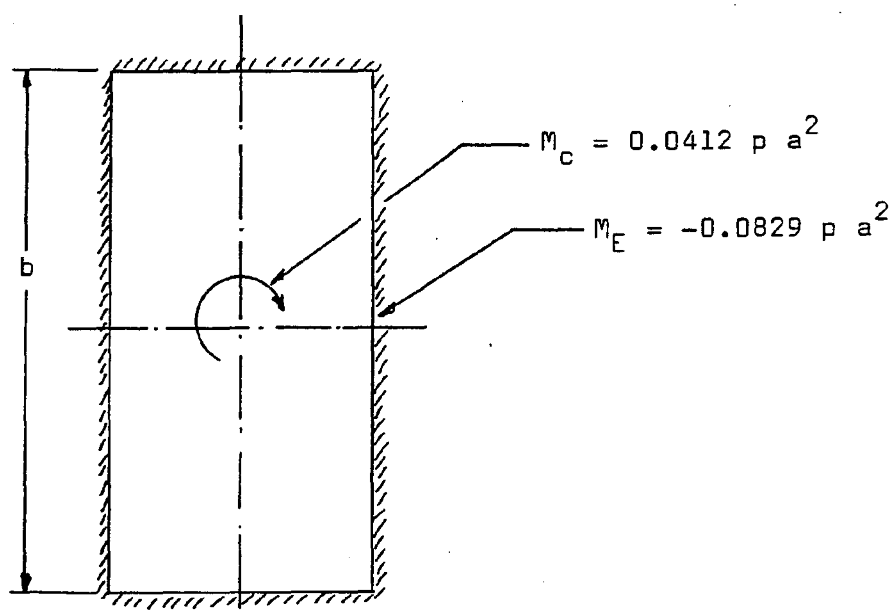
Flat plates require stiffening to provide sufficient bending resistance to the pressure loading caused by the muck. Suitable stiffeners are made of bars, or channels, or angles, or beams, or tees and may be on the inside or on the outside. Many skip bodies are made with channel-type stiffeners welded to the outside of the body.

When stiffeners are added to the plate, the bending section becomes a T-section as shown in Figure IV-3a. The tee consists of the body plate and the stiffener.

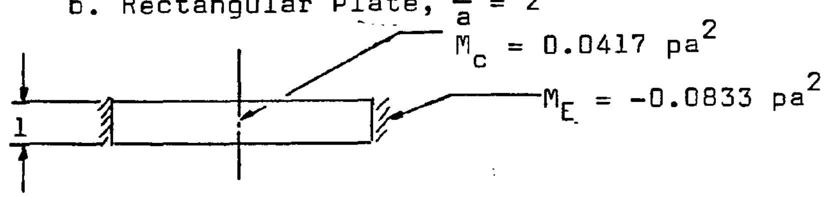
The effective width of plate acting with the stiffener is equal to 0.36 times the span, as given in Reference IV-6. To load the T-section, we can consider the body plate to act as a set of unit width strips spanning between the stiffeners. This is shown in Figure IV-3.



a. Square Plate, $\frac{b}{a} = 1$



b. Rectangular Plate, $\frac{b}{a} = 2$



c. Unit Width Strip

Figure IV-2 Comparison of Flat Plate Moments

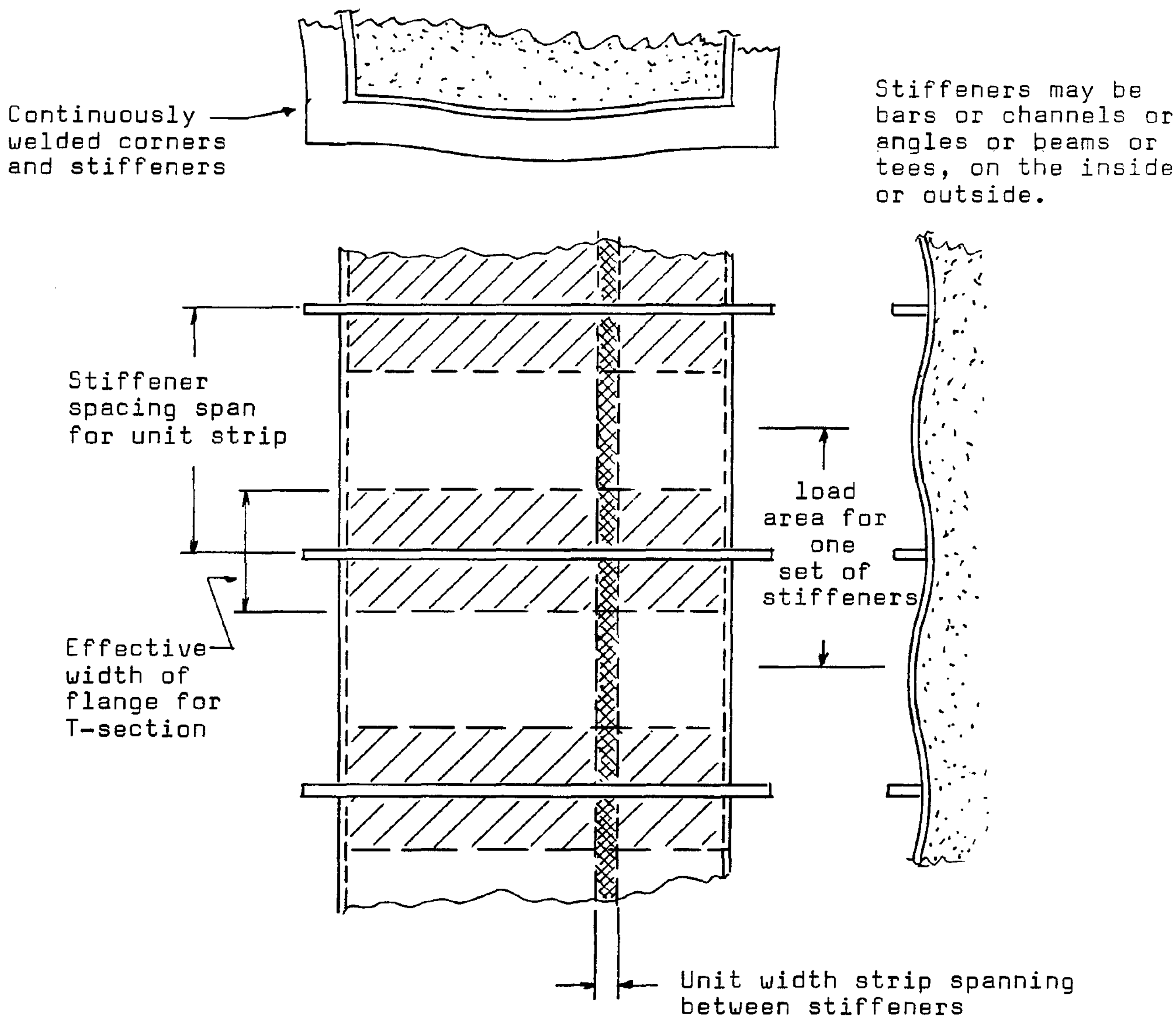


Figure IV-3 Design Elements of Skip Body

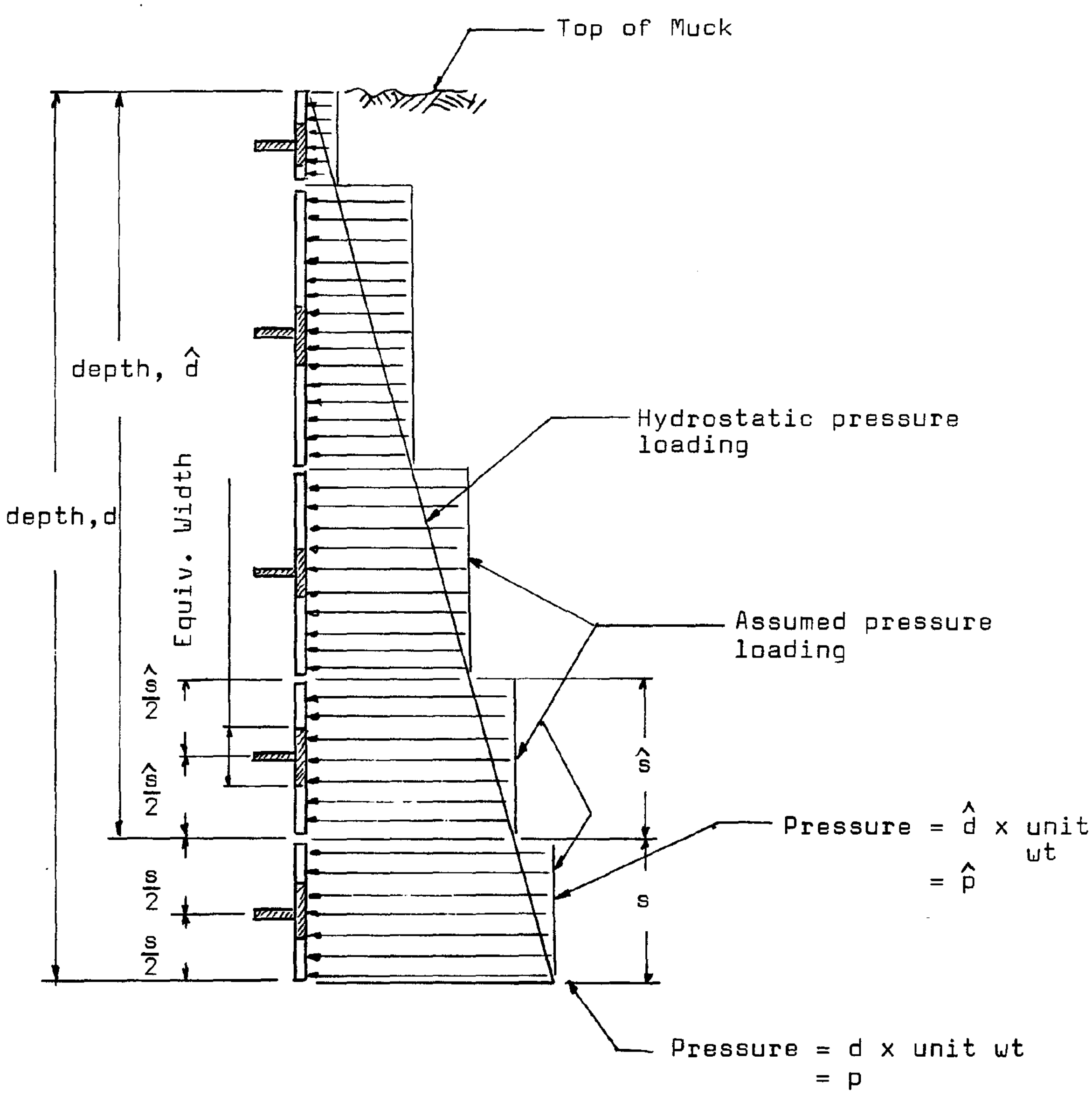


Figure IV-4 Pressure Loading on Skip Body

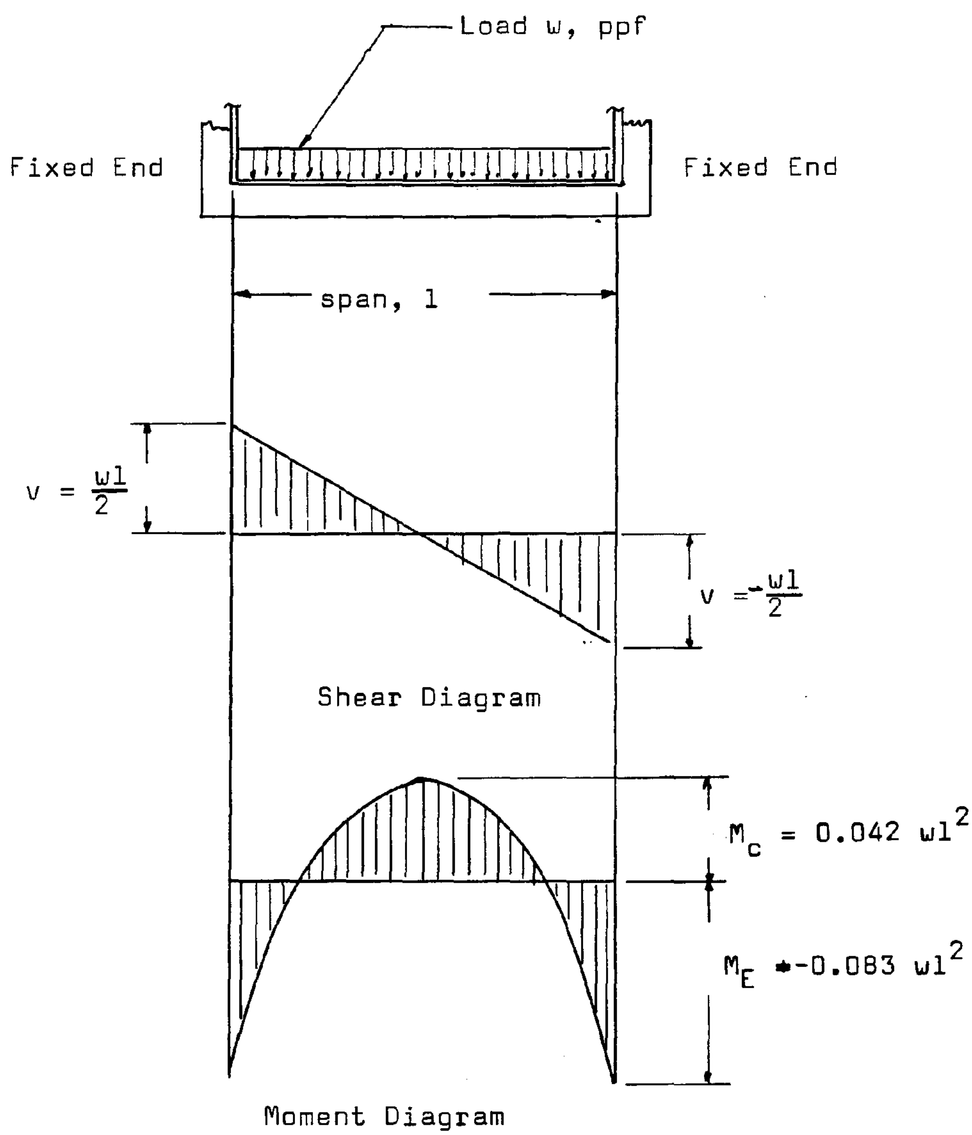


Figure IV-5 Loading, Shear, and Moment Diagrams

Therefore the design of the skip body for muck pressure is the design of the T-sections and the plate between the tees.

Pressure loading from the material in the skip can be considered as a hydrostatic pressure loading. The muck can be represented as a fluid having a unit weight ranging from potash or salt at 80 pounds per cubic foot to iron ore at 170 pounds per cubic foot. A general number of 100 to 120 pounds per cubic foot may be used for copper, silver, uranium, etc., ores. Quite often the material is wet and subjected to dynamic loading; therefore, treating it as a fluid is more correct than considering the material to have an angle of repose and using Rankine's formula.

Loading on the skip body can be found from Figure IV-4. The pressure at any depth in the body can be found from the following:

$$p = d\gamma \qquad \text{(IV-2)}$$

where p = unit press, psf
d = depth, ft
γ = unit weight density, pfc

Loading on a T-section becomes a function of the tee spacing. If we call the space defined by the midpoints between stiffeners the load area with a length s, then the T-section becomes

$$w = ps \qquad \text{(IV-3)}$$

where w = load per unit length, ppf
s = load area, ft

We can then write for the maximum positive and negative moment on the T-section

$$\left.\begin{aligned} M_C &= 0.042\ w\ l^2 \\ M_E &= -0.083\ w\ l^2 \end{aligned}\right\} \qquad \text{(IV-4)}$$

where M_C = Max. pos. mom. at center, ft lb
M_E = Max. neg. mom. at end, ft lb
l = span (length of side), ft

The negative moment M_E is twice as large as the positive moment; therefore, it will govern the design. The moment diagram is shown in Figure IV-5.

Maximum stress in the stiffener-plate combination can be found from the following equation:

$$\sigma = \frac{12\ M_E\ c}{I} \qquad \text{(IV-5)}$$

where σ = Max. bending stress, psi
c = Max. dist. to outer fiber, in
I = Moment of inertia, in^4
(taken about neutral axis)

The dimensions and properties are for a section as shown in Figure IV-6 below:

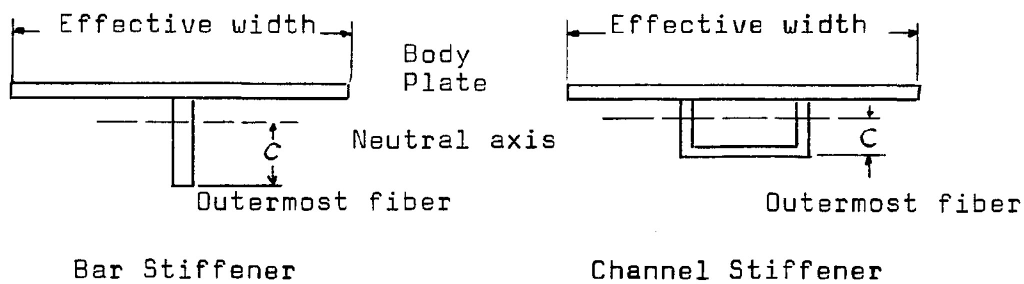


Figure IV-6 Section Through Stiffener on Body

Determining the location of the neutral axis and the moment of inertia about the neutral axis is given in many references.

The amount of weld required to fasten the stiffeners to the body can be found from the following equation:

$$H = \frac{VQ}{I} \qquad \text{(IV-6)}$$

where H = Horizontal shear, ppi (between body plate and stiffener)

Q = Statical moment of plate, in^3 (about neutral axis)

As shown by the shear diagram in Figure IV-5, the highest shear occurs at the ends; therefore, the amount of weld required varies from a minimum at the center to a maximum at each end.

Stiffener spacing is determined by making the total load the same on each stiffener. Referring to Figure IV-4, we can write for the total load, using the assumed pressure distribution,

$$W = p\ s\ l \qquad \text{(IV-7)}$$

where W = total load, lb

To establish the next spacing, the value of $\hat{s}$ is found by keeping W constant and using the next value of pressure as $\hat{p}$, or we can write

$$\hat{s} = \frac{W}{\hat{p}\ l} \qquad \text{(IV-8)}$$

It becomes evident that stiffener spacing is found from the following expression:

$$\text{spacing} = \frac{s}{2} + \frac{\hat{s}}{2} \qquad \text{(IV-9)}$$

A method of design is to establish the bottom stiffener and proceed upward to lower pressure levels and wider spacing. An alternate method is to design for the highest load at the bottom, establish s, and keep the same spacing to the top.

Body plate stresses arise from spanning between stiffeners, as shown in Figure IV-3. Loading on a unit strip can be found from Figure IV-4 and can be assumed to be the higher of the two pressures shown. If the strip is considered as a continuous beam, then the maximum bending moment is given by

$$Mp = \frac{d \gamma s^2}{12} \qquad \text{(IV-10)}$$

where Mp = Max. mom., ft. lb.

The corresponding maximum fiber stress is found from

$$p = \frac{6\ Mp}{t^2} \qquad \text{(IV-11)}$$

where t = plate thickness, in.

This stress is at right angles to the stresses in the T-section, thereby producing a biaxial stress condition. By finding principal stresses, the allowables for combined stress conditions are given in such codes as the AISC Steel Construction Manual (Ref. IV-4).

A problem area in skip body design is the area on the back side that receives the impact and abrasion from loading. An estimate of the loading pressures can be found by referring to Figure IV-7 and using the following expression derived from Reference IV-7:

$$q = 4 \gamma H \cos \theta \qquad \text{(IV-12)}$$

where q = pressure on back side, psf

γ from Eqtn (IV-2)

H = drop height, ft

θ = chute angle from horiz., deg.

A more readily available dimension than the drop height, H, is the distance from the front of the loading pocket to the back of the skip. We can rewrite Equation (IV-12) to read

$$q = 4 \gamma X \sin \theta \qquad \text{(IV-13)}$$

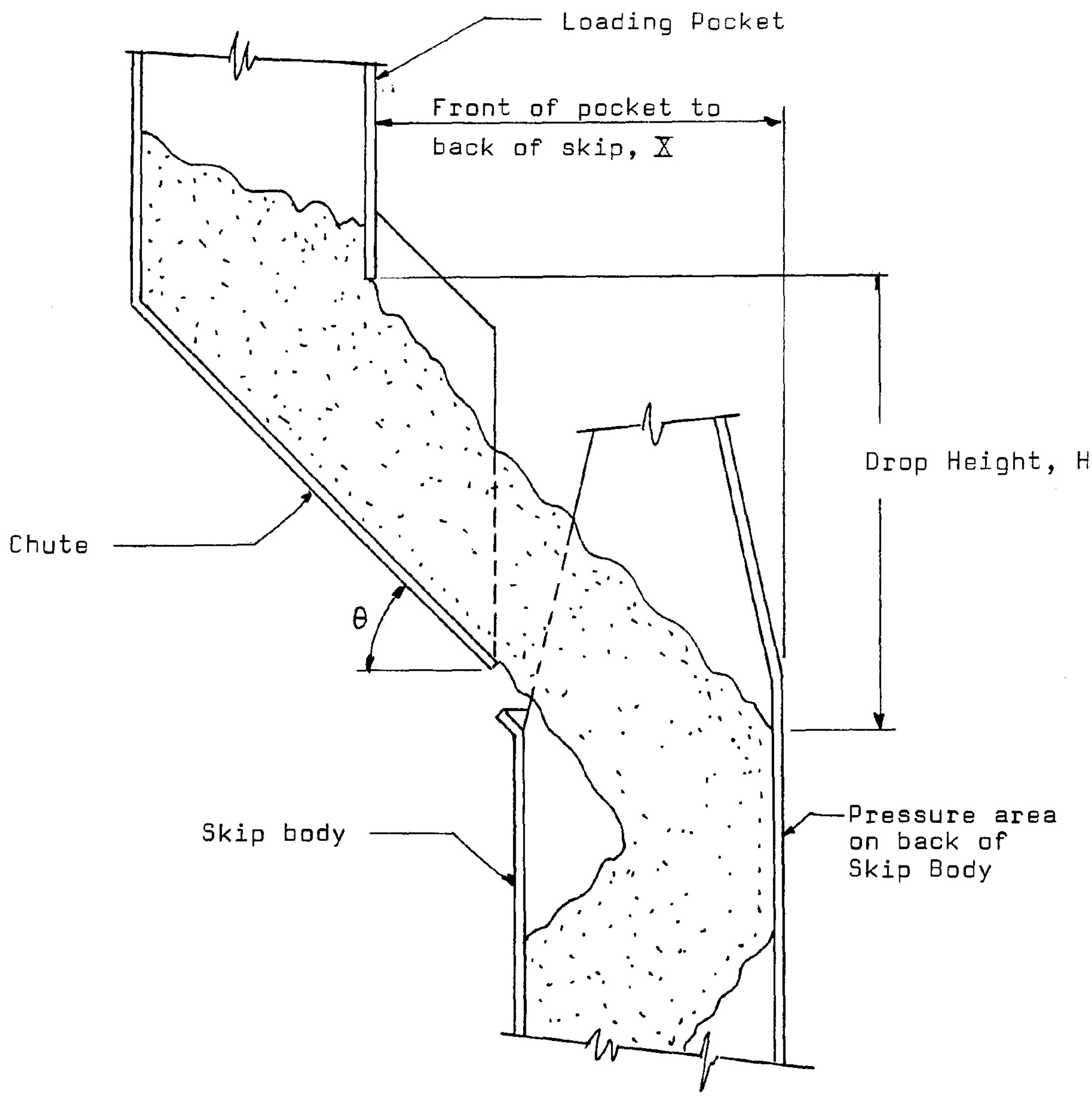


Figure IV-7 Loading Skip from Loading Pocket

As X and θ become larger, the pressure on the back side increases. It is apparent that the pressure on the back plate during loading can be greater than the pressure on the bottom when the skip is full. Equation (IV-2) gives the loading at the bottom when d is equal to the maximum muck depth.

Values of maximum moment are the same for a cubic or a prismatic skip body of equal volume. Figure IV-8 shows the notation for sides, of length, l, and depth, d, with the subscript c for cubic and p for prismatic bodies. The ratio n is the ratio of the lengths of the depth to a side on the prismatic body. We can show that for equal volumes

$$l_c^{\,3} = n l_p^{\,3} \qquad \text{(IV-14)}$$

From which we can establish that

$$l_p = l_c \, n^{-1/3} \qquad \text{(IV-15)}$$

and that

$$d_p = d_c \, n^{2/3} \qquad \text{(IV-16)}$$

Using the maximum negative moment from Equation (IV-4) and substituting for w the expressions in Equations (IV-3) and (IV-2), we can write

$$\left.\begin{aligned} M_{Ec} &= -0.083 \, \gamma \, d_c \, l_c^{\,2} \, s \\ M_{Ep} &= -0.083 \, \gamma \, d_p \, l_p^{\,2} \, s \end{aligned}\right\} \qquad \text{(IV-17)}$$

Substituting for l_p and d_p from Equation (IV-15) and (IV-16) into the second Equation (IV- 17), we have

$$M_{Ep} = -0.083 \, \gamma \, d_c \, n^{2/3} \, (l_c n^{-1/3})^2 s$$

which can be shown to reduce to the first of Equations (IV-17). For a given volume and the same density of muck, the body plates and stiffeners can be the same design.

The hydrostatic pressure, $p = d\gamma$, from Equation (IV-2) is the static pressure. Acceleration and impact will increase this pressure. An acceleration factor of 1.2 is common; however, the impact factor depends on shaft conditions. After the skip is full, it does not appear that impact can exceed 2.0, as shown by Equation (IV-1). An impact factor of 1.5, although arbitrary, seems appropriate. The use of a service factor on the skip body loads becomes a matter of judgment.

It is necessary to consider the appropriate value of plate

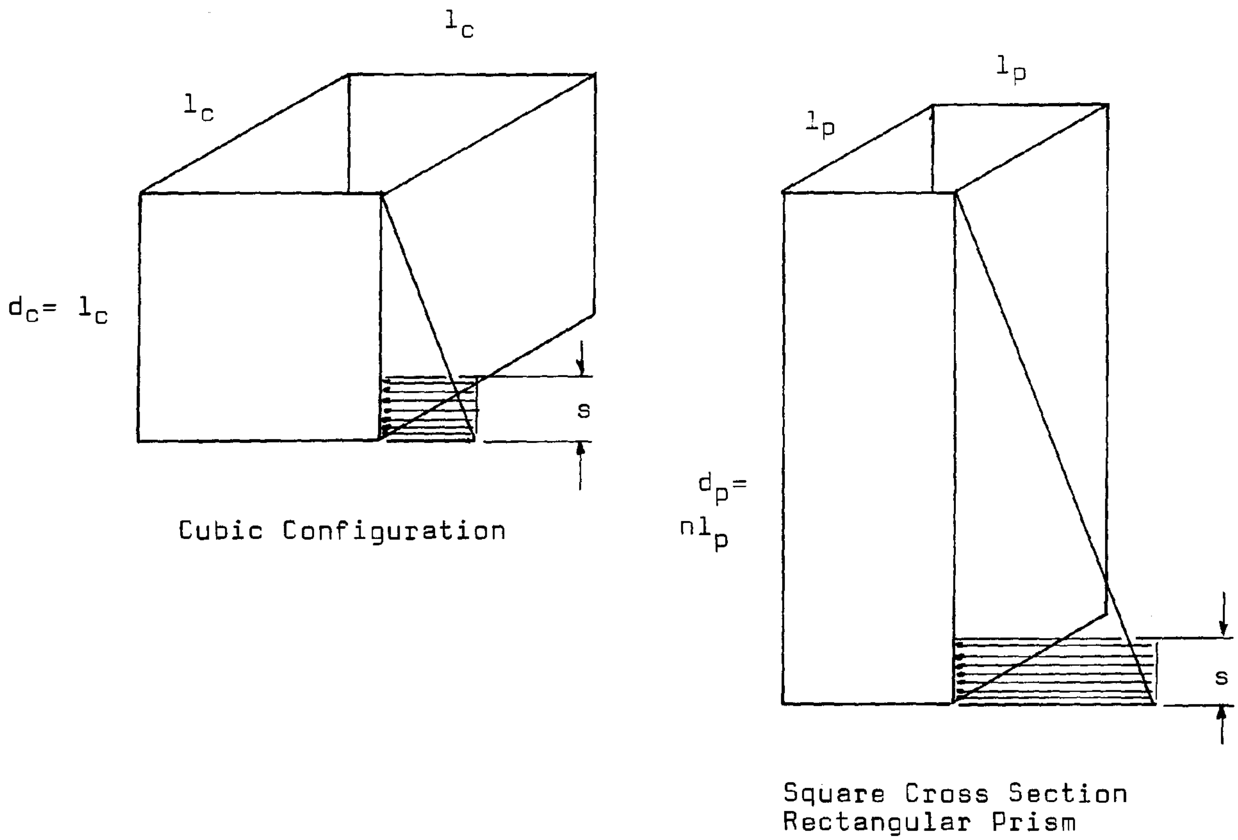


Figure IV-8 Cubic and Prismatic Skip Bodies thickness when calculating the section properties of stiffened bodies. The original thickness is subjected to wear and corrosion. A reasonable reduction in plate thickness can range from 3/4 to 1/2 of the original thickness. This consideration affects the value of I in Equation (IV-5) and the value of t in Equation (IV-11).

One method of skip body design is to use experience and good practice as stated earlier. The other method is to use design and analysis to accommodate the changes in size, ore density for a body of consistent strength.

D. DOOR AND GATE

Skips with swing-out bodies have doors hinged at the back of the skip. Fixed body skips have gates that may be flat or curved plates.

The elements of the dump door for a swing-out bottom dump skip are shown in Figure IV-9. The floor plate is supported by floor beams spanning between the side plates. The side plates are fastened to the body through a pair of pivots at the back. A pair of support beams hold the door closed through a support provided on the bail. Three methods of bottom dump discharge are shown in Figures II-1 through 3, each with its own type of support.

Floor plate design for the muck load follows from Equations (IV-10 and 11) where d is the maximum depth of muck, s is the floor beam spacing, and t is floor plate thickness. Factors for acceleration, impact and service must be applied to the muck density γ . If large rock is to be dumped in the skip, then the dynamic load caused by masses falling on the floor can be estimated from Equation (IV-1). The load area over which this load is to be applied requires some consideration to the framing system and the size of the rock. Both loads should be investigated to determine the governing load criterion.

Floor beams are designed as beams on simple supports, using the section properties of just the beam. Loads are as given before. Maximum fiber stress is limited to the selected allowable after considering acceleration, impact, and service factors. Deflection of the forward floor beam should be considered on a wider body skip, say from five feet and larger. If the deflection is too great and the muck is wet, then excessive leakage may result.

Several problem areas such as leakage paths and closing problems are identified on Figure IV-9. Leakage can occur from the front of the skip due to wear or deflection or misalinement. This can be corrected by adding bars across the opening. The door pivot area at the back of the skip can be a leakage path if not properly designed. Several means are available to correct this. One way is

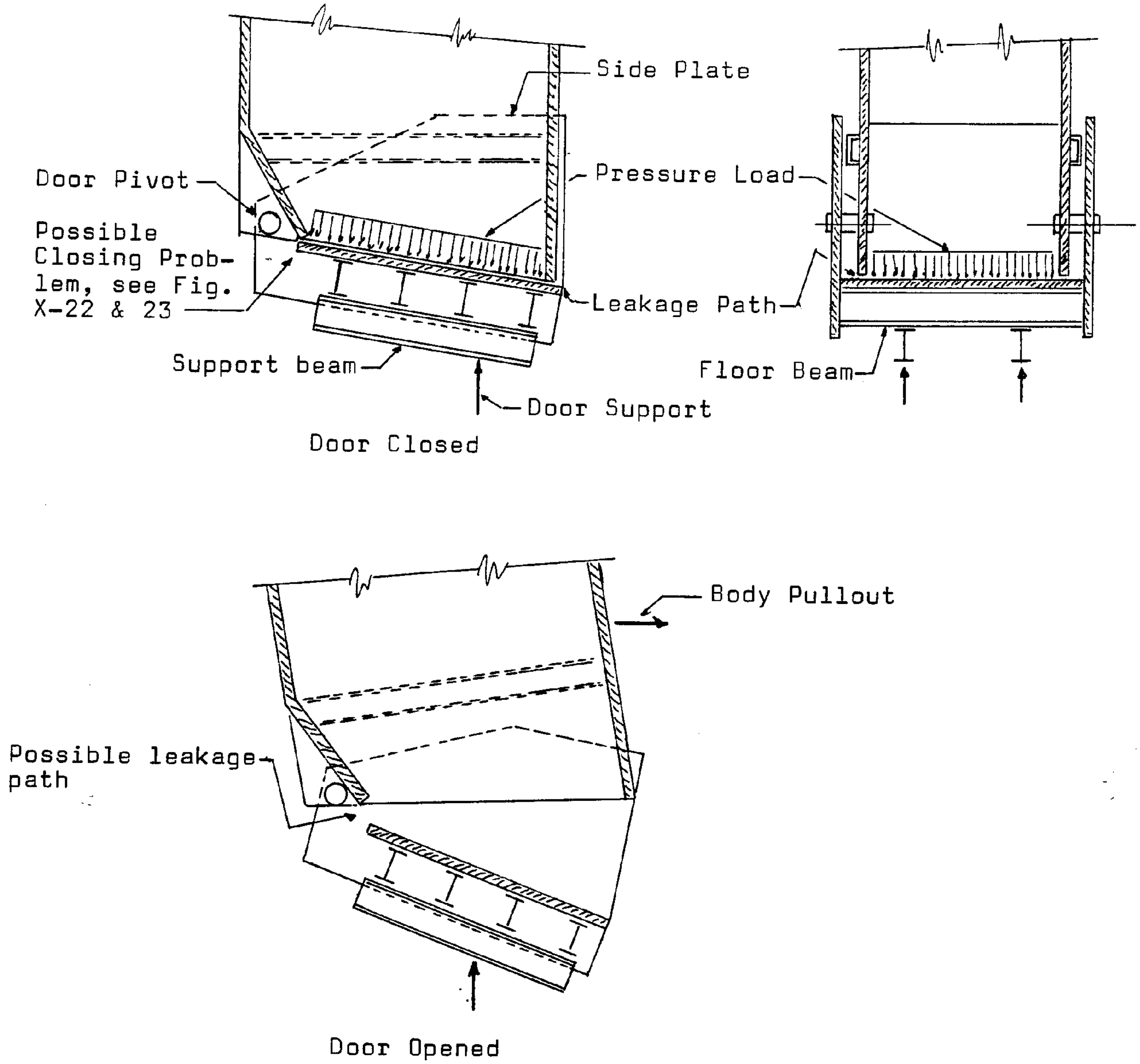


Figure IV-9 Elements of Skip Door for Swing-Out Body Skip to provide a cylindrical closure at the back as shown in Figures II-1 and 2. Another way is to curve the back plate along an arc to act as a closure when the door is open. Other means are rubber sheets or flexible materials that can move and provide a closure at the back side. A leakage path exists between the side plate and the body. Although not shown in Figure IV-9, a skip with external stiffening will have stiffeners at the bottom that must be cleared by the side plates. Wet muck may come over the top of the side plate if the skip is relatively long. Except to lengthen the side plates, there does not seem to be a simple solution.

The back of the door can have a closing problem if the skip does not clean thoroughly. A "nut cracker action" may exist between the floor plate and the back plate of the body. Trying to close with muck left on the floor will cause severe loads on the door pivot. If the door is prevented from closing, the skip will not go back down through the scrolls. A solution is to increase the dump angle to assure cleaning of the skip.

The door when closed may be set at a slope. This has the advantage of requiring less body swing to reach a given dump angle.

Fixed body skips with a gate generally have fewer leakage problems or closure difficulties. The gates may be opened by a scroll engaging a roller or by a self-contained system. Some gate designs require a locking system to keep them closed.

Locking devices may or may not be used on swing-out skips to prevent them from coming open in the shafts. The skip, if loaded from the side opposite the dump side, may come open at the loading station from the muck striking the front side. Equations (IV-12 and 13) give a measure of the pressure on the side being struck by muck. When skips are loaded on the opposite side from the dump side, then a vertical stop is placed for the dump roller to react against, thereby keeping the skip closed.

A means of preventing wear to the floor plate and of easing the shock at loading is shown if Figure IV-10.

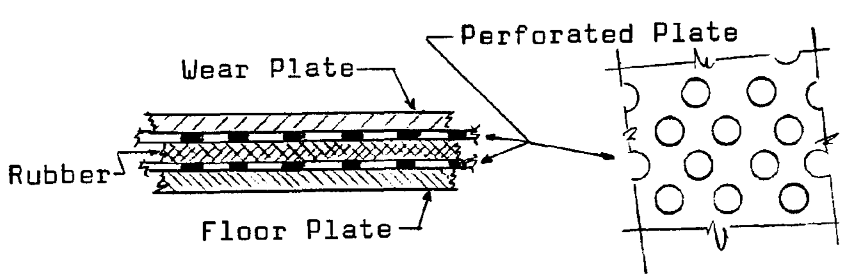


Figure IV-10 Rubber Sandwich Floor Plate on Door

The wear plate can be of any suitable material fastened by bolts to the floor plate. The rubber sandwich material provides a cushioning effect for rocks striking the bottom of the skip. Because rubber is incompressible, perforated plates are needed to give the rubber some place to go when it is squeezed.

E. CROSSHEAD

The crosshead fastens the bail to the rope. It acts as a simple beam with pinned ends, supporting a load at the center. It is required to take all the skip loads which include acceleration, impact, and service factor. To achieve a factor of safety of ten on ultimate (Ref. IV-1 and IV-8), the service factor for ordinary steel must be made equal to four.

Location of the crosshead above the top of the skip should be established after considering the loading system. The muck leaving the loading chute should pass below the crosshead. Attention must be paid to rope stretch when locating the crosshead. In deep mines the rope may stretch a few feet during loading. Rope stretch can be calculated from the following formula:

$$\text{rope stretch (ft)} = \frac{\text{muck load(lb)} \times \text{rope length(ft)}}{\text{rope area (in}^2) \times \text{rope modulus(psi)}} \qquad \text{(IV-18)}$$

In addition, consideration should be given to the consistency and accuracy of spotting the skip when deciding the crosshead location.

A pair of channels placed back to back made a suitable crosshead beam when connecting the skip through a pin. The shear center on a channel is located on the back of channel, the correct side for bending without torsion. Although the load may not be located exactly at the shear center, it will be approximately correct. This is an instance where channels can be loaded without introducing excessive torsional stresses.

F. BAIL

The bail during hoisting supports the body pivot, provides a body stop, and supports the door loads. When dumping, the bail supports the body pivot, reacts the horizontal dump forces, and supports the door. These force conditions are shown in Figure IV-11a for hoisting and Figure IV-11b for dumping. Calculations of these forces follow standard methods of analysis of free body diagrams. During dumping the dynamic loads caused by the body swinging out is covered by Soutar in Reference IV-9.

Both bending and axial loads are imposed on the bail. This produces a combined stresses situation that must be analyzed using an appropriate interaction formula. This can be found in the AISC Manual (Ref. IV-4). Also, the long bail member when subjected to bending must be considered for lateral instability. Again appropriate design criteria can be found in Reference IV-4.

Because of the importance of this structure, acceleration, impact, and service fasteners should provide a factor of safety of ten on ultimate strength.

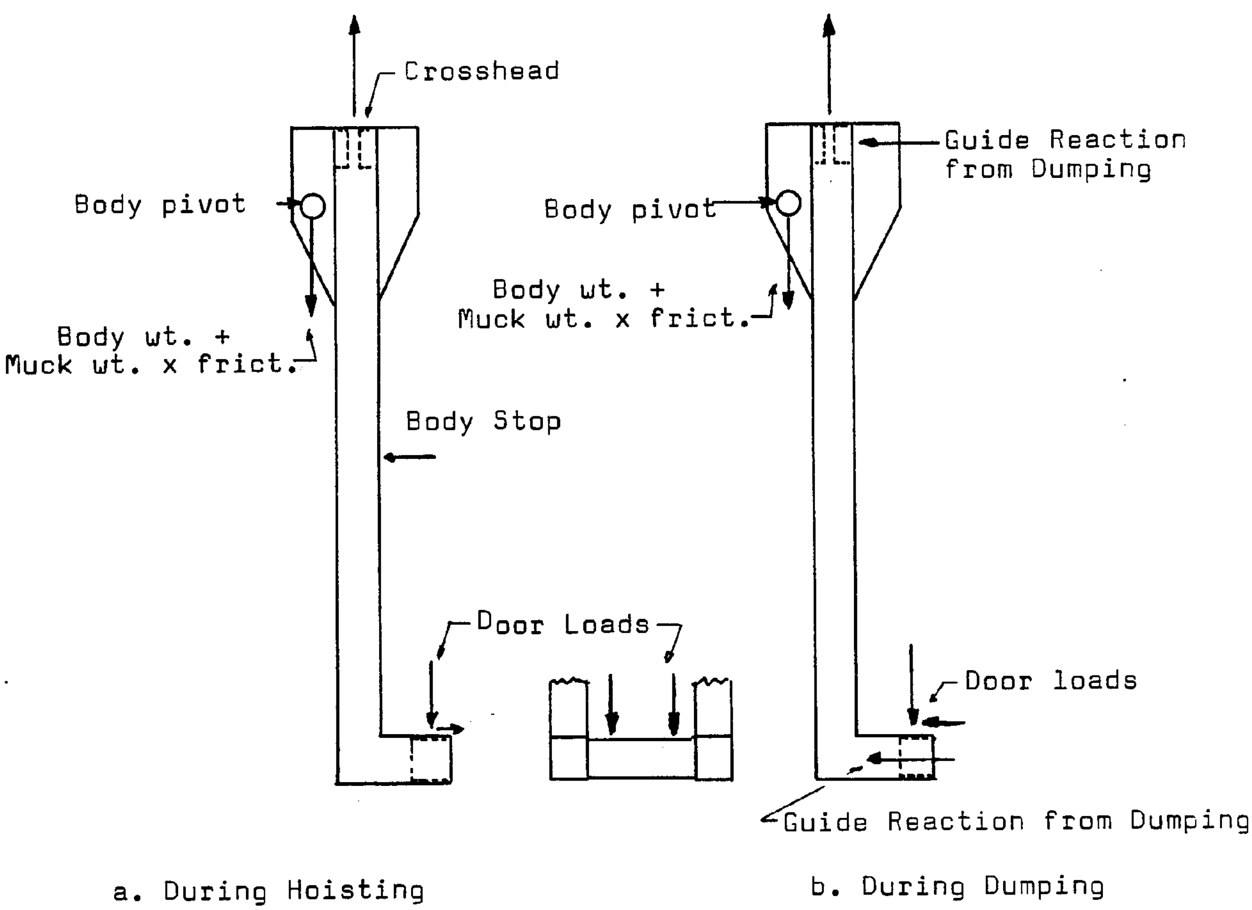


Figure IV-11 Loads on Skip Bail

An extensive treatment of the failure of a skip bail is treated by Heather in Reference IV-10. The muck load was about four tons in an aluminum skip that had been in operation for nine years. Cracks were observed, reinforcements added, and eventually the bail failed. Fatigue was considered to be the principal cause of failure.

By keeping stress levels below the endurance limit, where stresses can be determined, the possibility of such failure becomes more remote. A general approach involving analysis of a structure such as the bail can provide guidance for selection of proper materials and adequately sized structural shapes.

G. GUIDE ROLLERS

A relatively recent development in the design of vertical haulage equipment is the use of guide rollers. Under ideal conditions they provide the restraining forces needed to keep guide shoes from touching the guides. Guide rollers are not always used.

Numbers of papers have been written (References IV-10 to 17) on the dynamic interrelationship of skip mass, hoisting velocity, guide straightness, bunton spacing, guide mechanical properties, guide roller forces, and damping. Section V is the development of the dynamic computer program which includes these variables.

H. GUIDE SHOES

Guide shoes are used on all vertical conveyances. Their main purpose is to keep the structure from being worn away by the guides. Clearances between the guide shoes and the guides can vary from 1/4 inch to 1 1/2 inches by design. Clearances increase as the shoes wear. They are replaceable items to be replaceable when an inspection determines them to be too worn for further use.

Some designs incorporate rubber backing between the guide shoe and the bail. This lessens the chock of loading when the shoe contacts the guide. In general the forward and trailing edges of the shoes are rounded to prevent sharp edges from digging into the guides.

Typical guide shoe materials are mild steel, abrasion-resistant steel, brass, cast iron, high strength low alloy steel. They are fastened with countersunk bolts. If material softer than the bolts is used, the bolt heads will project beyond the guide shoe material and gouge the guides.

V. ANALYTICAL MODEL AND DYNAMIC COMPUTER PROGRAM

A. STATEMENT OF THE PROBLEM

The purpose of this effort is to provide an analysis and computer program for the study of the interacting dynamics of vertical shaft skips and guides.

This study is of interest primarily because of existing wear and maintenance problems in many present day mines. The wear and tear on both skips and guides is of major concern and analytical effort to understand and minimize or reduce the contact forces causing this wear is certainly justified.

In deriving and programming the analysis, an attempt has been made to keep the result realistic as well as versatile. It is desired to apply the analysis with little modification to wood, steel and rope guides as well as various skip geometries both empty and in various degrees of fullness and moving either up or down.

Guide flexibility and crookedness in the form of curved beams or bunton misalinement have been provided.

The computer program constitutes a design analysis tool not previously available and as such represents a new technological development. It provides a convenient method for the study of many practical problems of skip and shaft design for which there was no prior method.

Although the initial purpose of the analysis is parametric studies of conceptual systems for design guidance, the computer program will eventually be used for correlation of dynamic experimental data taken from actual shafts.

The possible applications of this analysis are numerous. Some specific areas of interest for further development are listed below.

B. BACKGROUND

Under this study a survey of the literature on skip dynamics and skip and guide interaction has been started. The literature studied so far indicates that some experimental work has been done and the basic dynamic equations of motion have been considered. Most of this work is foreign; in fact, the mathmatical studies have been, for the most part, Eastern European and Russian. This analytical work was limited in its scope and complexity because Russia has not had computers that can handle numerical integration of complex systems. The Russian derivations are of interest but, due to the emphasis on simplification to equation types with closed form solutions, their numerical results are, for the most part, completely useless. A list of related literature is given in Section XIV as References V-1 through 11.

C. MECHANICAL MODEL

In the present study, which is primarily aimed at parametric studies, a retangular skip has been considered (Figure V-1). The skip has spring mounted rollers which run on flexible guide beams. There are rollers in two directions at four points on the skip. The guide beams are modeled as simply supported beams without inertia which can bend in two places. Initial crookedness of the guides has been provided; and in cases where two rollers occupy the same simply-supported span, the influence of each roller on the beam is superimposed. When the roller spring is bottomed or when a roller passes over a bunton support, the flexibility becomes appropriatel small. When the displacement of the guide rollers is large enough to cause contact with the guide shoes, the program indicates this with a warning message.

The basic formulation consists of the linear dynamic equations of motion of a rigid body with five (5) degrees of freedom. The motion is referred to the center of mass (C.M.) which does not coincide with the geometric center (G.C.), so that the five basic degrees of freedom are two horizontal translations of the center of mass (x and z) and three angles of rotation about the center of mass (θ, ϕ, ψ).

So far, the motion has been considered to be upward and at constant velocity. The direction of motion will definitely make a difference, but the analysis of downward motion will present no special difficulty. Rope torsion has been included.

The guide contact forces can be arbitrarily defined. So far, only contact forces normal to the roller/guide contact surface have been considered. Friction forces and variable winding speeds are contemplated. Impact forces of contact present analytical difficulties, but their formulation is in progress.

The equation of kinematic constraint have been written for small deflections and are presently linear. Nonlinearities, however, present no special problems for the method of analysis we have chosen.

A rudimentary form of damping has been formulated. This area is of vital concern when considering stability and should be given prime emphasis if this method is further developed.

D. MATHEMATICAL FORMULATION

1. Introduction

The mathematical formulation consists of five (5) second order differential equations which define the motion of a damped spring-mass system consisting of a single rigid body

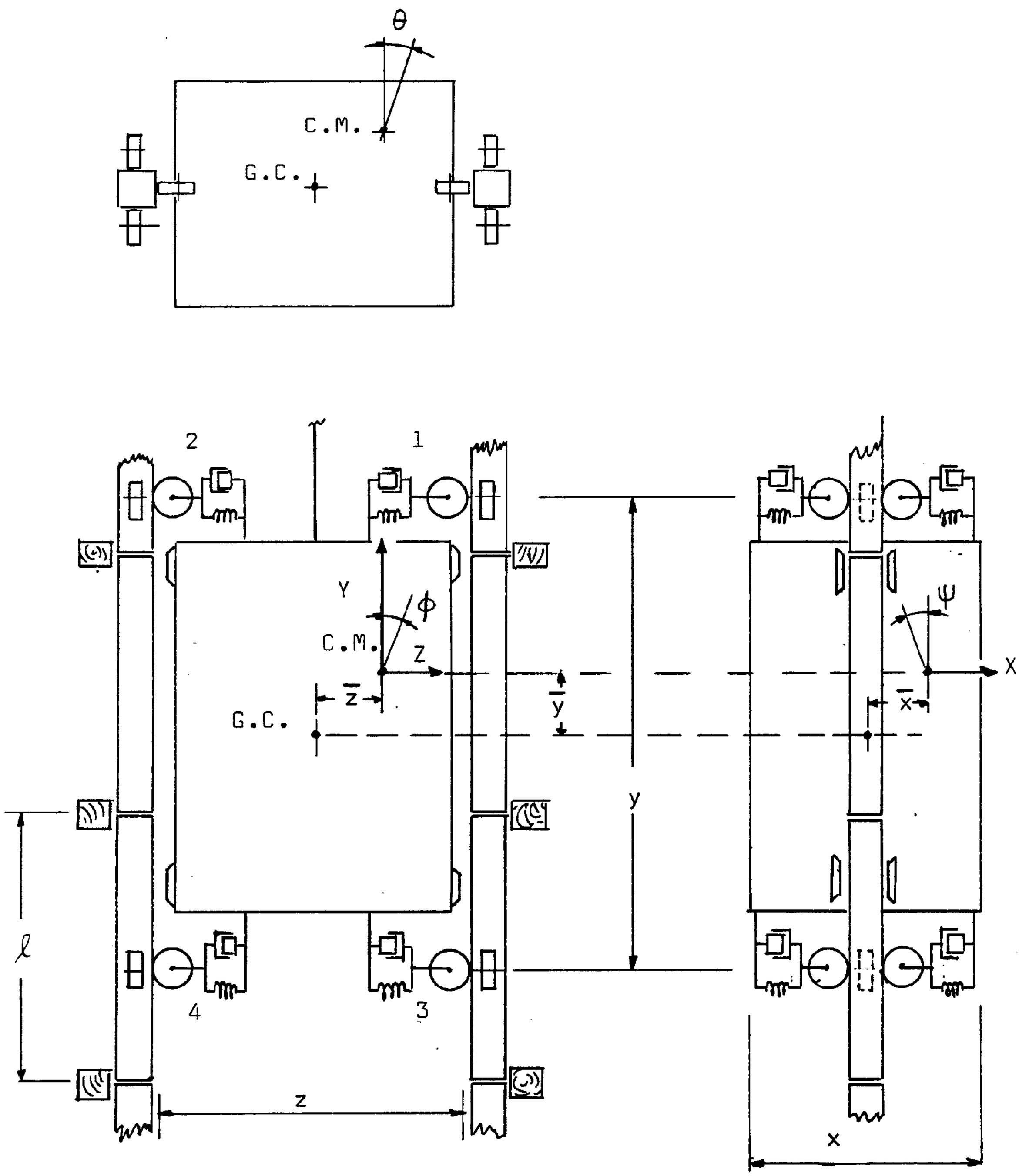


Figure V-1 Skip and Guide Model

with five (5) degrees of freedom and various time-dependent applied forces. The formulation is written with respect to the mass center. The vertical motion of the center of mass is considered to be at constant velocity. The velocity determines the speed at which the guide beams are traversed, the flexible guide beams, therefore, providing time dependent input forces to the system.

The equations of motion are written as ten(10) first order equations and are solved by means of Runge-Kutta initial value forward integration. The initial values are derived as a static equilibrium position compatible with the roller and guide flexibilities and the position of the center of mass. The vertical inertia of the system in starting is ignored.

2. Nomenclature

Numerical subscripts (1, ..., 4) . . . Roller positions

EI_x , EI_z . . . guide beam rigidities

F_{x_i} , F_{z_i} . . . x, z directed force at roller i

g . . . gravitational constant

h_x , h_y , h_z . . . skip linear dimensions

$\overline{I}_x$, $\overline{I}_y$, $\overline{I}_z$ . . . mass moments of inertia of the skip

k_x , k_z . . . guide spring constants

ℓ . . . bunton spacing

T . . . rope torque

W . . . skip weight

x, z . . . horizontal translations of mass center

$\overline{x}$, $\overline{y}$, $\overline{z}$ . . . coordinates of the mass center relative to the geometric center

x_i , z_i . . . horizontal translations of contact point i

x_{b_i} , z_{b_i} . . . guide beam misalinement at roller i

Y_{ij} . . . guide beam flexibilities

ΔSi . . . guide spring length before motion

δSi . . . guide spring length after motion

δBi . . . extension of spring due to motion

θ, ϕ, ψ . . . angles of rotation

ξi . . . position of roller i along guide beam, measured from bunton

$(\dot{\ })$ . . . time derivative of ( )

3. Basic Equations of Motion

The basic equations of motion are as follows:

$$W/g\ \ddot{x} = F_1 \quad ; \quad W/g\ \ddot{z} = F_2 \qquad (V\text{-}1, V\text{-}2)$$

$$\overline{I}_y \ddot{\theta} = T + W/g\ (\overline{x}\ \ddot{z} - \overline{z}\ \ddot{x}) + h_z\ F_3/2 \qquad (V\text{-}3)$$

$$\overline{I}_z \ddot{\psi} = W\ (\overline{y}\ \ddot{x}/g - \overline{x}) + h_y\ F_4/2 \qquad (V\text{-}4)$$

$$\overline{I}_x \ddot{\phi} = W\ (\overline{z} - \overline{y}\ \ddot{z}/g) + h_y\ F_5/2 \qquad (V\text{-}5)$$

where

$$F_1 = F_{x_1} + F_{x_2} + F_{x_3} + F_{x_4} \qquad (V\ 6)$$

$$F_2 = F_{z_1} + F_{z_2} + F_{z_3} + F_{z_4} \qquad (V\text{-}7)$$

$$F_3 = F_{x_1} + F_{x_3} - F_{x_2} - F_{x_4} \qquad (V\text{-}8)$$

$$F_4 = F_{x_3} + F_{x_4} - F_{x_1} - F_{x_2} \qquad (V\text{-}9)$$

$$F_5 = F_{z_1} + F_{z_2} - F_{z_3} - F_{z_4} \qquad (V\text{-}10)$$

4. Guide Forces

The following derivation of guide forces considers a case in which roller 1, at the top of the skip, and roller 3, at the bottom on the same side, are spring loaded against a single flexible misalined guide beam. The derivation references Figure V-2 which depicts the kinematics at roller 1.

From the figure, considering z - motion, we have

$$\Delta S_1 + z_{B_1} + \delta B_1 = z_1 + \delta S_1 \qquad (V\text{-}11)$$

from which the change in spring length from an alined equilibrium position is

$$\delta S_1 - \Delta S_1 = z_{B_1} + \delta B_1 - z_1 \qquad (V\text{-}12a)$$

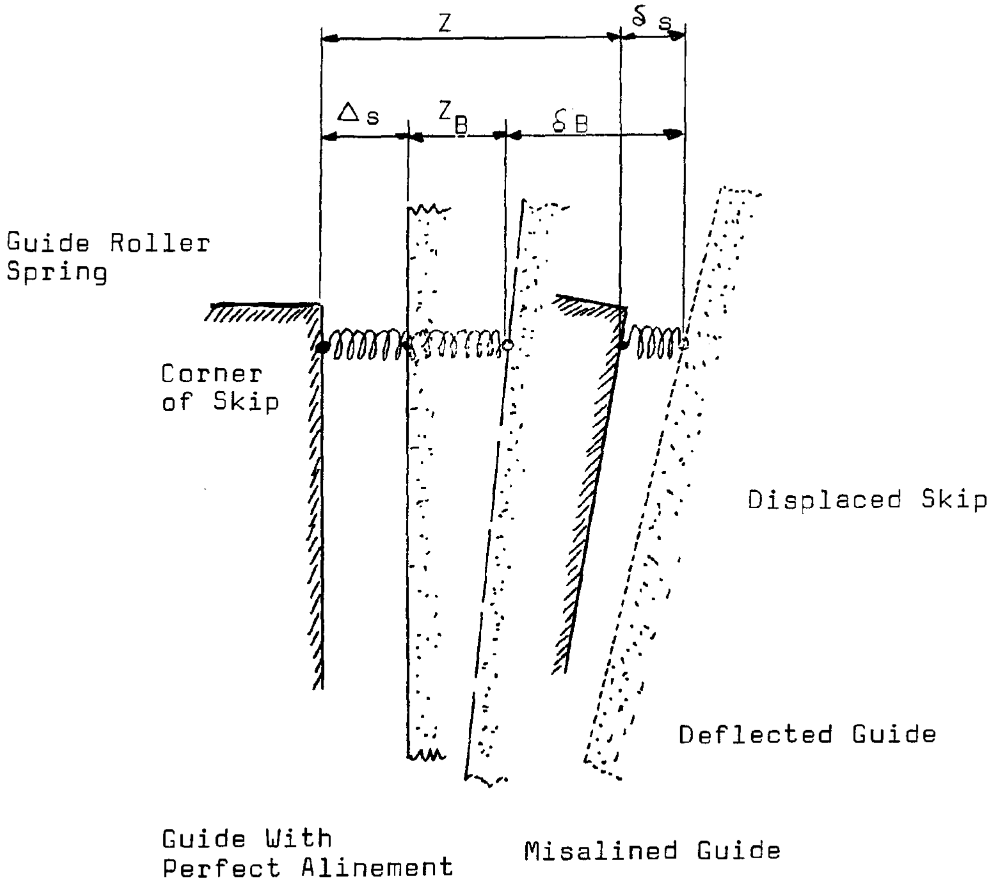


Figure V-2 Kinematics of a Guide Point

and similarly

$$\delta S_3 - \Delta S_3 = z_{B_3} + \delta B_3 - z_3 \qquad \text{(V-12b)}$$

Relating this to the force on the roller, we have

$$F_{z_1}/k_z = z_{b_1} + \delta B_1 - z_1 \qquad \text{(V-13a)}$$

$$F_{z_3}/k_z = z_{B_3} + \delta B_3 - z_3 \qquad \text{(V-13b)}$$

Solving these for δB_1 , δB_3 gives

$$\delta B_1 = F_{z_1}/k_z + (z_1 - z_{B_1}) \qquad \text{(V-14a)}$$

$$\delta B_3 = F_{z_3}/k_z + (z_3 - z_{B_3}) \qquad \text{(V-14b)}$$

Considering the direction of positive displacements relative to beam deflection, we have

$$\delta B_1 = -Y_{11} F_{z_1} - Y_{13} F_{z_3} \qquad \text{(V-15a)}$$

$$\delta B_3 = -Y_{31} F_{z_1} - Y_{33} F_{z_3} \qquad \text{(V-15b)}$$

Equating these expressions gives

$$F_{z_1}/k_z + (z_1 - z_{B_1}) = -Y_{11}F_{z_1} - Y_{13}F_{z_3} \qquad \text{(V-16a)}$$

$$F_{z_3}/k_z + (z_3 - z_{B_3}) = -Y_{31}F_{z_1} - Y_{33}F_{z_3} \qquad \text{(V-16b)}$$

or

$$\left(Y_{11} + \frac{1}{k_z}\right) F_{z_1} + Y_{13}F_{z_3} = z_{B_1} - z_1 \qquad \text{(V-17a)}$$

$$Y_{31} F_{z_1} + \left(Y_{33} + \frac{1}{k_z}\right) F_{z_3} = z_{B_3} - z_3 \qquad \text{(V-17b)}$$

and letting

$$Z_{11} = Y_{11} + 1/k_z \qquad \text{(V-18)}$$

$$Z_{13} = Y_{13} = Y_{31} \qquad \text{(V-19)}$$

$$Z_{33} = Y_{33} + 1/k_z \qquad \text{(V-20)}$$

we solve for F_{z_1} , f_{z_3} , obtaining

$$F_{z_1} = \frac{(z_{B_1} - z_1) Z_{33} - (z_{B_3} - z_3) Z_{13}}{Z_{11} Z_{33} - Z_{13}^2} \qquad \text{(V-21a)}$$

$$F_{z_3} = \frac{(z_{B_3} - z_3) Z_{11} - (z_{B_1} - z_1) Z_{13}}{Z_{11} Z_{33} - Z_{13}^2} \qquad \text{(V-21b)}$$

to which we can add damping forces proportional to the velocity of the contact point of the form

$$- c_z (\dot{z}_1 - \dot{z}_{B_1}) \qquad \text{(V-22a)}$$

$$- c_z (\dot{z}_3 - \dot{z}_{B_3}) \qquad \text{(V-22b)}$$

The beam flexibilities used above are (Figure V-3)

$$Y_{11} = \frac{\xi_1^2 (\ell - \xi_1)^2}{3EI_x \ell} \; ; \; Y_{33} = \frac{\xi_3^2 (\ell - \xi_3)^2}{3EI_x \ell} \qquad \text{(V-23,24)}$$

$$Y_{13} = \frac{(\ell - \xi_1) \quad \xi_3 \left[(\ell - \xi_1)^2 - \ell^2 + \xi_3^2 \right]}{6 EI_x \ell} \qquad \text{(V-25)}$$

Similar equations can be written for F_{z_2} and F_{z_4} and then for

F_{x_1} , F_{x_2} , F_{x_3} and F_{x_4} .

5. Kinematics

The motion of the points of guide contact are related to the motion of the center of mass by the following linear equations:

$$x_1 = x + a_1\theta - b_1\Psi \qquad \text{(V-26)}$$

$$x_2 = x - a_2\theta - b_1\Psi \qquad \text{(V-27)}$$

$$x_3 = x + a_1\theta + b_2\Psi \qquad \text{(V-28)}$$

$$x_4 = x - a_2\theta + b_2\Psi \qquad \text{(V-29)}$$

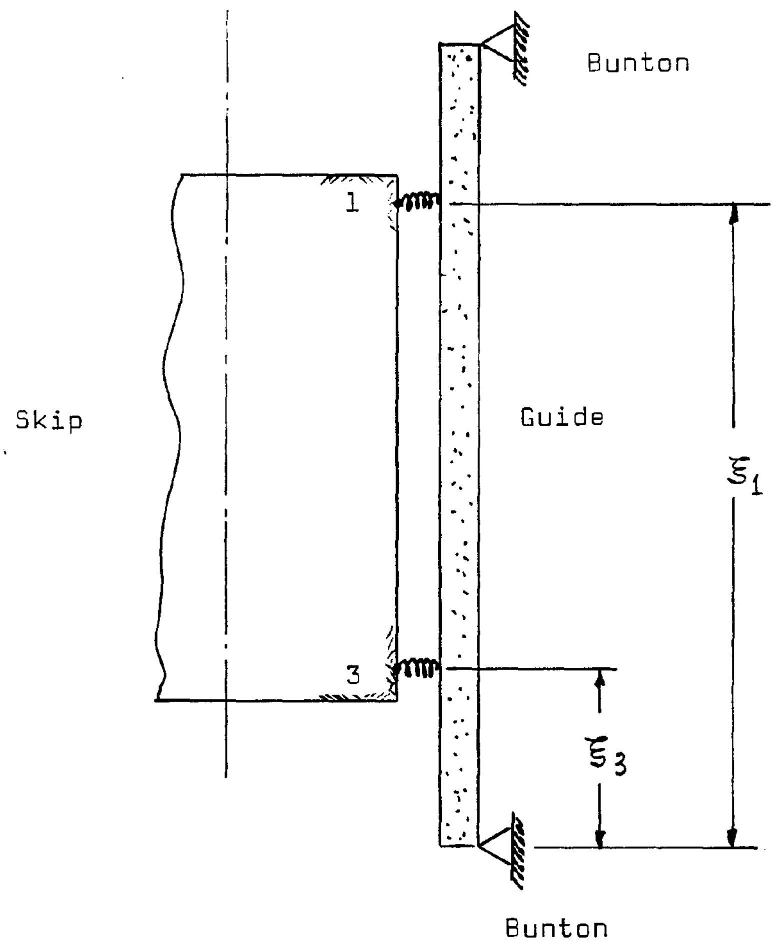


Figure V-3 Guide Beam

$$z_1 = z + b_1 \phi \qquad (V-30)$$

$$z_2 = z + b_1 \phi \qquad (V-31)$$

$$z_3 = z - b_2 \phi \qquad (V-32)$$

$$z_4 = z - b_2 \phi \qquad (V-33)$$

where

$$a_1 = \frac{h_z}{2} - \bar{z} \qquad (V-34a)$$

$$a_2 = \frac{h_z}{2} + \bar{z} \qquad (V-34b)$$

$$b_1 = \frac{h_y}{2} - \bar{y} \qquad (V-35a)$$

$$b_2 = \frac{h_y}{2} + \bar{y} \qquad (V-35b)$$

6. Initial Conditions

In order to establish a kinematically compatible position from which to begin the numerical integration, selection has been made of the static equilibrium position with the mass center and other parameters given.

We let

$$K_{z_{11}} = \frac{Z_{11}}{Z_{11} Z_{33} - Z_{13}^2} \qquad (V-36)$$

$$K_{z_{22}} = \frac{Z_{22}}{Z_{22} Z_{44} - Z_{24}^2} \qquad (V-37)$$

$$K_{z_{33}} = \frac{Z_{33}}{Z_{11} Z_{33} - Z_{13}^2} \qquad (V-38)$$

$$K_{z_{44}} = \frac{Z_{44}}{Z_{22} Z_{44} - Z_{24}^2} \qquad (V-39)$$

$$K_{z_{13}} = \frac{Z_{13}}{Z_{11} Z_{33} - Z_{13}^2} \qquad (V-40)$$

$$K_{z_{24}} = \frac{Z_{24}}{Z_{22} Z_{44} - Z_{24}^{2}} \qquad \text{(V-41)}$$

Substituting these expressions and Equations (V-26) to (V-33) into the force equations such as (V-21), substituting the resulting expressions into Equations (V-6) to (V-10), gathering terms and finally substituting into the equilibrium equations (V-1) to (V-5), we obtain

$$-C_{11} x_0 - C_{12} \theta_0 - C_{13} \Psi_0 + q_1 = 0 \qquad \text{(V-42)}$$

$$-C_{31} x_0 - C_{32} \theta_0 - C_{33} \Psi_0 + q_3 = \frac{-2T}{h_z} \qquad \text{(V-43)}$$

$$-C_{41} x_0 - C_{42} \theta_0 - C_{43} \Psi_0 + q_4 = \frac{2W\bar{x}}{h_y} \qquad \text{(V-44)}$$

$$-C_{21} z_0 - C_{22} \phi_0 + q_2 = 0 \qquad \text{(V-45)}$$

$$-C_{51} z_0 - C_{52} \Phi_0 + q_5 = \frac{-2W\bar{z}}{h_y} \qquad \text{(V-46)}$$

Where x_0 , z_0 , θ_0 , Φ_0 and Ψ_0 are the initial values of the coordinates of the mass center.

The equations for the C's and q's are

$$C_{11} = K_{x_{11}} + K_{x_{22}} + K_{x_{33}} + K_{x_{44}} - 2 K_{x_{13}} - 2 K_{x_{24}}$$

$$C_{12} = a_1 (K_{x_{33}} + K_{x_{11}} - 2 K_{x_{13}}) + a_2 (2 K_{x_{24}} - K_{x_{44}} - K_{x_{22}})$$

$$C_{13} = b_1 (K_{x_{13}} + K_{x_{24}} - K_{x_{33}} - K_{x_{44}}) + b_2 (K_{x_{11}} + K_{x_{22}} - K_{x_{24}} - K_{x_{13}})$$

$$C_{21} = K_{z_{11}} + K_{z_{22}} + K_{z_{33}} + K_{z_{44}} - 2K_{z_{13}} - 2K_{z_{24}}$$

$$C_{22} = b_1 (K_{z_{33}} + K_{z_{44}} - K_{z_{13}} - K_{z_{24}}) - b_2 (K_{z_{11}} + K_{z_{22}} - K_{z_{13}} - K_{z_{24}})$$

$$C_{31} = K_{x_{33}} + K_{x_{11}} - K_{x_{44}} - K_{x_{22}} - 2K_{x_{13}} + 2K_{x_{24}}$$

$$C_{32} = a_1 \left(K_{x_{33}} + K_{x_{11}} - 2K_{x_{13}} \right)$$
$$+ a_2 \left(K_{x_{22}} + K_{x_{44}} - 2K_{x_{24}} \right)$$

$$C_{33} = b_2 \left(K_{x_{11}} + K_{x_{22}} - K_{x_{13}} + K_{x_{24}} \right)$$
$$- b_1 \left(K_{x_{33}} - K_{x_{44}} - K_{x_{13}} + K_{x_{24}} \right)$$

$$C_{41} = K_{x_{11}} + K_{x_{22}} - K_{x_{33}} - K_{x_{44}}$$

$$C_{42} = a_1 \left(K_{x_{11}} - K_{x_{33}} \right) - a_2 \left(K_{x_{22}} - K_{x_{44}} \right)$$

$$C_{43} = b_1 \left(K_{x_{33}} + K_{x_{44}} + K_{x_{13}} + K_{x_{24}} \right)$$
$$+ b_2 \left(K_{x_{11}} + K_{x_{22}} + K_{x_{13}} + K_{x_{24}} \right)$$

$$C_{51} = K_{z_{33}} + K_{z_{44}} - K_{z_{11}} - K_{z_{22}}$$

$$C_{52} = b_1 \left(K_{z_{33}} + K_{z_{44}} + K_{z_{13}} + K_{z_{24}} \right)$$
$$+ b_2 \left(K_{z_{11}} + K_{z_{22}} + K_{z_{13}} + K_{z_{24}} \right)$$

$$q_1 = x_{b_1} \left(K_{x_{33}} - K_{x_{13}} \right) + x_{b_2} \left(K_{x_{44}} - K_{x_{24}} \right)$$
$$+ x_{b_3} \left(K_{x_{11}} - K_{x_{13}} \right) + x_{b_4} \left(K_{x_{22}} - K_{x_{24}} \right)$$

$$q_2 = z_{b_3} \left(K_{z_{11}} - K_{z_{13}} \right) + z_{b_1} \left(K_{z_{33}} - K_{z_{13}} \right)$$
$$+ z_{b_2} \left(K_{z_{44}} - K_{z_{24}} \right) + z_{b_4} \left(K_{z_{22}} - K_{z_{24}} \right)$$

$$q_3 = x_{b_1} \left(K_{x_{33}} - K_{x_{13}} \right) + x_{b_2} \left(K_{x_{24}} - K_{x_{44}} \right)$$
$$+ x_{b_3} \left(K_{x_{11}} - K_{x_{13}} \right) + x_{b_4} \left(K_{x_{24}} - K_{x_{22}} \right)$$

$$q_4 = x_{b_3} (K_{x_{11}} + K_{x_{13}}) + x_{b_4} (K_{x_{22}} + K_{x_{24}})$$
$$- x_{b_1} (K_{x_{33}} + K_{x_{13}}) - x_{b_2} (K_{x_{44}} + K_{x_{24}})$$

$$q_5 = z_{b_1} (K_{z_{33}} + K_{z_{13}}) + z_{b_2} (K_{z_{44}} + K_{z_{24}})$$
$$- z_{b_3} (K_{z_{11}} + K_{z_{13}}) - z_{b_4} (K_{z_{22}} + K_{z_{24}})$$

Solving equations (V-42) to (V-44) for x_0 , θ_0 , and ψ_0 and equations (V-45) and (V-46) for z_0 and ϕ_0 yields

$$z_0 = \frac{C_{52}\, q_2 - C_{22}\, q_5 - C_{22}\, (2W\bar{z}/h_y)}{C_{21}\, C_{52} - C_{22}\, C_{51}}$$

$$\phi_0 = \frac{C_{21}\, q_5 + C_{21}\, (2W\bar{z}/h_y) - C_{51}\, q_2}{C_{21}\, C_{52} - C_{22}\, C_{51}}$$

and with

$$\Delta = C_{11} (C_{32}\, C_{43} - C_{33}\, C_{42})$$
$$+ C_{12} (C_{33}\, C_{41} - C_{31}\, C_{43})$$
$$+ C_{13} (C_{31}\, C_{42} - C_{32}\, C_{41})$$

$$\Delta \cdot x_0 = q_1 (C_{32}\, C_{43} - C_{33}\, C_{42})$$
$$+ (q_3 + 2T/h_z)(C_{13}\, C_{42} - C_{12}\, C_{43})$$
$$+ (q_4 - 2W\bar{x}/h_y)(C_{12}\, C_{33} - C_{13}\, C_{32})$$

$$\Delta \cdot \theta_0 = q_1 (C_{33}\, C_{41} - C_{31}\, C_{43})$$
$$+ (q_3 + 2T/h_z)(C_{11}\, C_{43} - C_{13}\, C_{41})$$
$$+ (q_4 - 2W\bar{x}/h_y)(C_{31}\, C_{13} - C_{33}\, C_{11})$$

$$\Delta\Psi_0 = q_1 (C_{31} C_{42} - C_{41} C_{32}) + (q_3 + 2T/h_z)(C_{41} C_{12} - C_{42} C_{11}) + (q_4 - 2W\bar{x}/h_y)(C_{11} C_{32} - C_{12} C_{31})$$

E. COMPUTER PROGRAM

1. Description

The computer program is written in FORTRAN. It consists of a main program, four major subroutines, and a four subroutine integration package.

The program has been run successfully on a CDC 6400 cybernet system.

The fundamental variables of the integration are ordered in a matrix y as follows:

$y_1 = z$	$y_6 = \dot{\theta}$
$y_2 = \dot{z}$	$y_7 = x$
$y_3 = \phi$	$y_8 = \dot{x}$
$y_4 = \dot{\phi}$	$y_9 = \Psi$
$y_5 = \theta$	$y_{10} = \dot{\Psi}$

2. Input Quantities

The input quantities presently include the following:

- Mass moments of inertia of the skip
- Weight of the skip
- Skip dimensions
- Coordinates of the mass center
- Vertical velocity
- Rope torque
- Damping constants
- Roller spring stiffnesses and deflection limits
- Bunton spacing
- Guide beam moments of inertia
- Bunton misalinement
- Guide beam elastic modulus
- Integration time step
- Total integration time

3. Output Quantities

The present output includes the input data and for each time step and;

Time

Vertical position of mass center

x and z for each corner of the skip

F_x and F_z for each corner of the skip

Printer plots and CALCOMP plots of any desired variable can be obtained by means of a postprocesser program.

4. Program Listing

A FORTRAN listing of the program in operational form is provided in Appendix A.

F. EXAMPLE PROBLEMS

1. Basic Input

Several example problems have been run. The input common to these problems is as follows:

W = 29,400 lbs.

h_z = 6.0 feet

h_y = 25.0 feet

h_z = 6.0 feet

$\bar{I}_x$, $\bar{I}_y$, $\bar{I}_z$ = 20,000; 5500; 20,000 lb · sec^2·ft

$\bar{x}$ = 0.0 feet, $\bar{y}$ = 0.2 feet

$\bar{z}$ = 0.2 feet

Roller stiffness

x . . .17280 lb./inch

z . . .17280 lb./inch

Deflection limits 0.25 inch

Rope torque . . . 0.0 ft.-lb.

Bunton spacing . . .6.0 feet

Guide beam

Elastic modulus 30.0 x 10^6 psi

Moments of inertia 5.1, 25.3 $in.^4$

Velocity . . . 1200 ft./min.

Damping constants

ηx, ηz = .01%, .01%

These parameters are based on an existing skip and, therefore, the example problems demonstrate the applicability of the analysis to real situations.

2. Problems Run

The following problems have been run. Typical printer plot output is given in the figures designated.

a. Perfect alinement of buntons, plane motion. Figures V-4, V-5.

   In this problem the only motion excited is due to the leaning of the skip on the flexible guide beams. The motion takes place in the y - z plane only.

b. Right bunton 4 out-of-line 0.5 inch in the z-direction. Figure V-6.

   The motion takes place in the y - z plane. Figure V-6 shows the effect on the motion of one out-of-line bunton.

c. Right bunton 4 out-of-line 0.5 inch in the z-direction and left bunton 5 out-of-line 0.5 inch in the x-direction. Figures V-7 to V-11.

   All five degrees of freedom are excited. The contact forces are guite large exceeding 2 1/2 tons.

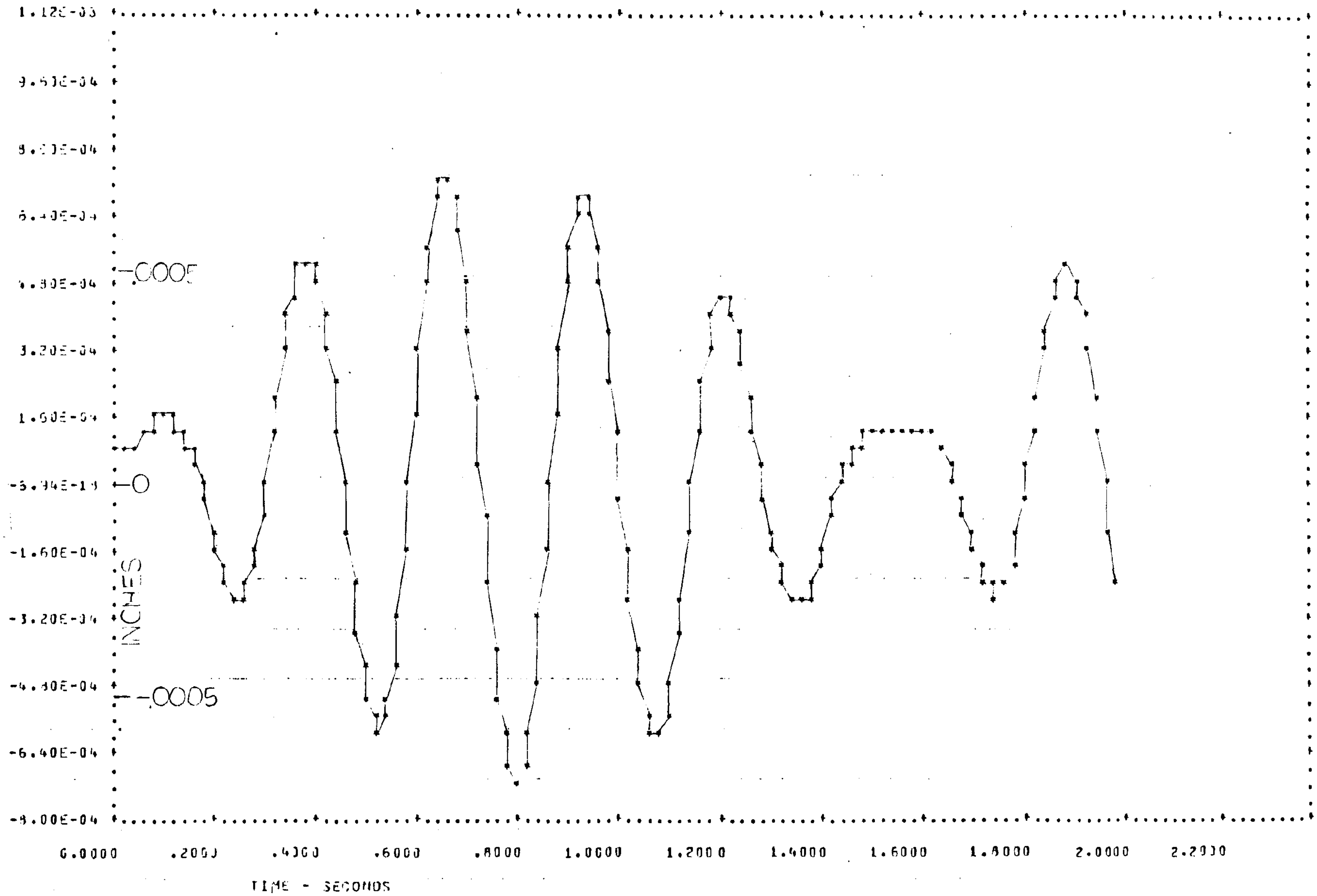


Figure V-4 z-Displacement for [illegible]rfect Alinement, No Interaction
y/z Plane Motion

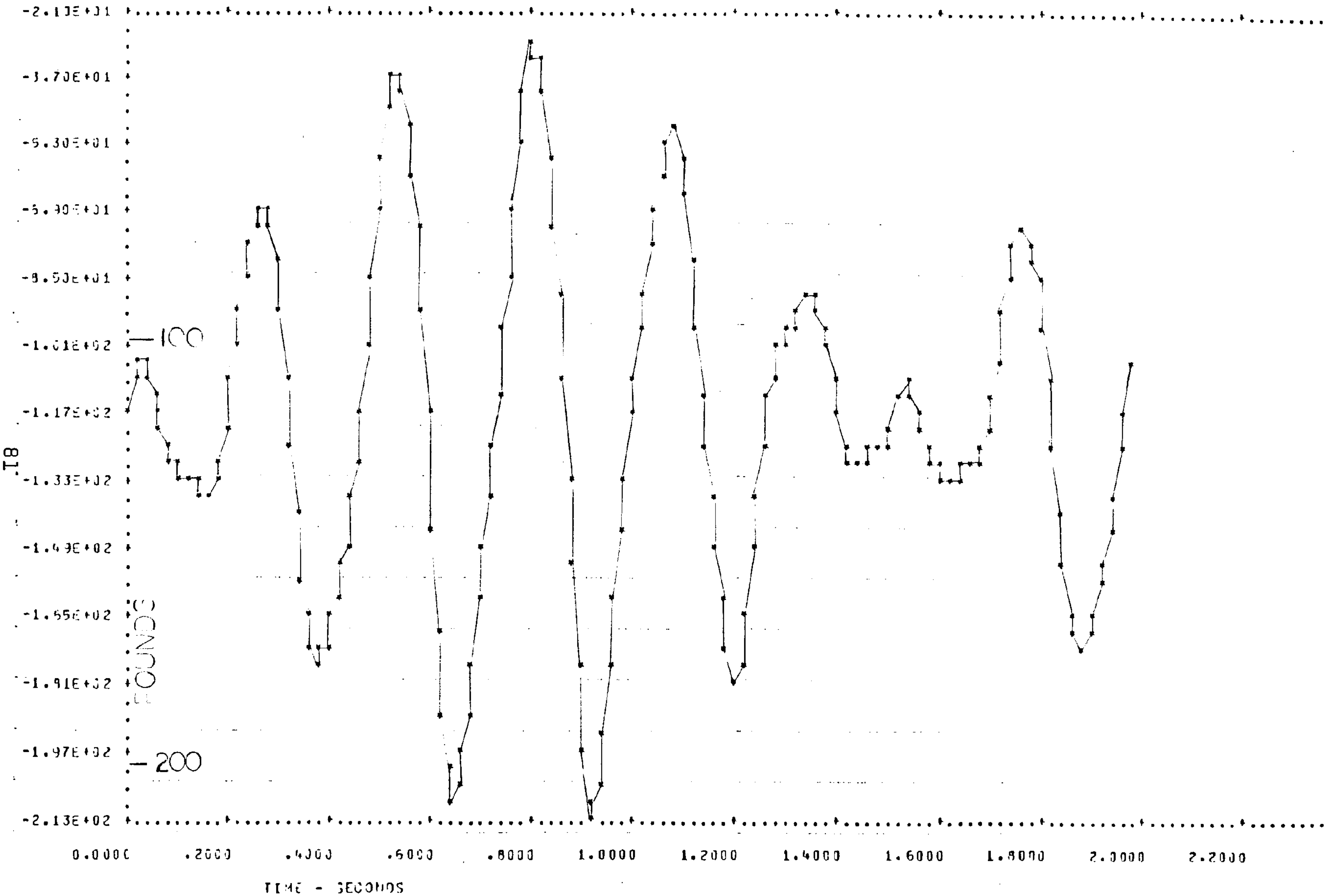


Figure V-5 Roller 1 Contact Force, for Perfect Alinement, No Interaction, y/z Plane Motion

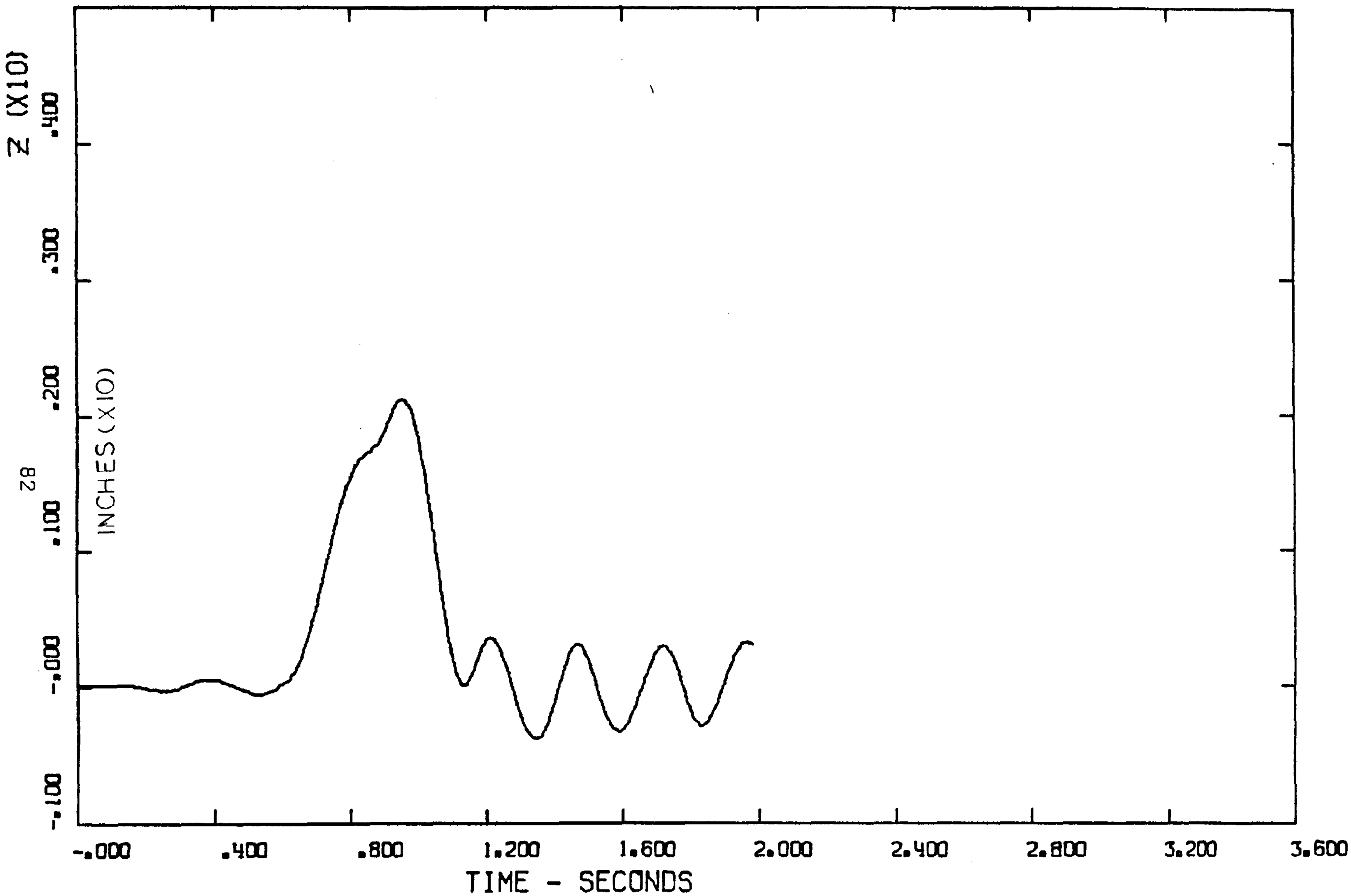


Figure V-6 z - Displacement, Bunton 4 Misalined 1/2 inch
No Interaction, y/z Plane Motion

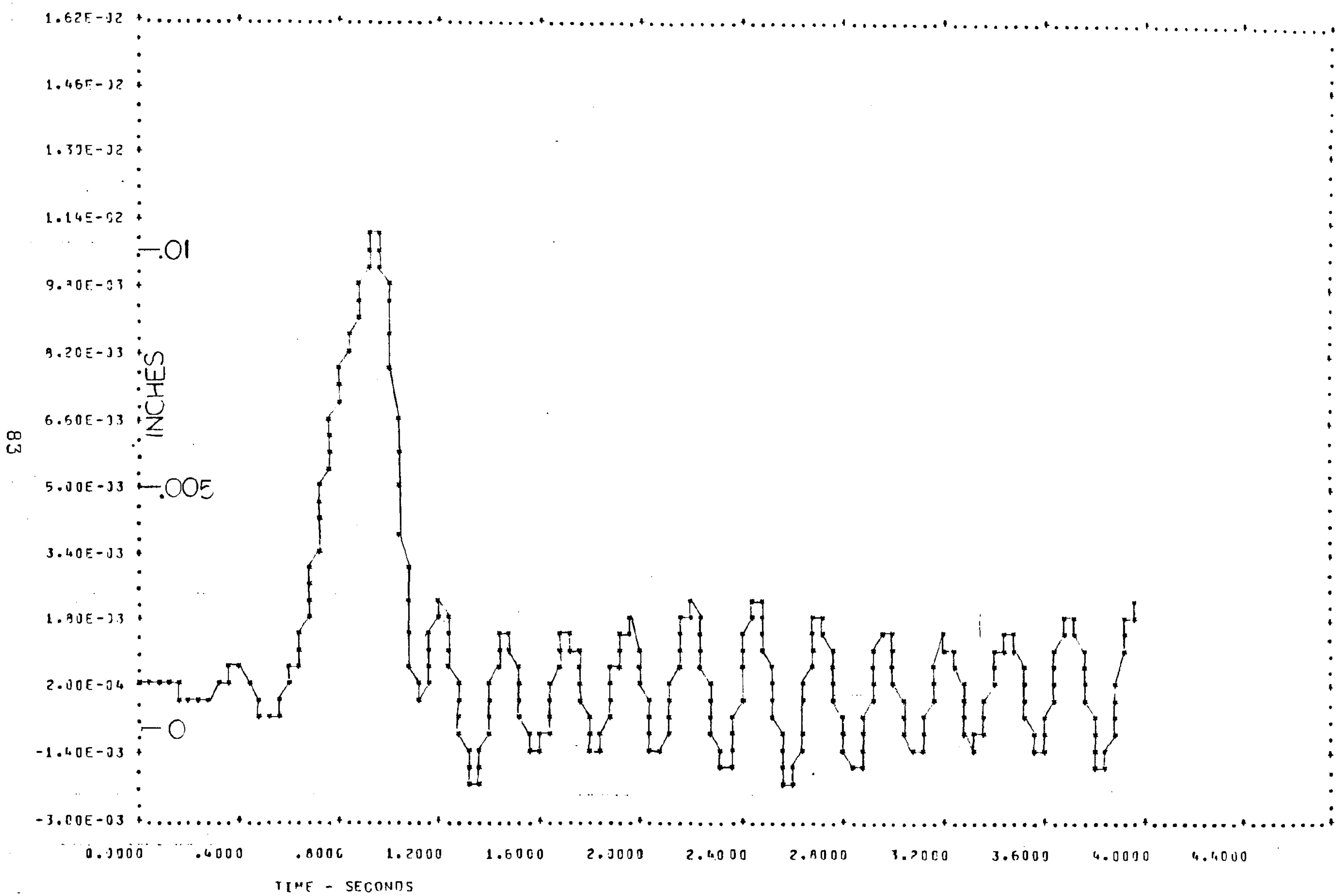


Figure V-7 z - Displacement, Buntons 4 and 5 Misalined 1/2 inch
Three Dimensional Motion

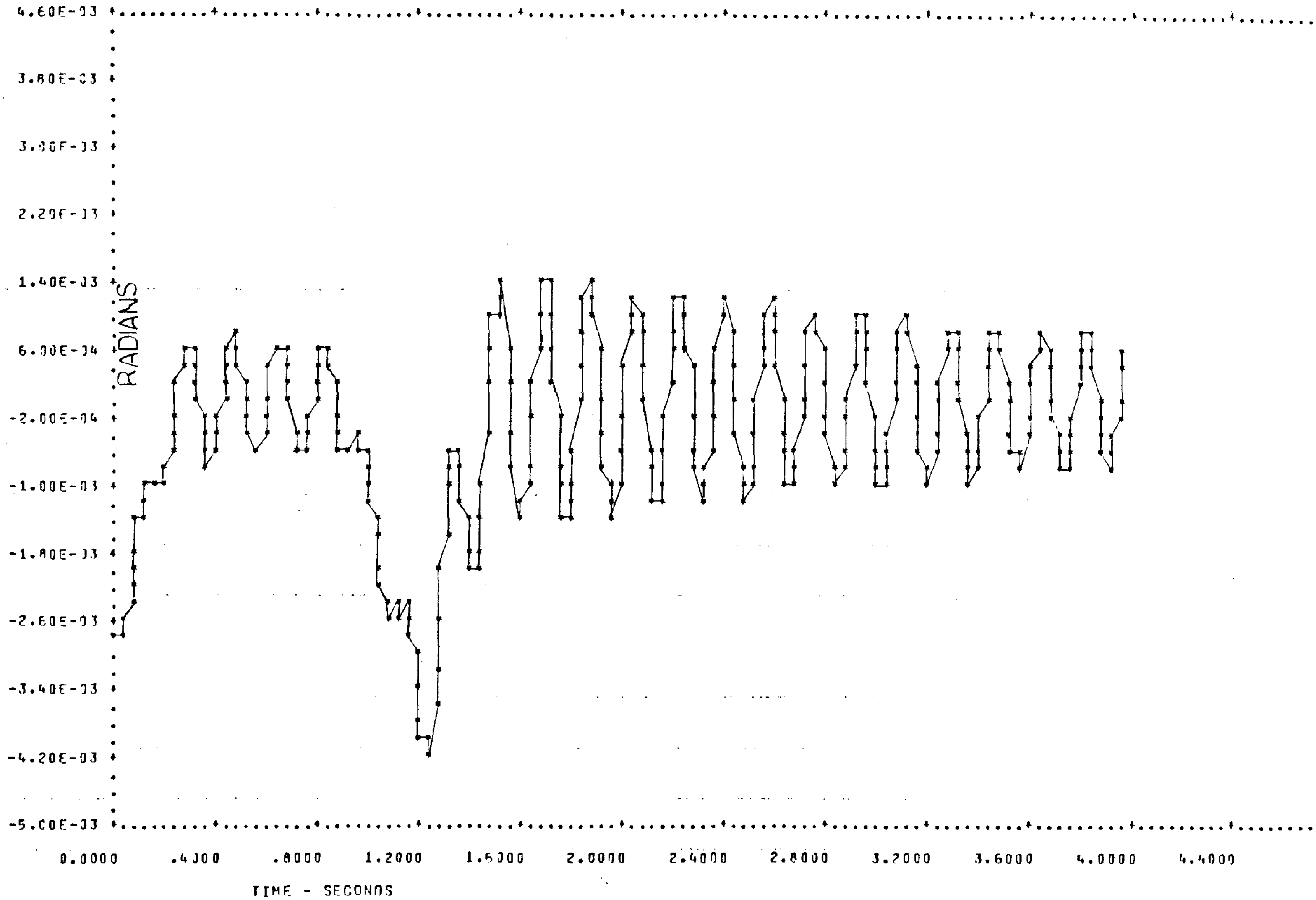


Figure V-8 Theta-Rotation, Bunton 4 and 5 Misalined 1/2 inch Three Dimensional Motion

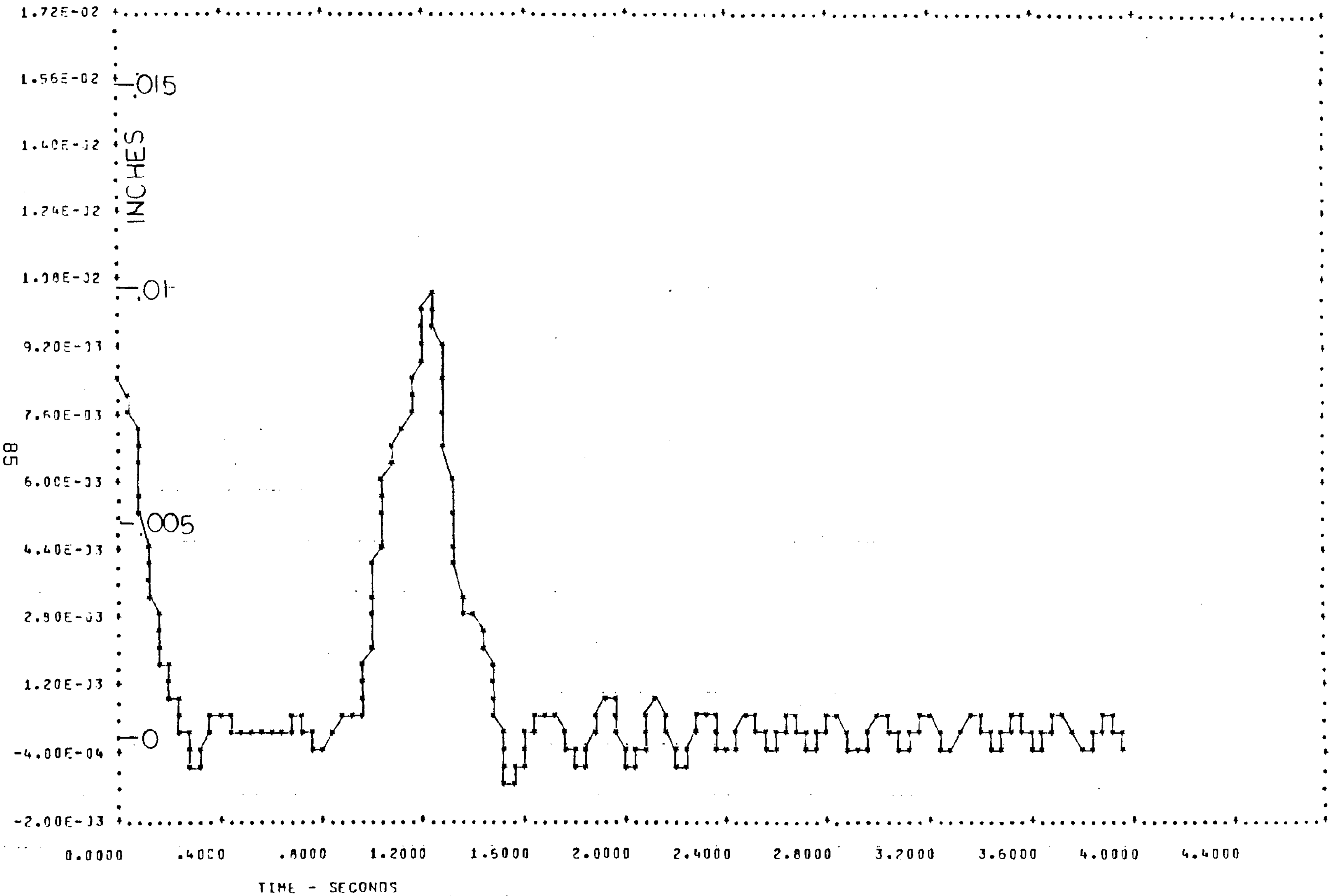


Figure V-9 x – Displacement, Buntons 4 and 5 Misalined 1/2 inch
Three Dimensional Motion

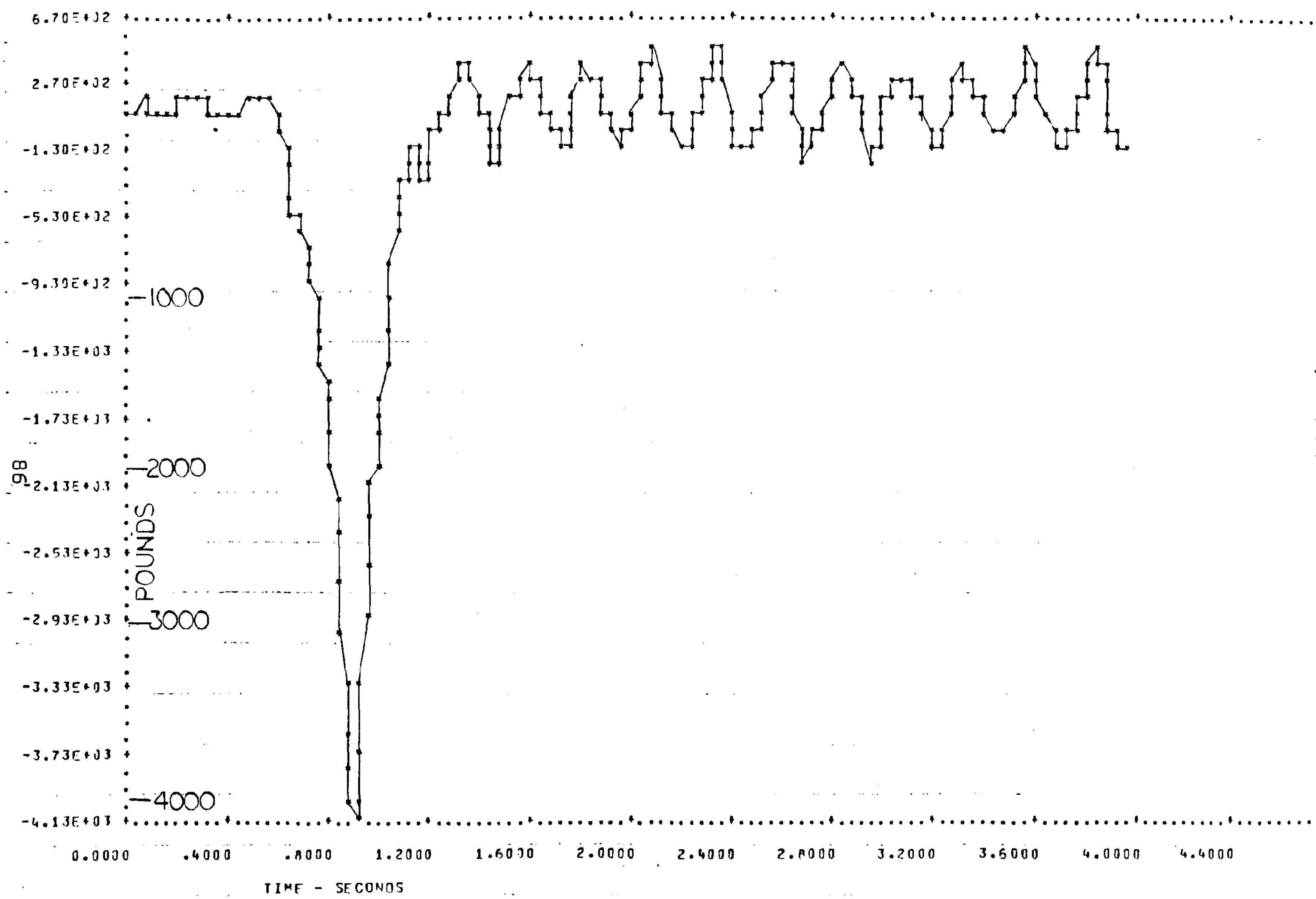


Figure V-10 z - Contact Force on Roller 4, Buntons 4 and 5 Misalined 1/2 inch, Three Dimensional Motion

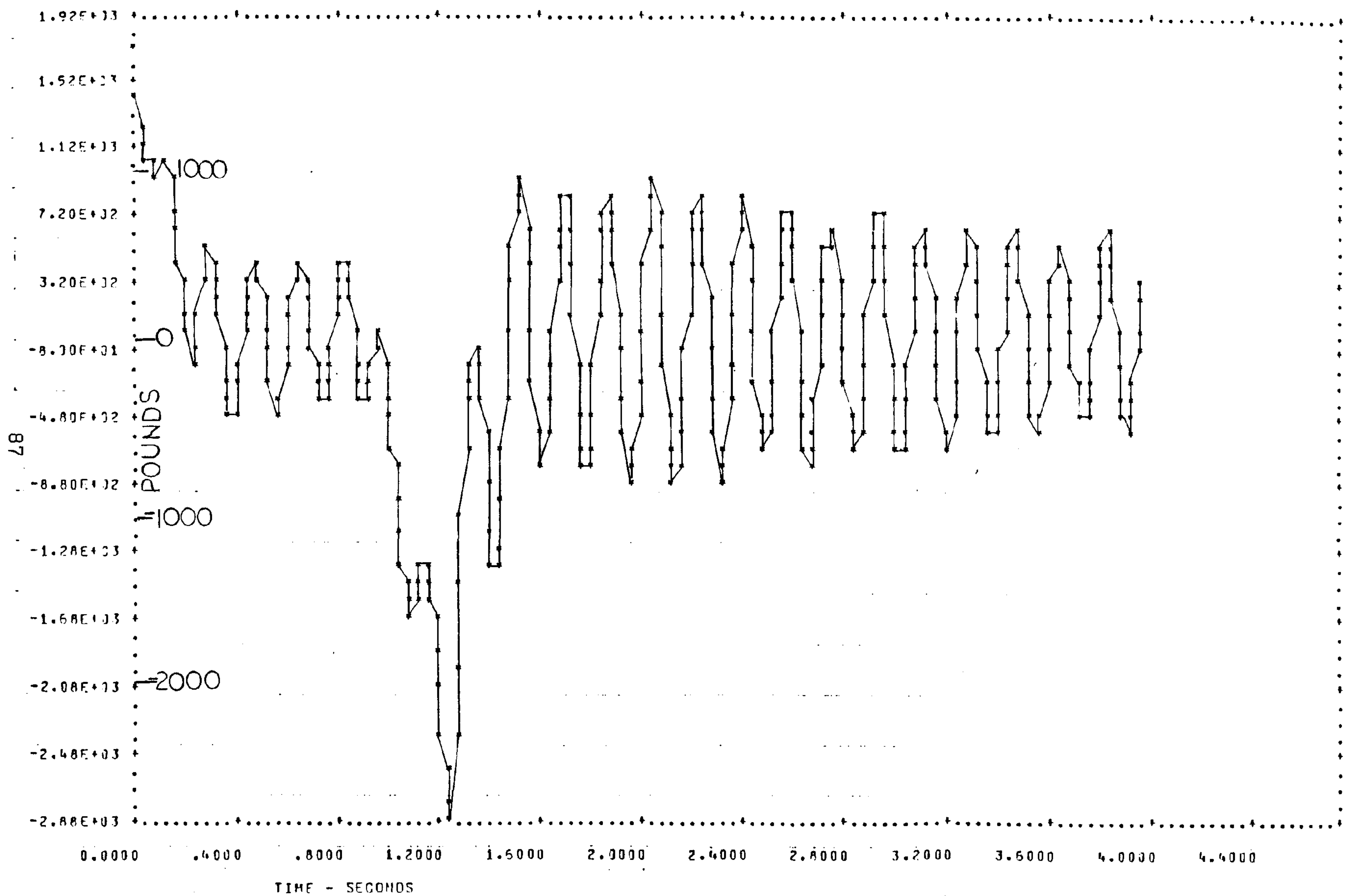


Figure V-11 x - Contact Force on Roller 2, Buntons 4 and 5 Misalined 1/2 inch, Three Dimensional Motion

VI. SKIP MATERIALS

Ferrous and non-ferrous materials have been used in the making of skips. Steels are usually used to satisfy structural, abrasion resistance, and shafting requirements. Elastomers are sometimes used for skip liners. Aluminum had been used in the 1930's to replace steel but is not widely used today, at least in the United States. Furthermore, fiber-reinforced composites have a high strength to weight ratio and offer reductions in skip weight. Following is a discussion of these various materials.

A. STRUCTURAL STEEL

Structural steels are preferred for skip bodies, bails, liners, guide shoes, and other miscellaneous skip parts that require good strength. ASTM A36 steel has probably been used most frequently for these applications. Table B-1 gives properties of ASTM A36. The high strength low alloy (HSLA) steels, however, have been used and offer the skip design engineer an attractive alternative because of higher mechanical properties and corrosion resistance. These steels are available from steel manufacturers as proprietary products and have yield strengths ranging from 40,000 psi to 115,000 psi. Table B-2 (reproduced from Metal Progress Databook; © American Society for Metals, 1976) identifies these steels, compositions, properties, and producers. It is divided into nine different groups, the last of which is abrasion-resistant alloys discussed in the next section. Because of a large number of HSLA steels available covering a range of strengths, corrosion resistance, and fabrication procedures, it is not practical to discuss each one in detail. It is of value, however, to discuss in general the improved effects of strength and corrosion resistance on skips and the resulting improvements on skipping. In addition, the fabrication of HSLA steels plays an important part in material selection for a particular skip component and is worthy of discussion.

HSLA Steels have higher strength, toughness (in conjunction with lower transition temperature), and abrasion resistance than plain carbon structural steel. The benefits of these properties were first used by manyfacturers of mobile equipment to achieve lighter units and increase payloads (Ref. VI-1). The economies achieved from this principle have offset the higher material cost of the HSLA steels many times over during the equipment service life (Ref. VI-2). When used for skips, other cost savings can be achieved through increased capacity, increased speed, decreased power consumption, decreased hoisting-machinery costs, and/or hoisting from greater depths. Less material is required, resulting in parts that are lighter, easier, and safer to work during skip manufacturing.

The improved atmospheric corrosion resistance of HSLA steels compared to that of structural carbon steel has been used as an index in evaluating materials for service such as in corrosive mine shaft environments. Improved corrosion resistance helps protect the thinner sections used where high strength steels reduce weight. On the other hand, a longer service life will be achieved if similar sections of corrosion-resistant materials are used to replace plain carbon structural steel. The increased corrosion resistance of HSLA steels has the added effect of increasing the paint life of coated surfaces. With some steels paint life has been increased by a factor of two (Ref. VI-3).

Incidently this point is a strong argument for these steels being used as buntons and guide sets where repainting is an expensive and time-consuming operation. Also, these high strength steels with smaller sections reduce shaft ventilation resistance. An interesting aspect of better corrosion resistance is the reduction in surface irregularities and thereby improving fatigue resistance.

Fabrication procedures play an important part in determining the most suitable material for a particular skip component. The single most involved fabrication operation is welding. Consideration must be given to pre-weld and post-weld heat treatment, proper filler metal (low hydrogen rods are often required), and material condition. Flame cutting requires similar attention. Minimum bend radii are characteristic of each material and must be considered when selecting a material for a part which is to be fabricated by bending. Machinability of HSLA steels is similar to that of alloy steels with similar hardnesses. Tables B-3 through B-8 give properties and outline fabrication operations for the T-1 and Cor-Ten families of HSLA steel. Careful comparison of these tables will show differences not apparent in the data given in the previous tables, such as availability in section thickness and fabrication operations. When considering HSLA steels, information should be sought from the steel maker and thoroughly studied to assure making a good selection.

B. ABRASION RESISTANT STEELS

Abrasion-resistant steels are used as skip liners mainly to protect the skip body and door from abrasion during loading and dumping. The most common abrasion-resistant steels used are AR steels and manganese steels. Mild steel and Cor-Ten steel have been used as liner materials but are really not abrasion-resistant steels. The abrasion service seen by a skip liner is usually grinding abrasion to impact abrasion, and a suitable abrasion-resistant material should have high hardness and impact resistance. AR and manganese steels have these properties and can be fabricated.

Table B-2 , Group IX, identifies abrasion-resistant steels, manufacturers, chemistry, condition, and hardness. Most of these steels are referred to as AR steels and are members of the HSLA family. A number of different grades are available to offer the user a range in wear resistance and fabricability. This is achieved through chemistry, controlled rolling and cooling, and section size. In general, abrasion resistance is dependent on the carbon content, microstructure, and hardness (Ref. VI-4); however, these properties affect fabricability. Therefore, consideration must be given to fabricability as well as wear properties when selecting an AR steel for skip liners. Fabrication information is available from manufacturers of these steels and should be referenced when making a selection. Usually the quenched and tempered steels are preheated prior to welding but are not post-weld stress relieved. The hot rolled steels can usually be preheated and stress relieved. Cold formability requires large radii and only the hot rolled steels are hot formed. Tables B-9 through B-11 give properties and fabrication information of three USS AR steels.

Manganese steel ("Hadfield's manganese steel") is of interest to the skip design engineer as a liner material. It has interesting work-hardening characteristics that produce up to 600 Brinell. Furthermore, the impact strength of this material is very high. The atmospheric corrosion resistance is approximately twice that of carbon structural steel. Three manganese steels are given at the bottom of Table B-2, Group IX. The difference in these steels is in the chemistry which affects weldability and high temperature properties. Machinability is possible but difficult, and finishing should be done by grinding. Manganese steels are available in both wrought and cast products. They have historically required a solution heat treatment at 1850°F followed by a water quench. Climax Molybdenum has been working the problem of developing as-cast manganese steels (Ref. VI-5).

Manganese steels are very attractive from an impact-abrasion standpoint but are not always acceptable to mine management. There is the possibility of a manganese steel part becoming a tramp metal in the ore. Since it is non-magnetic, it will not be removed by magnetic separators before entering process equipment and doing serious damage. Also, with the existence of this material at a mine site, there is the possibility that it will be mistakenly used as structural steel.

This can result in serious accidents while attempting to shear or punch it (Ref. VI-6).

C. SHAFTING STEELS

Shafting steels are used on skips. Typical locations of application are door hinges, body pivots, scroll roller shafts, and safety dog shafts. In these applications strength is the primary design criteria and the selection of an economical material with a high strength to weight ratio is desired. Cold finished plain carbon steel rounds have been widely used for many of these applications. Where very high stresses are involved, cold finished alloy steels are used.

Cold finished carbon steel rounds are commonly produced by two different methods: (a) the removal of surface material by turning and/or guiding and (b) drawing the material through a die (Ref. VI-7). In most cases method (b) produces the preferred material. Typical commercial drafts of 1/32 to 1/16" create a reduction in cross sections of usually less than 12 percent. Through these reductions, increases in properties such as yield and tensile strength, yield/tensile ratio, hardness and machinability are economically realized. Post draw stress relief provides a further increase in strength due to the relief of stresses created during drawing. Table B -12 gives mechanical properties of some common cold drawn steels. Values are listed for three different conditions: as drawn and two different stress relief treatments. Data on the effect of cold drawing on impact strength and transition temperature are limited, but these properties worsen in the range of commercial drafts. Cold drawing improves the machinability of carbon steels because the increased hardness causes chips to tear away more readily and break. Table B -13 gives the machinability ratings of many cold drawn steels as well as the strengths for hot rolled and cold drawn conditions. Also included in this table are similar data for the 1100 series resulpherized free machining steels. Properties for two commonly used cold drawn steels, 1018 and 1040, are presented in Tables B -14 and B -15.

Close control of properties of cold drawn steel is not practical. Variables such as (a) variation of composition in a certain grade, (b) temperature of bar in last pass of hot rolling mill, (c) rate of cooling after last pass of hot rolling mill, (d) amount of cold draft, (e) wide tolerance bands of hot rolled bars to be cold drawn yielding different drafts, all affect final properties. Consequently minimum properties listed for cold drawn steels are far below the normal expectancy. Above 6" diameters, cold drawn carbon steels are not produced.

For high strength applications alloy cold finished steels are used. Cold drawing is performed, but alloy steels are more commonly heat treated to the desired strength level and then turned and polished. By this procedure the high strength levels characteristic of the alloy steels can be accurately achieved. Some of the more commonly used alloy steels are AISI 4340, 4140, and 4150. Yield strengths from 150-200 ksi are obtainable with these grades. Because of the high strengths, machinability of these steels is not particularly easy. Leaded or resulpherized grades can be used to increase machinability.

D. ELASTOMERS

Elastomers have properties that make them attractive from a skip liner material standpoint. Manufacturers of rubber type elastomers claim that ten times the abrasion resistance of steel can be achieved with these materials despite being considerably softer (Ref. VI-8,9). This is attributed to the resilience of sufficiently thick linings in absorbing energy through elastic deformation. Hard wear plate materials do not have this property and consequently chip and wear when exposed to abrasion service. The energy-absorbing liner between the skip and muck also may reduce some of the vibration, shock and fatigue normally transmitted to the skip during hoisting. Noise is likewise reduced. A further benefit of a flexible liner is the lessened sticking and build-up of material, thus enhancing handling of sticky ores.

Elastomeric liners can be furnished in grades resistant to corrosion. The corrosion problem typical of most metal liners is therefore greatly overcome.

These liner materials are normally available in sheet or rubber-faced steel plate. Direct casting to metal parts, however, has been performed with some elastomers when ultimate adhesion was required. Fastening of elastomer sheet can be done by adhesive bonding or mechanical fastening. Rubber-faced steel plate is almost always fastened with mechanical fasteners, but welding is possible if skill and care are exercised. Mechanical fastening can be very advantageous in high wear areas where frequent and rapid replacement is desired to minimize down-time. Many fabrication procedures and techniques for rubber-faced steel plate are described by the The Gates Rubber Company (Ref. VI-10). In general, rubber linings can be quite convenient to work with because they are flexible and weigh approximately one-seventh that of steel. This ease of workability can reduce man-hours in both the manufacture and field replacement of skip linings. Because of the weight reduction over a steel liner, hoisting benefits of a lighter skip result.

Determination of the most economical elastomeric liner for a given application can best be made by actual use tests. Basic material properties such as durometer hardness, tear strength, impact strength, and others are available from elastomer manufacturers and can be helpful in making a selection decision. The following are some manufacturers of abrasion-resistant elastomers located during this study: B. F. Goodrich Chemical Company; Linatex Corporation of America; Acushnet Company; Dupont; Trelleborgs (Sweden); The Gates Rubber Company; Ohio Rubber; Masonite Corporation; Plastic-Techniques, Incorporated.

E. ALUMINUM

Aluminum skips received attention as early as the 1930's and 1940's because of reduced skip weights of 30 to 60 percent (Ref. VI-11, 12). This weight reduction offered the benefits of a lighter skip (presented in the structural steel section of this report). Aluminum skips have, however, had service problems due to low fatigue strength (Ref. VI-13), corrosion, and abrasion resistance. Soutar (Ref. VI-6) states the status of aluminum skips in South Africa:

> "The use of duralumin was discontinued due to problems arising from fatigue failure, electrolytic corrosion, high capital cost and the introduction of improved higher tensile and corrosive and abrasive steels."

In Canada 80 percent of shaft conveyances are made of aluminum. Lower strength, more corrosion-resistant alloys have achieved a payload ratio of one half for skips.

F. FIBER-REINFORCED COMPOSITES

Fiber-reinforced composites have high strength and low density. The benefits of composites have been widely used in the aerospace industry where a high strength to weight ratio is of the utmost importance. Application to skips could provide skips with lower payload ratios. Figure VI-1 shows specific strength vs. specific modulus for some common fibers, composites, and metals. It can be clearly seen that glass, boron, and graphite fibers are significantly stronger than bulk metals. Typical values of strength for these fibers are 400,000, 400,000, and 500,000 lb/in^2, respectively; and tensile moduli are 10.5×10^6, 55×10^6, and 75×10^6 lb/in^2, respectively, (Ref. VI-14). Understandably, then, the existence of these fibers as a phase in a plastic matrix contributes largely to the high strength to weight ratio of fiber-reinforced composites. It is

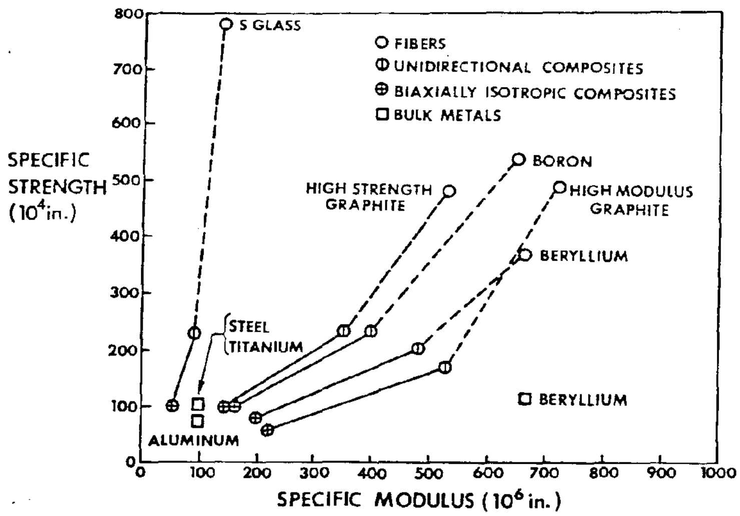


Figure VI-1. Strength and Stiffness of Advanced Composite Materials (Ref. VI-15).

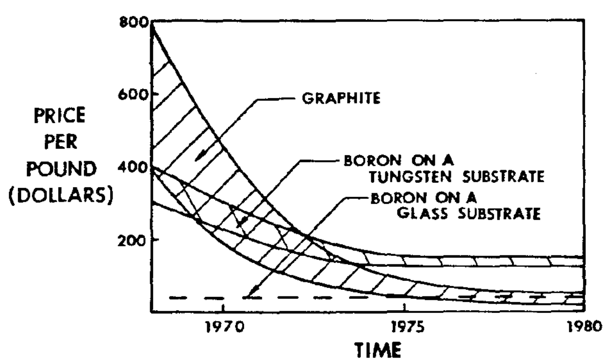


Figure VI-2. Predicted Cost of Boron and Graphite Fibers (Ref. VI-15).

possible to tailor composites to have the same strength and stiffness as steel-yet be 70% lighter. Other composites are three times as strong as aluminum but only weigh 60% as much (Ref. VI-15). The attractive application of composite materials to skips is circular skip body reinforcement, where the high strength properties of a unidirectional composite could be maximized in reducing hoop stress.

A projected cost per pound is given in Figure VI-2 for the high strength fibers - graphite and boron. It is anticipated that with the imminent increased interest in advanced composites, production techniques of these fibers will improve to the point where advanced composites will be competitively priced with other materials. Another economical consideration of fiber-reinforced composites is the minimal scrap created during manufacture of a part. This is due to the ease of making a variety of shapes and sizes with minimal machining and stock removal.

VII. GUIDE MATERIALS

Three types of guides are used for conveyances in mine shafts. They are wood, steel, and rope. Following is a discussion of these guides:

A. WOOD GUIDES

Formerly the main structural material in mine shafts was wood. It is well suited for the short spans of rectangular shafts and was commonly used for guides. Modern shafts, however, are generally circular and require longer bunton spans. Steel is now used to meet these structural requirements and is also used for guides in many applications (Ref. VII-1,2). Wood guides, however, are still preferred to steel wherever a man conveyance operates because of the reliability of wood guides engaging the arrestment device. Safety dogs bite into wood guides to stop a conveyance. Steel guides have not been used as much for this arrestment application. Wood guides may be selected where anticipated mine life is short and replacement of guides is not expected. Also, the arbitrary factor due to past experience of mine personnel using wood guides may determine the selection of wood guides for a shaft.

Disadvantages of wood guides lie primarily in shaft maintenance. Wood guides require frequent wetting to prevent drying out. Wear life of wood guides is often less than that for steel guides and may need to be replaced during the life of the mine.

Several different woods are used for guides. They include Karri (Eucalyptus diversicolor), Tallowwood (Eucalyptus microcorys), Douglas fir (P. taxifolia), Keruing (Dipterocarpus spp.), Apitong (Mahogany), and local pine (Pinus ponderosa). Of these, Karri and Tallowwood have consistently good mechanical properties and density. Douglas fir has probably been used most frequently but is becoming quite expensive for clear grade. Keruing and Apitong, though having good mechanical properties, are inclined to warp and vary in density. Local pine has the advantage of easy availability. See Table VII-1 for wood properties.

B. STEEL GUIDES

Steel guides are used because they have good strength, promote smoother skipping operations than wood guides, last longer than wood guides, and require less maintenance. Ventilation considerations result in bunton spacings of 15-20 feet. Steel guides 30 to 40 feet long satisfy this requirement conveniently and provide good strength (Ref. VII-1). Smooth skipping operations have been ascribed to

Table VII-1 Properties of Some Wood Guide Materials*

Property	Douglas Fir	Karri	Keruing-Apitong**
Straightness	Straight	Straight and Flat	Some species warp
Strength***	Moderate	High	Moderate
Grain Interlock	No	Yes	No
Grain Straightness	1" in 12"	1" in 15"	No Malaysian spec
Uniformity of Grain	Canadian-U.S. Grades	Australian Authority	Malaysians accept No Responsibility
Consistency of Properties	Coast species Uniform	One Species, 65 pcf	31 Species, 40 to 55 pcf
Availibility	To 40 feet	Occasionally 48 ft Readily to 30 ft	Occasionally 30 ft. Readily to 20 ft.
Durability Under Wet & Dry	Requires treatment	No treatment req'd	Requires treatment
Pricing	Clear Grade-High Struc. Grade- Like Best Hardwood	Similar to Untreated Apitong, lower than Untreated Fir	Reasonable
Usage	Mostly U.S. & Canada-30 to 40 yrs.	World-wide - 50 yrs	U.S.- 10 yrs.

* U.S., Canadian, Australian and Malaysian grading rules as compiled by EVJU Products Co., Inc.
** Also referred to as Mahogany
*** See handbook for green and 12 percent moisture strengths

the use of steel guides (Ref. VII-2,3). This contributes to longer life that is expected from steel guides, because loads due to skip-guide interaction are reduced. Furthermore, the use of steel guides eliminates the need for wetting and/or chemically spraying that is required with wood guides. One disadvantage of steel guides is that safety dogs cannot be used in conjunction with manned conveyances. When manned conveyances are operated on steel guides, an alternate safety concept must be incorporated. Consequently, skips usually operate on steel guides, and manned conveyances operate on wood guides.

Sections used for steel guides are the rail, elevator tee, hat, and square or rectangular hollow shape. Rail sections are generally suitable for small capacity conveyances or service conveyances in large shafts (Ref. VII-4). The elevator tee is available in four sections (Ref. VII-5) and apparently is not widely used. This could be because it is similar to the rail section but not as readily available. The hat section was developed in South Africa to replace channel guides that were common there (Ref. VII-4). It was designed for superior wear and transverse stiffness. Recently, especially in Canada, the square and rectangular hollow shape has been accepted and used for steel guides (Ref. VII-1,2).

The material used for rail guides is common railroad rail sections. Consequently, they are made from typical rail steel (Ref. VII-6). Square or rectangular shapes are now commonly produced with 50,000 psi yield strength steel (Ref. VII-3). Some mines galvanize their guides to reduce corrosion and maintenance costs (Ref. VII-6,7). Cor-Ten steel has been used as a guide material because of its good corrosion resistance and improved effect on paint life.

C. ROPE GUIDES

Rope guides are used because they provide less resistance to air flow, eliminate dynamic forces that occur from misalined fixed guides, are faster and more economical to install, and require considerably less shaft maintenance and guide alinement. Disadvantages of rope guides are little resistance to small continuous side loads, shaft depth limitations due to rope weight, problems associated with arrestment of conveyances, and operation of multiconveyances in a shaft.

Guide ropes fpr conveyances in mine shafts are made with a locked coil construction. Figure VII-1 shows two different configurations for this type of construction (Ref. VII-10). Locked coil ropes have a smoother outside area than stranded rope. This is shown in Figure VII-2 and results in longer wear because of more surface area carrying frictional loads. The outer wires are locked together by their shape, which results in broken wires being contained, obviating an accident. The locked construction inhibits internal corrosion by containing internal lubricant and preventing moisture from entering the rope. Generally locked ropes are stronger than stranded rope and are virtually non-rotating. (Ref. VII-10).

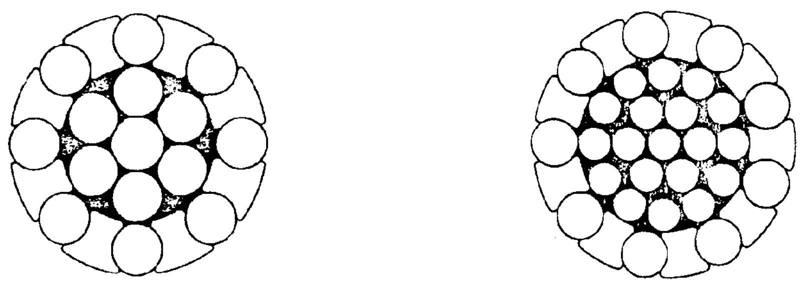


Figure VII-1 Locked Coil Construction of Two Guide Ropes

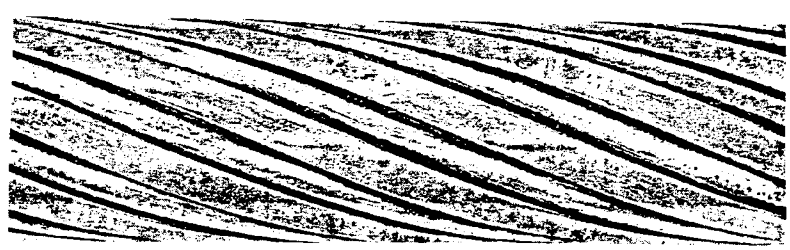


Figure VII-2 Smooth Surface of Locked Coil Guide Rope

VIII. VENTILATION AND PAYLOAD RATIO

A. VENTILATION

Ventilation power costs are significant over the life of the mine and are dependent primarily on mine shaft air flow resistance. Obstructions in the ventilation airway create power losses that have to be compensated for by increased fan output. The cost of operating the fan unit is the primary expense of ventilation (Ref. VIII-1). By good downcast shaft design, the obstructions to air flow can be reduced and ventilation economies achieved.

1. Air Flow Through a Shaft

Mine air flow is described by Atkinson's formula, as given by Hartman, p. 89 (Ref. VIII-2).

$$P = \frac{KSV^2}{A} \qquad \text{(VIII-1)}$$

where P = head loss pressure, psf

K = friction factor, lb min^2/ft^4

S = rubbing surface of airway, ft^2

V = velocity of air flow, ft/min

A = cross-sectional area of airway, ft^2

and the air horsepower drawn for this pressure is

$$H.P. = \frac{PQ}{33000} \qquad \text{(VIII-2)}$$

where Q = flow rate in ft^3/min = VA

If Q, and hence V, is specified for a fixed diameter shaft of area A, one way to reduce P (and the power costs) is by reducing K, the friction factor of the shaft, (Ref. VIII-2).

2. Parameters Affecting K

There are a number of shaft parameters that affect K, but three concerned with buntons have been given primary consideration by investigators. These considerations are:

(1) The aerodynamic shape of the buntons in the shaft,
(2) The bunton frontal area, and
(3) The bunton spacing.

Research has been done to determine the effect of bunton shape on shaft resistance. Table VIII-1, reproduced from

Table VIII-1. Shaft Resistance of Bunton Sections

Type of bunton section	Unit cost percentage	Approximate shaft resistance
6" Beam	100	100
6" Mushroom I beam	120	64
6" Covered I beam	150	45
6" Aerofoil I beam	200	42
6" Wide flattened pipe section	130	45
6" Prismatic section	125	43
6" Aerofoil section	350	40
4" Prismatic section	110	28
4" Aerofoil section	300	25

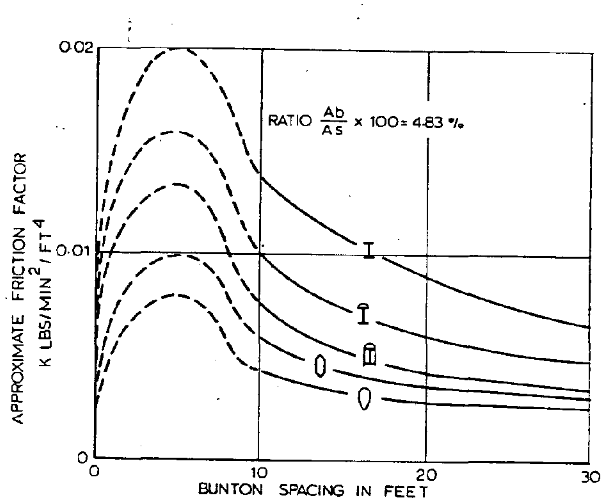


Figure VIII-1 Effect of Bunton Spacings on Friction Factor for Four Inch Wide Buntons a paper by Van Wyk (Ref. VIII-3), gives comparisons of the effect of different bunton shapes on shaft resistance as well as relative costs of these different shapes. It can be seen that for the six-inch thick shapes, the most efficient one is the aerofoil section. This section, however, has not received much attention because: 1) it is expensive, 2) it is not easy to install or connect to other components, 3) other more convenient shapes are available at less cost, having only slightly poorer aerodynamic properties. The prismatic section (or hexagonal diamond shape, as it is often called), developed in South Africa, has been used successfully. The current trend, however, in both South Africa and Canada is the use of the flattened pipe section.

The reduction of bunton frontal area provides further reduction in shaft resistance, and this is also shown in Table VIII-1. The 4-inch prismatic section has a resistance of 28 compared to 43 for that of the 6-inch section.

Figure VIII-1, reproduced from a paper by Gibson and Anderson (Ref. VIII-4), shows the effect of spacing on friction factor for different bunton shapes. Reductions in friction factor for spacings over 15 feet are small. Consequently, 15-20 foot spacings are considered good design practice and are common in many mines designed with ventilation economies in mind. Also this spacing results in convenient guide lengths of twice the spacing and is widely accepted by mine personnel.

Wind tunnel tests have been used in determining the effect of bunton shape, frontal area, and spacing on shaft resistance. It has been demonstrated that scale models with true geometrical detail provide accurate friction factors for real shafts. This is because of the independence of friction factor with Reynolds number, thus eliminating the scale effect (Ref. VIII-1 and VIII-5).

Two other major parameters that can contribute to reduce shaft resistance are the use of rope guides instead of fixed guides and smooth shaft linings.

The use of rope guides instead of fixed guides reduces the shaft resistance significantly because transverse bunton obstructions are eliminated. The resistance of rope guides to that of buntons is relatively small because they are parallel to the direction of flow. Van den Bosch and Drummond (Ref. VIII-6) state that " tests confirm previous investigations that obstructions running parallel with the air flow cause only minor increases in the K value, but even small obstructions normal to the flow have a marked adverse affect."

Shaft lining smoothness contributes to efficient ventilation. Concrete-lined shafts are used partially for this reason. Furthermore, bunton connections to the shaft wall should be made as flush as possible to minimize transverse obstructions at the wall. Minimum shaft lining area per unit cross section contributes to reduced friction and is achieved with the circular shaft (Ref. VIII-7). Consequently, circular shafts are desirable from a ventilation standpoint.

The pipe columns, cables, and supports for the auxiliary services and the skip do not contribute to resistance significantly. Bentley (Ref. VIII-5) claims that: "Pipes and other services and their supports contribute from 10 to 20 percent of shaft ventilation resistance " No similar information was found for skips. Fairing of skips, however, is suggested by Wells (Ref. VIII-8) to reduce shaft resistance.

3. Criteria for Selecting Buntons

Gregory (Ref. VIII-1) gives four criteria for bunton selection:

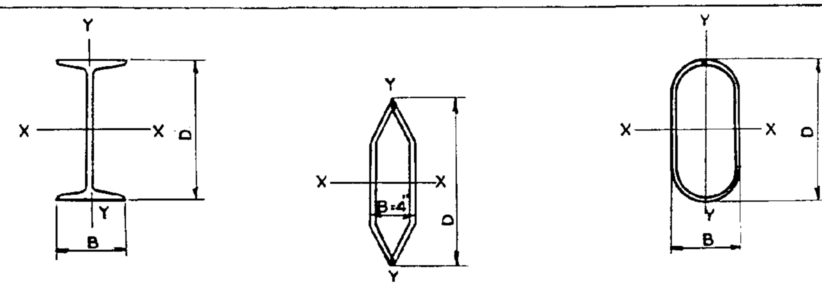
(1) The member should have sufficient structural strength to resist all normal and expected loads, with a reasonable margin of safety.
(2) It should be of narrow width and of streamlined profile so as to offer minimal resistance to air flow.
(3) It should be inexpensive to purchase, to install and to maintain in operation.
(4) It should be easy to handle, place, aline, adjust, and connect to other components.

Aerodynamic shapes lend themselves quite well to satisfying most of these criteria and in some areas have significant advantages over standard I-beams.

Table VIII-2, reproduced from Bentley's paper (Ref. VIII-5), gives weights, moments of inertia, and section modulus for some I-beams, diamond sections, and squashed pipe sections developed in South Africa. Researchers doing static and dynamic loading tests on these sections favored the diamond and squashed pipe sections over the standard I-beam. It has been established that hollow structural shapes have a superior weight to compression strength ratio (Ref. VIII-4). These shapes are being produced to yield strengths of 50,000 psi. One example stated that a weight saving of 30 percent was achieved by using aerodynamic bunton shapes in a shaft (Ref VIII-5).

Other non-aerodynamic advantages of hollow structural shapes stem from improved installation, corrosion resistance, and easier maintenance. Installation of these sections is easier because they are lighter and have

Table VIII-2 Comparative Section Properties



SIZE D X B	WEIGHT PER FOOT	MOMENTS OF INERTIA		MODULI OF SECTION		SIZE	WT/FT	M OF I	M OF I	M OF S	M OF S	SIZE D X B	WEIGHT PER FOOT	MOMENTS OF INERTIA		MODULI OF SECTION	
		X – X	Y – Y	X – X	Y – Y	D×D	LBS	X-X	Y-Y	X-X	Y-Y			X – X	Y – Y	X – X	Y – Y
8" x 6"	35	115·06	19·54	28·76	6·51							8" X 6"	27·3	56·1	37·8	14·2	12·6
10" x 6"	40	204·8	21·76	36·11	7·25							10" x 6"	32·3	104·27	49·6	20·85	16·5
12" x 6"	44	316·76	22·12	52·79	7·37							12" x 6"	37·4	154·6	61·4	28·4	20·5
14" x 6"	46	442·57	21·46	63·22	7·15	14½" x 4"	41	215	24	30		14" x 6"	44·5	257·16	73·2	36·7	24·7
16" x 6"	50	618·09	22·47	68·88	7·49	16½" x 4"	46	314		38		16" x 6"	47·6	367·1	85·0	45·88	28·4
18" x 6"	55	841·76	23·64	93·53	7·88							18" x 6"	52·7	515·26	96·8	57·9	32·2
20" x 6	65	1226·17	32·56	122·62	10·02	19½" x 4"	53	502		53		20" X 6	57·9	688·6	108·6	68·86	36·2

better torsional rigidity. Corrosion resistance is improved because of the closed smooth section and reduced surface area. This makes galvanizing and painting operations more economical. With the improved corrosion resistance, maintenance costs are reduced.

Mine personnel at the Kidd Creek Mine of Ecstall Mining Limited in Canada expect the following results from new zinc-coated aerodynamic shaft steel (Ref. VIII-6):

> "We expect ten to fifteen years of service before any significant maintenance will be required. We know that the coating will get scratched and gouged during installation and service, but the galvanic action of the zinc will still give us the protection we need. And after ten years or more, when we finally have to do some painting, our costs will be minimal because we will not have to do a major surface preparation job."

4. Impact of Ventilation Considerations on Mines

Mines with dry shafts should be especially concerned with the aerodynamic considerations of ventilation in shafts because of the savings that can be achieved by reducing the friction factor K. This is discussed in papers describing efforts of some mines to reduce the friction factor K to achieve ventilation economies (Ref. VIII-4 thru 7 and 9 thru 12). In general, friction factors have been reduced by at least a factor of two by using aerodynamic shapes (hollow structural sections) for shaft steelwork rather than standard structural I-beams. In the case of one mine (Ref. VIII-5), annual savings in ventilation costs were 26 percent of the capital cost of the aerodynamic steelwork over that of structural joists, when manufactured to required jig accuracy, was only 10 percent. At another mine (Ref. VIII-4), the effect of installing streamline buntons was to increase the air flow through the mine from 800,000 to 1,000,000 cfm, an increase of 25 percent. Anaconda reported in 1938 that an increase in power of 70 percent more than doubled the air flow by streamlining air shafts, while working temperatures were lowered 6 - 10°F at 1000 working places (Ref. VIII-13). Figures like these indicate that aerodynamic shaft efficiency is a worthwhile design consideration. In addition, mining to deeper depths becomes more attractive; and the possibility exists in some cases that with better air flow resulting in reduced air-strata contact time, mine refrigeration may be postponed or eliminated (Ref. VIII-14).

Most of the information found regarding ventilation was from Canada and South Africa. Rope guides are popular in Sweden and Great Britain (Ref. VIII-16). Consequently, most of their shafts do not have added air flow resistance created by guide sets. It appears from the mine visits and contacts made during this study that most mine shafts in the United States are not aerodynamically equipped.

Note that the mining law for coal mines requires that fans be operated continuously (CFR Title 30, Coal Mine Health and Safety Act of 1969, Section 75,300-3). Presumably, efforts are being made to combine the coal mine law into the metal and non-metal mine law for a unified law. It can be expected that in the future metal mines may have the same requirements for continuous ventilation.

B. SKIP PAYLOAD RATIO

The skip payload ratio (Ref. VIII-16) is given by

$$r_p = \frac{w_s}{w_c} \qquad \text{(VIII-3)}$$

where r_p = payload ratio
w_s = skip dead weight, lbs
w_c = skip muck capacity, lbs

A low payload ratio indicates an efficient skip. Existing skip payload ratios generally vary from 0.4 to 1.3 and are not related to tonnage (see Figure I-3). Since skip and muck weight have a pronounced effect on hoisting, it is felt that the effect of this parameter on hoisting should be investigated since it is derived from skip weight and skip muck capacity. A practical reduction in payload ratio can reduce the hoisting rope size and thus reduce capital and operating costs of hoisting equipment.

The effect of payload ratio can be shown with an example. Rope size is determined for a 2500-foot shaft for two 20,000-pound capacity skips with different payload ratios. Payload ratios of 0.75 and 0.50 are considered. The rope load is the sum of the loaded skip weight and 2500 feet of Roebling Royal Blue Mine Shaft Rope (Ref. VIII-17). It is found that the 0.75 payload ratio skip requires a 1 5/8 inch rope and the 0.50 payload ratio skip requires a 1 1/2 inch rope; thus, by reducing the payload ratio 0.25, the rope size is decreased.

When a shaft depth of 6500 feet is evaluated, the 0.75 r_p skip requires a 1 7/8 inch rope and the 0.50 r_p skip requires a 1 5/8 inch rope. In this case the difference in rope size is 1/4 inch. These calculations consider safety factors of 5 for the 2500-foot shaft and 4 for the 6500-foot shaft, as given in the Code of Federal Regulations. (Ref. VIII-18).

The effect of reducing the skip payload ratio can reduce hoisting capital costs because hoist drum size and headframe sheave size can be reduced; smaller diameter rope requiring smaller drums and sheaves. Also, smaller diameter rope is less expensive. Operational power costs will be reduced since some load due to rope weight is reduced. Smaller diameter drums and sheaves operate at reduced torque and this contributes toward reduced operating power required. Furthermore, rope installation and replacement are easier with smaller ropes.

IX. THE SELECTION OF SKIP MATERIALS

The materials selection process for skip-guide systems must consider strength, abrasion resistance, and corrosion resistance properties of materials. In addition, material cost and fabrication requirements must be considered, since these can significantly affect the feasibility of a material for use in a skip-guide system.

This section is broken into three main parts and discusses materials with good strength, abrasion resistance, and corrosion resistance properties. The fabrication and cost of many of these materials is emphasized.

A. FOR STRENGTH

Increasing the payload of transportation equipment is a constant concern of the design engineer. For skips, the ratio of dead weight of the skip to the skip muck capacity is defined as the payload ratio and is a measure of the skip's efficiency. The lower the payload ratio, the more efficient the skip. To achieve a low payload ratio, materials with high strength to weight ratios are required.

1. High Strength Steels

High strength steels are listed in Table B -2. They have a higher strength to weight ratio than structural plain carbon steel; in fact, some high strength steels have strength to weight ratios equal to aluminum. The relative costs of various steels compared to AISI 1020 are given in Table IX-1 (Refs. IX-1,2). It can be seen that the costs of the high strength steels listed range from slightly more to three times that for structural carbon steel (A 36). When considering the savings effected on hoisting capital and operational costs due to reducing the payload ratio (see section VIII-B), the increase in skip material cost appears to be somewhat trivial. In general, "higher initial costs for high strength steels on mobile equipment may be justified several times over during total service life" (Ref. IX-3). Furthermore, fabrication of these high strength steels can usually be done using standard equipment found in most fabrication shops.

High strength steels are generally weldable by any of the arc welding processes used for structural steel. E60- and E70-class electrodes are commonly used. Low-hydrogen welding techniques become necessary for the higher strength steels (heat treated); thus, low hydrogen rods, preheating, and post-weld stress relieving are common in

Table IX-1 Relative Cost of Plate Steels (AISI 1020 taken as 100)

STEEL	RELATIVE COST
AISI 1020	100
AISI 1045	122
AISI 4130	240
A 36	102
USS EX-TEN 50	119
USS EX-TEN 60	128
USS COR-TEN A	140
USS T-1 REG	260
USS T-1 A	240
USS T-1 B	270
USS AR	126
USS AR 350	190
USS T-1 A 321	240
JALLOY AR 360	225
302 SS	546
304 SS	546
316 SS	755
430 SS	483

welding procedures. When maximum joint efficiency, ductibility, and toughness are desired, weld heat input must be carefully controlled. Submerged-arc and gas-shielded welding are not suitable for these higher strength alloys when they are less than 1/2 inch thick. Resistance welding is recognized for most high strength steels. Cutting is done by either gas or carbon-arc. With the hardenable steels, pre- or post-weld heating may be necessary to reduce edge hardness resulting from mass quench effect.

Forming of high strength steels can be done with the same equipment as that used for plain carbon steels (Ref. IX-4); however, more power, greater bend radii, increased die clearance, and greater allowance for spring-back may be required. Minimum bend radii as a function of material thickness, t, for the more formable grades are given in Table IX-2 for thicknesses to 1/2 inch (Ref. IX-3):

Table IX - 2 Minimum Bend Radii

Thickness, in.	Minimum Yield Strength, psi		
	45,000	50,000	100,000
To 1/16	1/2 t	1 t	2 t
1/16 to 1/4	1 t	2 t	2 t
1/4 to 1/2	2 t	3 t	2 t

For thicknesses greater than 1/2 inch, hot forming is recommended. For the quenched and tempered grades, hot forming at 1600 to 1800°F must be followed by quenching and tempering to restore the high strength properties.

Other fabrication operations such as machining, punching, and shearing are very similar to that for plain carbon steel. Although machinability is slightly more difficult, the higher strength material yields superior surface finish and better resistance to tearing when cutting threads. Shearing up to one-inch and punching up to 1/2-inch plates is possible (Ref. IX-3).

2. Shafting Steels

Shafting steels are normally used for structural purposes; hence, a high strength to weight ratio is desired. Shafting steels supplied are generally cold finished either by drawing or turning. This is done to achieve a smooth surface finish and close dimensional tolerances. In the case of cold drawn rounds, an increment of strength is imparted to the steel economically. Strengths of cold drawn bars are given in Table B-12. Cold drawn bars are not produced in diameters larger than six inches. For many high strength applications, alloy heat treated and cold finished rounds are selected. Cold finishing in these cases usually consists of turning. Strengths of 150-200 ksi can be achieved with AISI grades 4340, 4140, and 4150.

The main fabrication operation used on skip shafting steels is machining. The machinability of plain carbon cold drawn steels is better than that of hot rolled steels of similar chemistry. The added increment of strength from cold drawing results in chips that are crisper and break away easier. Reduced tool wear and power requirements are experienced. Plain carbon cold finished rounds that are supplied in just the turned condition have machinability ratings equal to hot rolled steel. Table B-13 gives machinability ratings of hot and cold finished carbon steel bars. When high strength steels are used for shafting, machinability becomes a problem and the strength used is usually limited by machinability.

Welding of shafting steel is done on skips; however, the cold finished properties in the heat-affected zone are destroyed. If welding is considered, careful attention must be given to pre- and post-weld heating of the shafting steel. Many steels require a stress relief treatment at 1200°F after welding, which destroys the cold drawn properties of a steel like 1040. Consequently, welding of shafting steels, though done in practice, should probably be avoided.

B. FOR ABRASION RESISTANCE

Abrasion resistant materials are normally used for skip liners. Two types of steels are used for the applications. Elastomers are also sometimes used. This section discusses AR and manganese steels as well as elastomers.

1. AR Steels

Abrasion resistant AR steels are given in Table B-2, Group IX. They are members of the high-strength low-alloy family of steels and have high hardness to resist abrasion. Three types of abrasion are generally accepted as existing in mining operations. They are sliding, grinding, and gouging. Grinding to gouging abrasion characterizes skip loading and dumping operations. When selecting an abrasion resistant material, the suitability of the material for the abrasion service should be considered.

Fabricability of AR steels is limited because of the high hardness that characterizes them. Welding, gas cutting, and forming are the most common fabrication operations performed with these steels. The hot rolled AR steels can be preheated and stress relieved before and after welding, respectively. The quenched and tempered AR steels should be preheated prior to welding and gas cutting but not stress relieved after welding. When weld bead hardness is not important, electrodes of the low-hydrogen E70 grades or austenitic grades may be used.

Where high weld bead hardness is important, electrodes of the low-hydrogen E90, E100, E110, and E120 grades should be used, depending on hardness desired. Cold forming of the AR steels is restricted to large bend radii, especially in the quenched and tempered grades, and only the hot rolled grades can be hot formed.

2. Manganese Steels

Manganese steels are used for abrasion resistance applications because they have the ability to work harden to 600 Brinell during service. Their impact resistance is so good that test specimens have often not separated in the course of an impact test. Three manganese steels are listed at the bottom of Table B-2, Group IX.

Fabrication of manganese steels is limited because of their high rate of work hardening and tendency to embrittle with heat. Welding of manganese steels should be done manually by metal arc. No pre- or post-weld heating should be done before or after welding or gas cutting. Best quality welds are achieved with austenitic stainless electrodes E309 or E310; however, best abrasion-resistant weld beads will be achieved when high manganese electrodes (E-FEMN-A) are used. Weld beads should be promptly peened after 6 - 9 inches of weld bead has been deposited. Forming is done best after being annealed by quenching in water from 1850°F. Minimum bend radius is 2t for thicknesses up to one inch. Machining of manganese steels is difficult but possible when carbide or cobalt high speed steel tools are used with a sulpher bearing lubricant. Finishing is best done by grinding. Manganese steels have the disadvantage of being non-magnetic and difficult to shear or punch. These properties can cause problems at mine sites as discussed in Section VI-B.

3. Elastomers

Elastomers have been used as skip linings and offer advantages over steel liners in the areas of reduced weight, lessened ore sticking and build-up, corrosion resistance, and installation and replacement. Furthermore, lining life of elastomers is longer than steel in some instances.

Elastomers for skip linings are usually available as either plate or rubber faced steel plate. Fastening is done with adhesive bonding or mechanical fasteners and rubber faced steel plate can be welded (Ref. IX-5). The Gates Rubber Company's "Handbook of Rubber Faced Steel Plate Fabrication" describes many fabrication operations (Ref. IX-5). Because of the lighter weight of elastomers (approximately 1/6 to 1/7 that of steel), handling is easier and safer during installation and replacement.

Some of the trade names for elastomers made by a number of companies that responded to inquiries are:

Trellex	Trelleborg Rubber Co., Inc.
Tivar-88	Poly-Hi, Inc.
Vulcobond	The Gates Rubber Company
Linatex	Linatex Corporation of America
Irathane	Kryptonics, Inc.
Estane-Polyurethane	B. F. Goodrich Chemical Co.
Elastocast	Acushnet Company
Adiprene	E. I. duPont deNemours & Co., Inc.
Polyarmor	Foamcraft, Inc.
Rubber Lining	Intermountain Rubber Industries
Orthane	Ohio Rubber
Benelex	Masonite Corporation
Gatorlite	Plastic Techniques, Inc.
Elastuff	United-Tech

C. FOR CORROSION RESISTANCE

It is generally accepted that the mechanism of corrosion of metals is an electrolytic process when attack depends on moisture and oxygen. The presence of an electrolyte tends to increase corrosion by virtue of increased conductivity. This explains why corrosion attack in salt and acid environments is more severe than in uncontaminated dry environments. The environment of a mine shaft is atmospheric, and corrosion behavior in a shaft appears to follow atmospheric type corrosion patterns. Figure IX-1 (Ref. IX-6) shows atmospheric corrosion data for three types of steel in five locations; one location is a mine shaft. It can be clearly seen that the curves for this environment are very similar to those of the other four atmospheric environments shown.

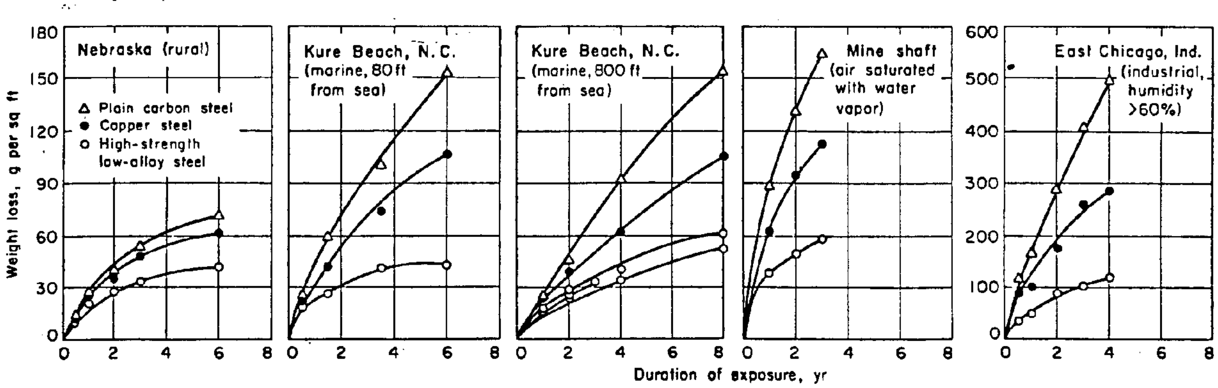


Figure IX-1 Corrosion Data for Steel in Five Locations

Composition of the steel at the five locations is given in Table IX-3.

Table IX-3 Composition of Steels Tested

Location	Steel	C	Mn	P	S	Si	Cu	Ni
Neb., Kure (80ft), Mine Shaft, E. Chicago	carbon	.07-.09	.30-.60		low		.03-.06	
	copper	.07-.09	.30-.60		low		.20-.25	
	Cu-Ni	.07-.10	.65-.75	.06-.08	low	.05-.10	1.05	.55-.65
Kure Beach (800ft)	1	.06	.36	.007			.08	.02
	2	.06	.35	.008			.31	.13
	3	.05	.37	.007			.91	1.65
	4	.16	.46	.073			.98	1.93
	5	.10	.82	.016			.42	.53

1. High Strength Steels

One of the major considerations during development of the high strength, low alloy steels (HSLA) was improved atmospheric corrosion resistance. The improved atmospheric corrosion resistance of HSLA over that of plain carbon structural steel is well documented. Relative values of atmospheric corrosion resistance for these steels is given in Table B-2 and range from one to eight. These steels have been used extensively in industrial environments where atmospheric corrosion is severe. They are often painted and display increased paint life. This has the effect of reducing maintenance costs.

2. Stainless Steels

The atmospheric corrosion resistance of most stainless steels is slight in uncontaminated environments despite variations in humidity. When industrial contaminants and chlorides are introduced, the corrosion resistance among the different grades is not so uniform and the selection of a suitable stainless steel requires evaluation and study. It has been determined that 201, 301, 302, 304, and 316 stainless steel offer good atmospheric corrosion resistance when exposed to contaminated industrial and marine environments. Type 430 has been used for transportation equipment and has performed well in many instances but in some cases has had to be replaced by the more atmospheric corrosion resistant grades given above (Ref. IX-7). It seems reasonable to compare mine shaft environments to contaminated industrial and marine environments because of the high corrosiveness that is often demonstrated. Consequently, one or more of the more corrosive resistant steels given above should probably be selected for mine shaft applications.

Aside from superior corrosion resistance, stainless steels have some disadvantages. Cost is approximately five to eight times that of A-36 although the yield strengths of these steels are similar to A-36. (Relative costs of 302, 304, 316, and 430 stainless steels are given in Table IX-1). No significant weight savings can be achieved by their use other than that saved due to less corrosion allowance.

3. Painting and Galvanizing

Painting is one way to improve corrosion resistance as well as appearance. By painting, corrosion is inhibited due to barrier protection of the painted surface. Paint life is therefore of interest. High strength steels provide superior paint life compared to plain carbon structural steel.

Galvanizing of steel not only provides barrier protection but galvanic protection as well. This means that protection is still imparted to bare steel surfaces as long as some zinc remains on the surface in the presence of an electrically conductive film of water. Shaft steelwork has been galvanized prior to installation in order to take advantage of this principle during service (Ref. IX-8,9). In addition, it has been shown that abrasion resistance of galvanic coatings is 15 to 45 times better than various paint systems (Ref. IX-9).

4. Non-metals

The use of elastomers, plastics, and fiber composites is attractive from a corrosion standpoint. These materials, not being conductors, will not corrode by the electrolytic action that characterizes metallic corrosion. They are finding more and more use in corrosive mine environments where their mechanical properties are satisfactory. One mine visited during this study was changing badly corroded metal electrical boxes to fiberglass because of fiberglass's superior corrosion resistance.

X. DESIGNING FOR THE SHAFT ENVIRONMENT

Analyzing the shaft environment can be helpful during the design phase and field modification of skips and guides. After a skip is in operation, there may be parts that show signs of distress caused by fatigue, corrosion, wear, or overstress. The mining environment can cause cracks, loss of material, or distortion. In all cases the severity of structural degradation increases with increasing stresses. Analysis can be used to decrease the stress levels, thereby reducing or eliminating the problem.

A new skip is expected to be trouble-free. It is after repeated exposure to the shaft environment and after many skipping cycles that skips begin to deteriorate. With proper concern for the survival of the structure, design decisions can be made to delay the aging process.

A. FATIGUE, CORROSION, AND WEAR

Design of a skip and guide system must give consideration to the rope pull plus the environmental factors that cause material degradation. Such factors are: dynamic loading; repeated loading; acidic, basic, neutral, or salt water; and abrasive, sliding, and gouging wear. In general, the environmental factors will appear in combination. When two or more of these act together, the combined effect is greater than the sum of the factors acting separately (synergism).

1. Fatigue

Fatigue, as a design consideration, has been considered since 1864 when Fairbairn, through experiment, recognized the existence of a limiting stress than can be applied safely an infinite number of times (Ref. IV-6). At one time it was thought that metals crystallized from repeated loadings. This does not happen. Fatigue causes a crack to develop and spread until fracture occurs. The fracture surface may appear crystalline due to the absence of plastic flow or because of course grain structure. Fatigue failures occur at stress levels considerably below the ultimate stress.

For complete stress reversal (full tension to full compression each cycle) the lowest stress level that will not cause failure under infinite cycling is called the endurance limit. This behavior is shown by the s-n diagram in Figure X-1. Values for the endurance limit are given in the tables in Appendix B.

A skip, by the nature of its role, is a structure undergoing constant cyclic stresses. The process of loading,

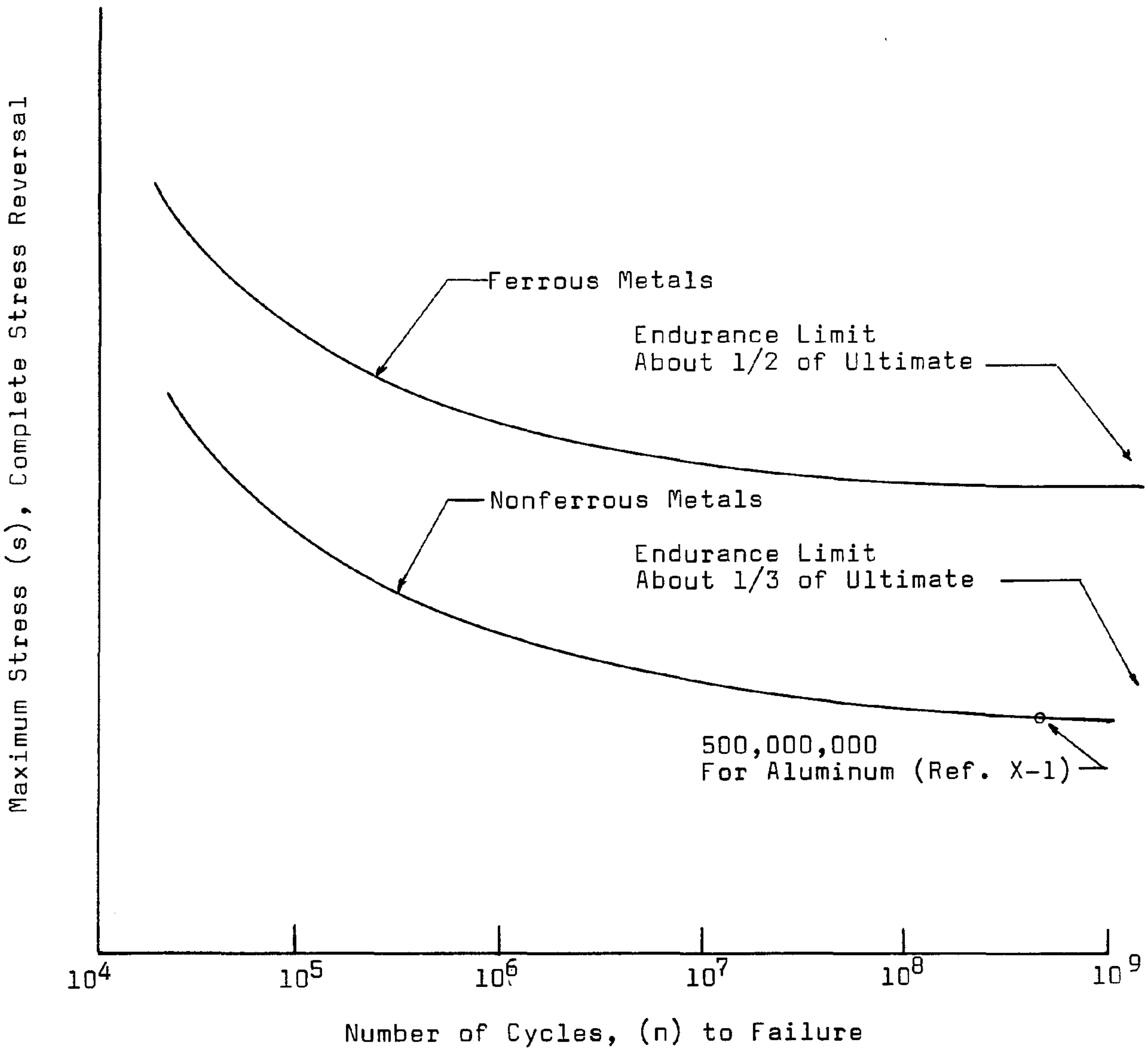


Figure X-1 s-n Diagram For Metals

hoisting, dumping, lowering, and reloading is a cyclic process that must be considered when designing.

During each cycle of hoisting and lowering additional stress cycles come from the skip-guide dynamic interaction. The bail and crosshead will undergo a series of stress cycles in the shaft while on the way up and on the way down. These additional cycles occur at those places where the guides are out of line. If there are ten out-of-line places in the shaft, that means the bail will receive at least twenty stress cycles (ten up and ten down) on each trip. There is some indication (Ref. V-6) that the empty skip on the way down will have higher loads between the skip and guide than the full skip on the way up.

In order to calculate the number of days needed to reach a million cycles, Figure X-2 shows a plot of days and hours of hoisting per day for various cycle times. If there are no bad spots, a bail will undergo a million cycles in 1500 days for 15 hours of hoisting per day with a 1 1/2 min cycle time. Assuming there are ten bad spots in the guides, the bail will reach a million stress cycles in one-twentieth the time (10 up and 10 down for each trip) or 75 days. Other examples are possible using Figure X-2. An indication of the dynamic interaction forces is given in Section V. One example is for a bunton 1/2 inch out of line resulting in a dynamic force greater than 4,000 pounds.

The AISC Steel Construction Manual (Ref. IV-4) established maximum allowable stress ranges for cyclic loading. The definition of stress range is illustrated in Figure X-3. Values vary from a range of 40 ksi for less than 100,000 cycles on the most simple structure to a range of 6 ksi for more complex construction.

An example of the most simple structures and more complex construction is shown in Figure X-4. Simple structures shown are symmetric members, symmetric bolted joints, and full penetration, ground and inspected welded joints. This type of structure has the highest allowable stress range. The stress range and maximum stress for full reversal is shown for 20,000 cycles to more than 2,000,000 cycles. Allowable stresses in welds for more complex construction is also shown in Figure X-4. The allowable stress range is considerably below the simple structure type. The AISC Steel Construction Manual gives detailed descriptions of general conditions, situations, and kinds of stress along with loading conditions and allowable stresses. The illustrations and examples shown here are a general indication of the design information that is available in the current literature.

Stress levels for ASTM A36 steel are shown in Figure X-5.

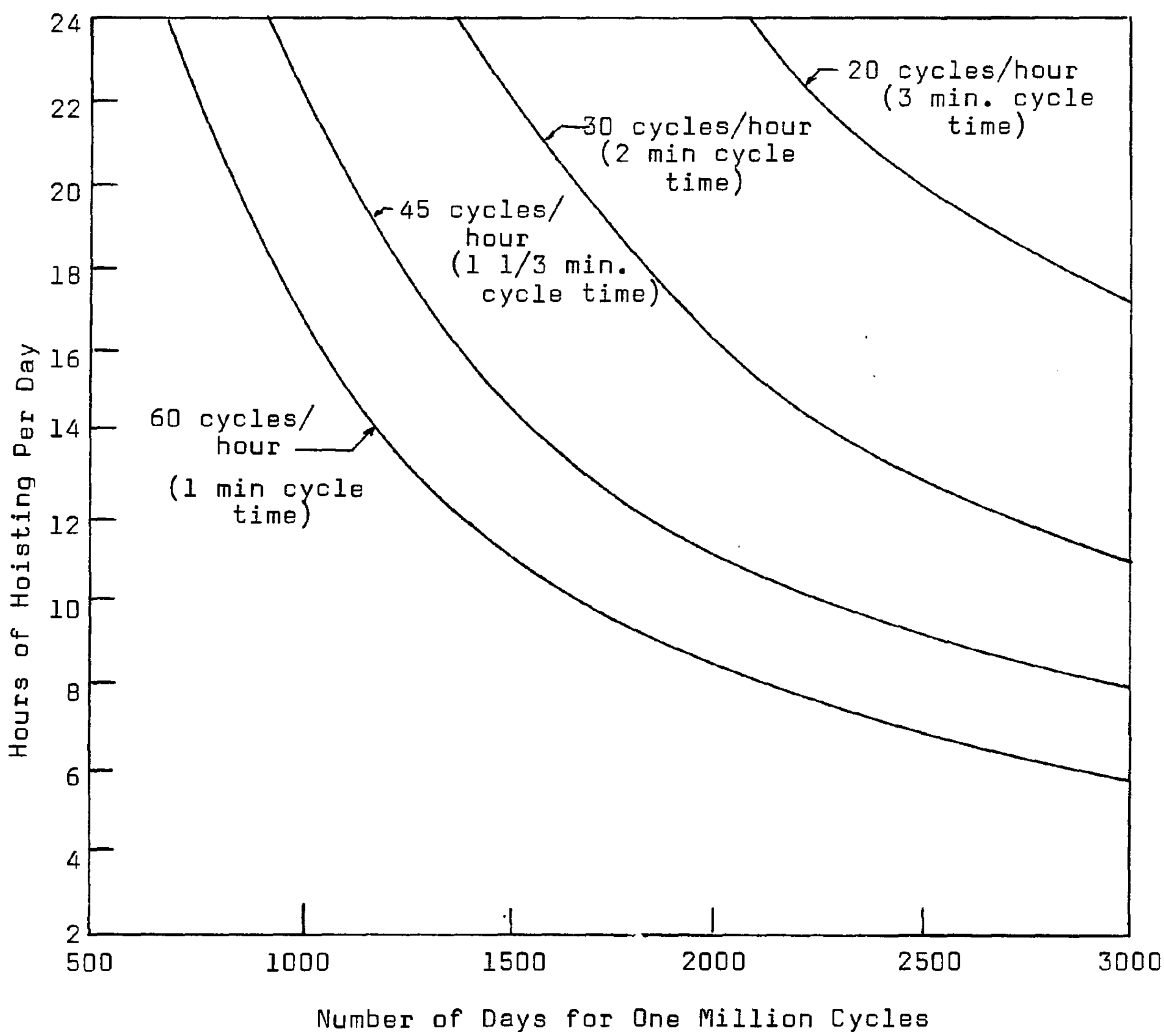


Figure X-2 Days Needed for a Million Cycles for Various Hours Per Day and Cycle Time

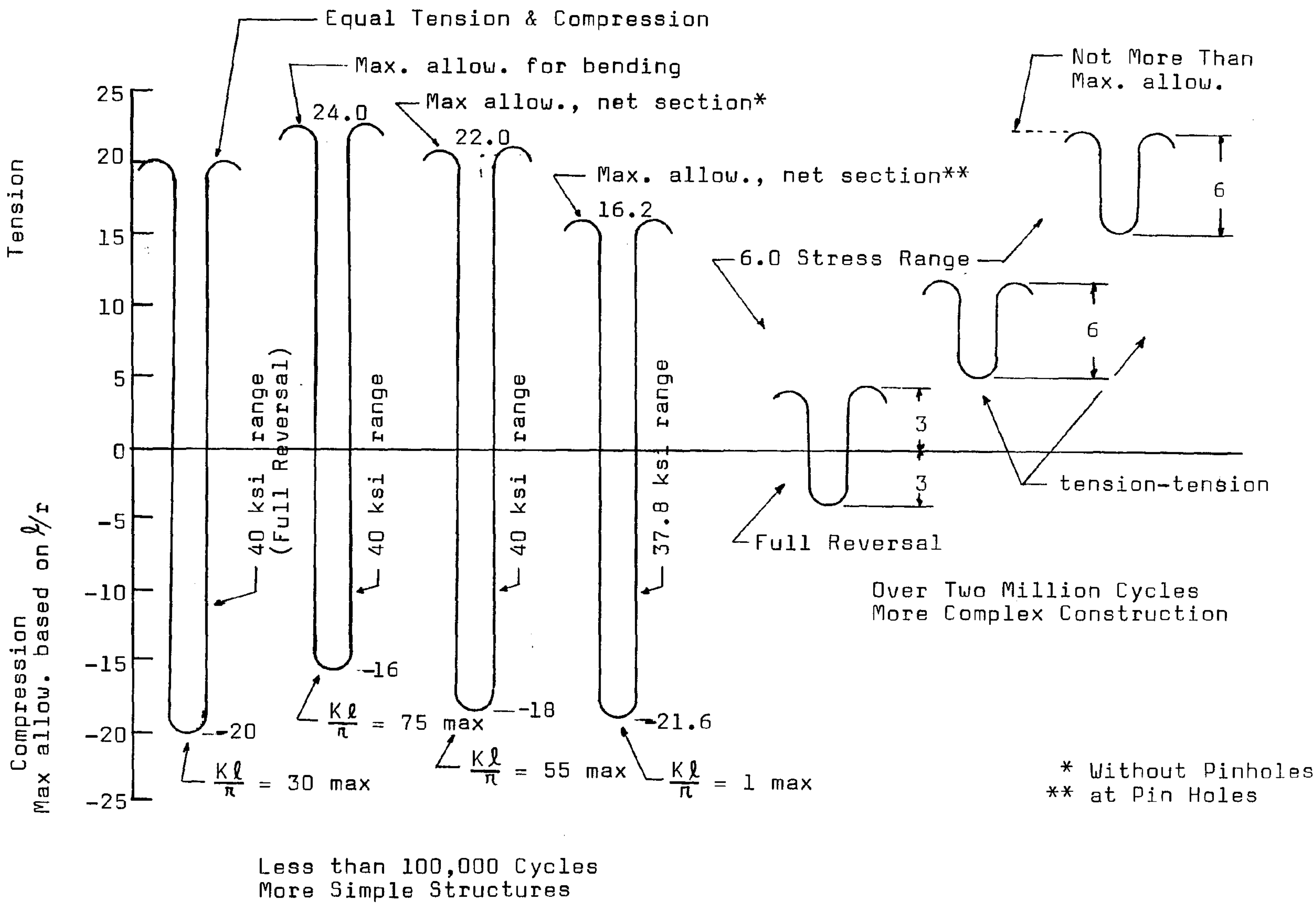


Figure X-3 Illustration of Stress Range, Full Reversal, and Tension-Tension

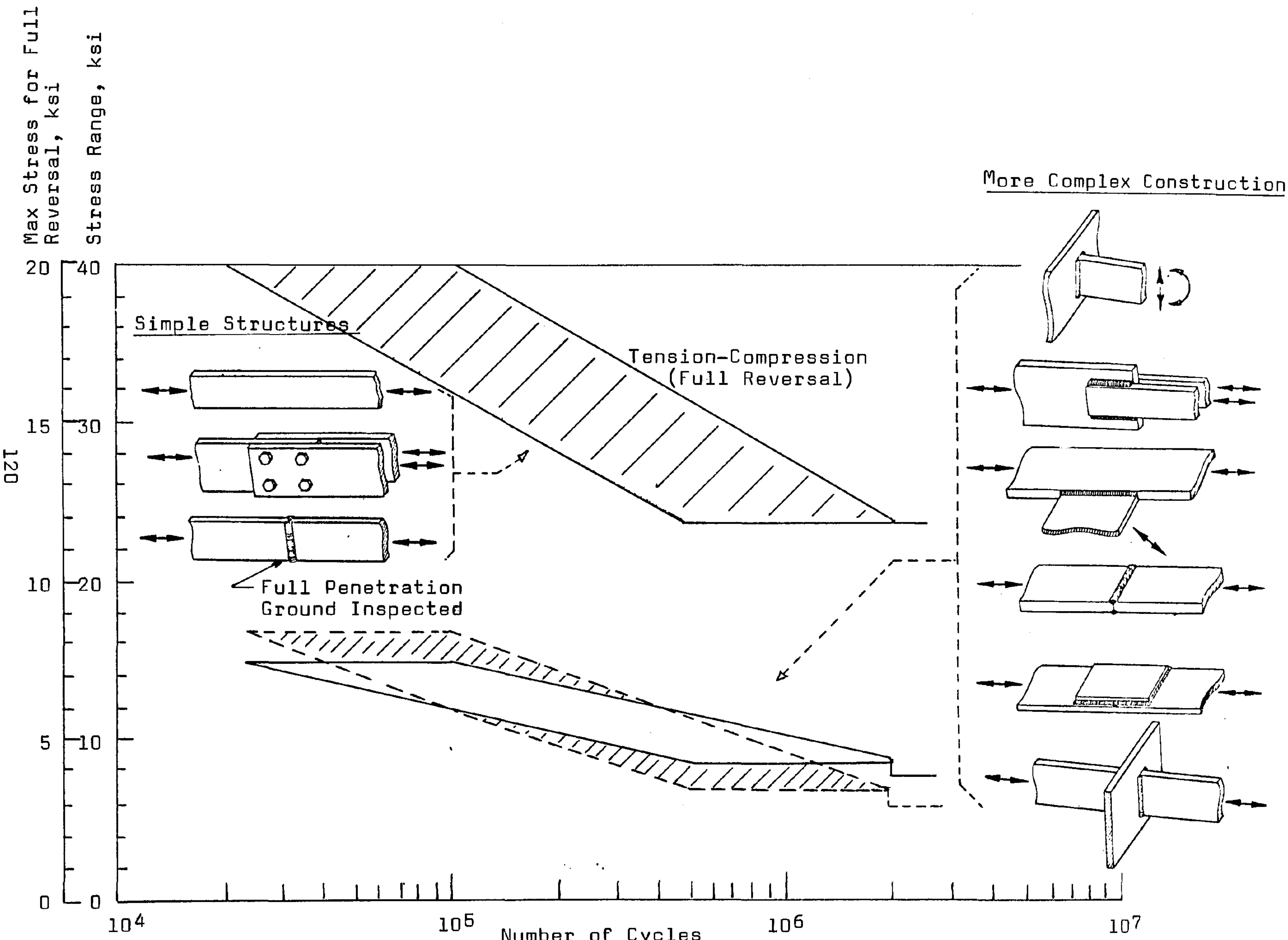


Figure X-4 Examples of Stress Allowables for Fatigue Loading From AISC Steel Construction Manual

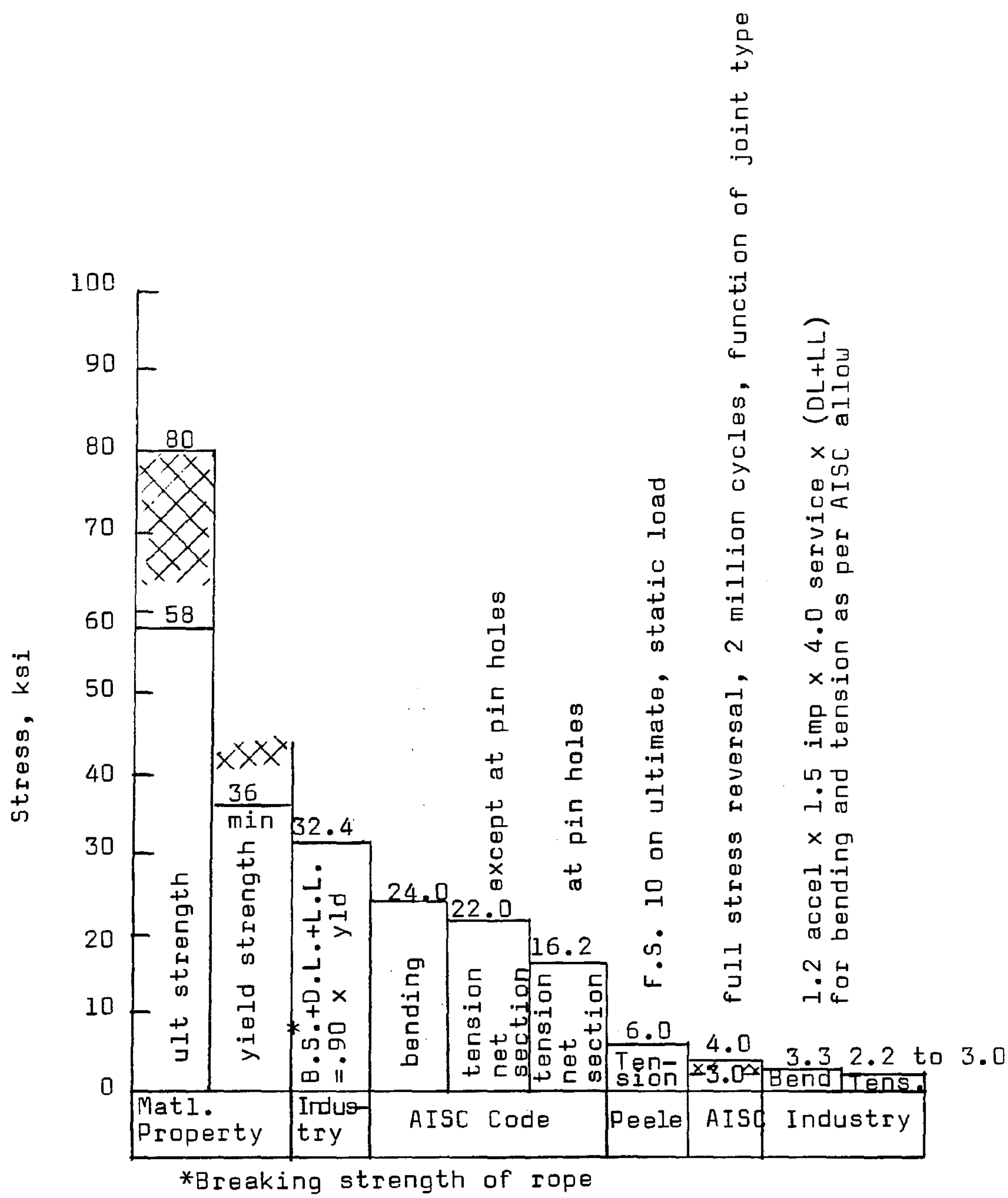


Figure X-5 Stress Levels for ASTM A36 Steel

Specifications for the ultimate strength and minimum yield strength are shown. Also shown is an industry approach to design for the breaking strength of the rope. The purpose of this design method is to ensure a usable structure in the event the rope pull should be great enough to break the rope. By keeping the stresses below the yield stress, the structure does not deform permanently, thereby retaining its original shape and remaining usable.

Some of the allowable stresses for ordinary steel construction are shown in Figure X-5 as per the AISC Code. Peele (Ref. IV-1) recommends a factor of safety of 10 on ultimate. Full stress reversal, for the worst conditions, as given by AISC, limit the tensile stresses to 3 to 4 ksi. Another industry approach, given in Section IV, is to use a factor of 7.2 on the loads and use the AISC allowable stresses for member selection. This has the advantage of providing a consistent design method when using high strength steels. This method also keeps the stresses within the allowable levels for fatigue loading.

A usual approach is to design critical members using both industry methods to select the more conservative design.

2. Corrosion

Corrosion has three effects on the skip material. One is the loss of material due to forming corrosion products; another is causing stress concentrations because of an irregular surface from pitting, and the third is fracture caused by stress-corrosion cracking.

The high strength steels have an inherent resistance to corrosion. The lower strength aluminum alloys are more corrosion resistant. Acid water can come in contact with the skip from wet muck and from the shaft making water. Guides, crosshead and bail will generally be affected by water in the shaft. The skip body will be in contact with the muck.

Surface protection can be obtained from several kinds of paint. Proper surface preparation prior to painting is important for good adhesion. In addition, keeping the paint from being worn off and maintaining it in proper repair can resist the action of corrosive waters.

A nonpaint method has been used to strengthen and protect skip bodies. This is the use of fiberglass reinforced mats bonded to the skip. Proper surface preparation is important to insure adhesion. There is evidence that the shaft environment can cause delamination of some application. The choice of resin systems should be left to specialists; however, the properties of the resins used as the matrix for fiberglass reinforced plastics are discussed generally (Ref. X-2).

Epoxy resins have high mechanical strength which is maintained under moist and humid conditions. Water absorption is very low. Cured resins are very resistant to caustic materials, many solvents, and most acids. The use of fillers and reinforcements can improve or degrade chemical resistance. Asbestos fiber is recommended when maximum chemical resistance is required. Epoxies, used in structural applications, have excellent adhesive properties.

Polyester resins are used in fiberglass -reinforced plastics. Areas of application include automobile, truck, and aircraft panels, storage tanks, and other components. Properties of thermoset polyesters are dependent on formulation. Resistance to weather and other environmental factors is generally good. Resin modifications can be made to improve specific properties as needed, by formulation.

Polymide resins retain excellent mechanical and physical properties at high temperatures (500°F). As a fiberglass-reinforced plastic they show good wear resistance and low coefficients of friction. Polymide parts are not affected by exposure to dilute acids. They are attacked by dilute alkalies and concentrated inorganic acids. Applications include jet-engine doors, piston rings for air compressors, bearings, oil well equipment, plus many other items.

In general, the use of a resin system for a fiberglass-reinforced plastic should be determined with those who are knowledgeable of specific formulations.

Attention to details in the design of a skip is of major importance in increasing corrosion resistance (Ref. X-3). Crevices are the most prominent design defect causing most equipment failures. Crevices cause localized corrosion by creating a chemical composition different from the material outside of the crevice. The different materials cause corrosion by galvanic action.

A difference in oxygen concentration causes corrosion. For the joint shown in Figure X-6 internal corrosion forms in the region of lower oxygen concentration. A difference in metal ion concentration causes external joint corrosion as shown in Figure X-7. Such crevices should be avoided or minimized. One solution to the problem of joint corrosion is to fill the crevice with a plastic or an elastomer. Another solution is to weld the joint shut.

When using fasteners in combination with welds, the engi-

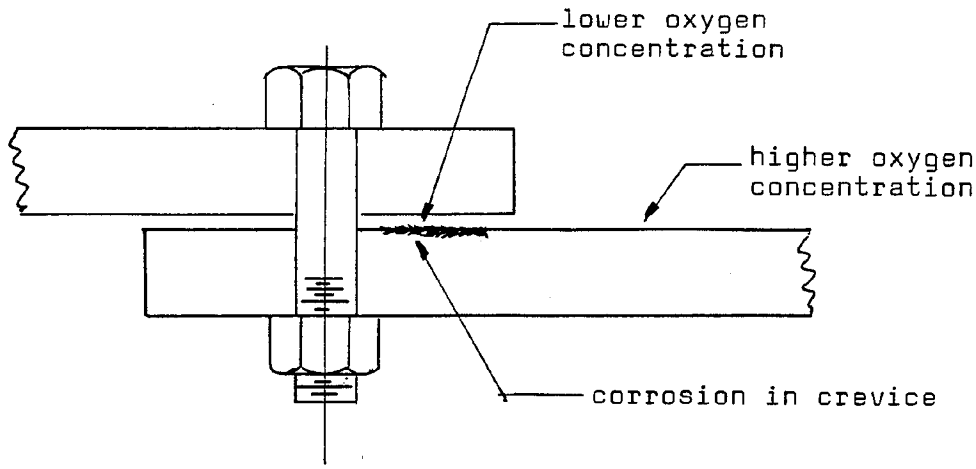


Figure X-6 Internal Joint Corrosion

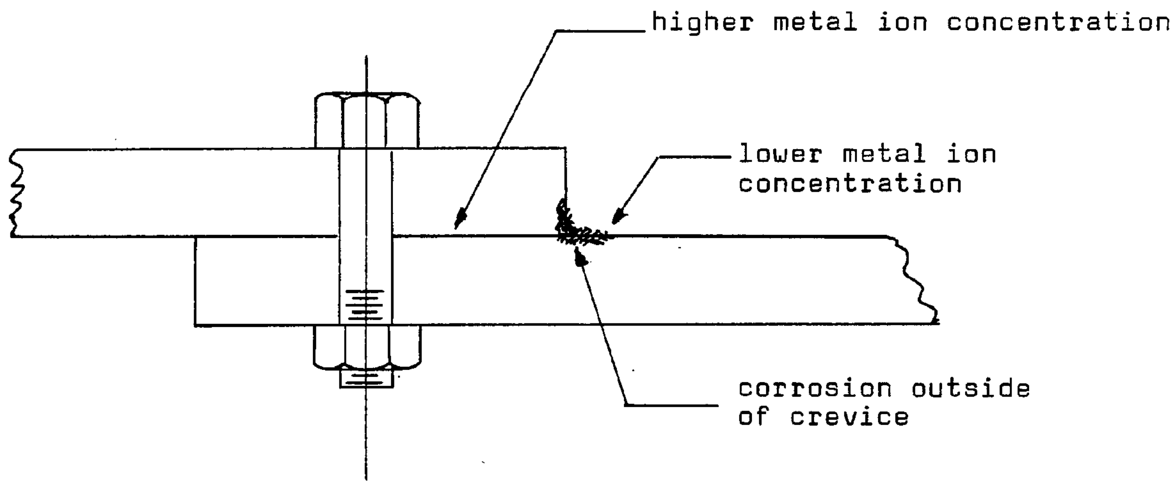


Figure X-7 External Joint Corrosion

neer and/or designer must decide whether to consider the fasteners acting alone or the weld acting alone or in combination. The AISC specifications for the design of structural steel for buildings makes the following statement:

> In new work, rivets, A307 bolts, or high strength bolts used in bearing-type connections, shall not be considered as sharing the stress in combination with welds. Welds, if used, shall be provided to carry the entire stress in the connection. High strength bolts installed in accordance with the provisions of Sect. 1.16.1 as a friction-type connection prior to welding may be considered as sharing the stress with the welds.
>
> In making welded alterations to structures, existing rivets and properly tightened high strength bolts may be utilized for carrying stresses resulting from existing dead loads, and the welding need be adequate only to carry all additional stress.

Welded joints can be made to reduce the corrosion caused by crevices. Figure X-8 illustrates good and poor details. The good details have fewer crevices or no crevices from which to form corrosion products. Figures X-8 a and b show a butt joint. The back-up ring commonly used in this type of joint forms a crevice. There are three ways to make full penetration welds without using back-up rings. These are: TIG welding with gas back-up, nonmetallic removable back-up, or consumable inserts such as the EB (for Electric Boat) or Grinnell Corporation insert.

In Figure X-8 b and c, the lap joint with fillet welds is shown. With both ends welded it is a fair joint. With one end welded it is a poor joint.

A usual construction is shown in Figure X-8 e and f. Where the piece is welded continuously, the result is a good joint. Where skip welds are used, a common practice, the joint has crevices and is considered a poor joint.

Similarly in Figure X-8 g and h where skip welds are used the resulting crevices make a poor joint for corrosion resistance. The continuous weld makes a good joint.

Stress-corrosion cracking is that condition which lowers the ultimate strength of a material when the material is under a steady state stress condition in a corrosive environment. Failure frequently is caused by simultaneous exposure to a seemingly mild chemical environment and to a tensile stress well below the ultimate strength (Ref.X-4). Several theories attempt to explain the mechanism. Two

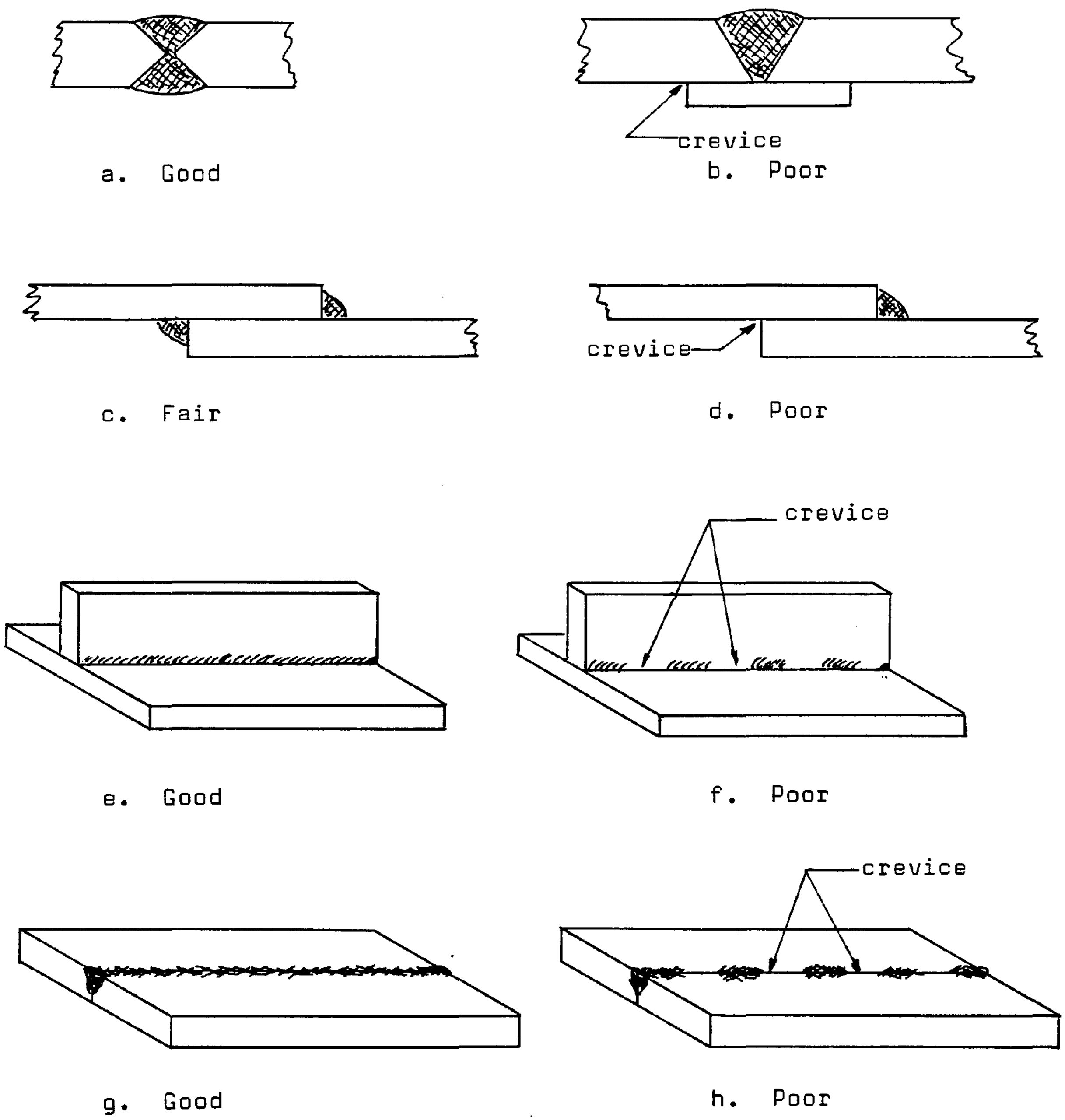


Figure X-8 Good and Poor Weld Details to Reduce Corrosion from Crevices theories are generally accepted. One is an electrochemical theory and the other is stress-sorption.

Stress-corrosion cracking occurs only under tensile stress which may be internal, as from locked-in stress, or external from load. The higher the stress level, the shorter the time required to produce cracking.

Grain direction affects cracking. Transverse stress is far more detrimental than longitudinal stress. Also, short transverse stress is more detrimental than long transverse stress. These conditions are shown in Figure X-9 below:

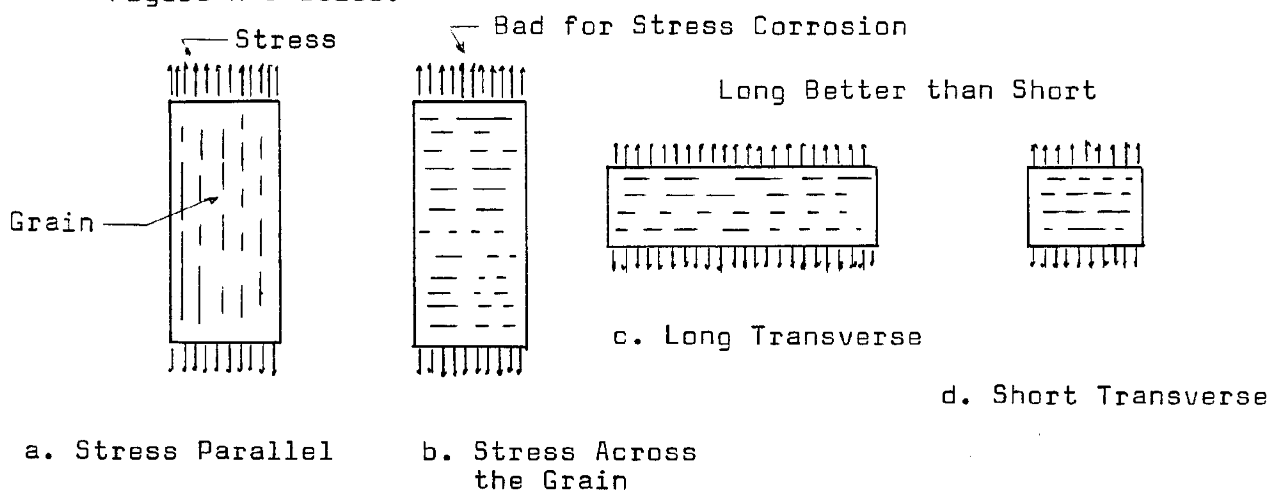


Figure X-9 Grain Direction and Stress Direction

During manufacturing, high local stresses can be caused by thermal processing, stress raisers, surface finish, fabrication, and assembly.

Welding is the most common thermal process to cause local stresses. Cooling of weld metal causes shrinkage which acts against the cool rigid metal adjacent to the weld. Tensile stresses as high as 30 to 40 ksi can remain as residual stresses. Other thermal process stresses occur from improper heat treating and quenching.

Stress raisers during manufacturing occur from notches, cracks from bending or heat treating, electric-arc strikes, and rough machining and grinding. Control over these items can reduce stress-corrosion cracking.

Fabrication processes such as bending can cause residual surface stresses as high as 30 to 60 ksi. Straightening pieces after heat treating can also cause residual stresses.

During assembly, locked-in tensile strength can be caused by improper fit, welded assemblies, shrink fits, and press fits.

The prevention of stress-corrosion cracking requires many considerations including the engineer, analyst, designer, and manufacturer. Keeping a skip free of stress-corrosion cracks is one means of increasing the useful life of skips.

3. Corrosion Fatigue

The combined action of fatigue under corrosive conditions is known as corrosion fatigue. In general, the fatigue life is reduced. Maintaining a corrosive environment generally will decrease the fatigue life of a material as shown is Figure X-10.

Mine waters were used as the corrosive medium for tests on four types of steel and one aluminum, as discussed in Reference X-5. At ten million cycles, the damage ratio (corrosion fatigue to fatigue strength) was found to be as given in Table X-1.

Table X-1 Ratio of Corrosion Fatigue to Fatigue Strength at Ten Million Cycles (from Ref. X-5)

Material	Average from Three Mine Waters
T-1 Steel	0.35
AR Steel	0.45
A-7 Steel	0.70
Stelcoloy-G	0.46
6061-T6 Al	0.72* (see footnote)

Although mine waters were used for these experiments, it has been established that fresh water has a similar effect on corrosion fatigue of carbon steels of varying carbon contents (Ref. IV-6). The presence of any water can be as important as the composition of the mine water as affecting corrosion fatigue.

Many factors influence failure from corrosion fatigue such as cyclic stress, frequency, stress amplitude, mean stress, stress-wave shape, and the environment. The mechanism is the initiation of cracks, crack growth, and finally failure. A design approach that considers these factors is needed to ensure a long lasting skip.

4. Wear

Wear occurs on the surface of a material whenever the material is displaced and removed. The main types of wear are:

* One mine water caused no loss in corrosion fatigue strength.

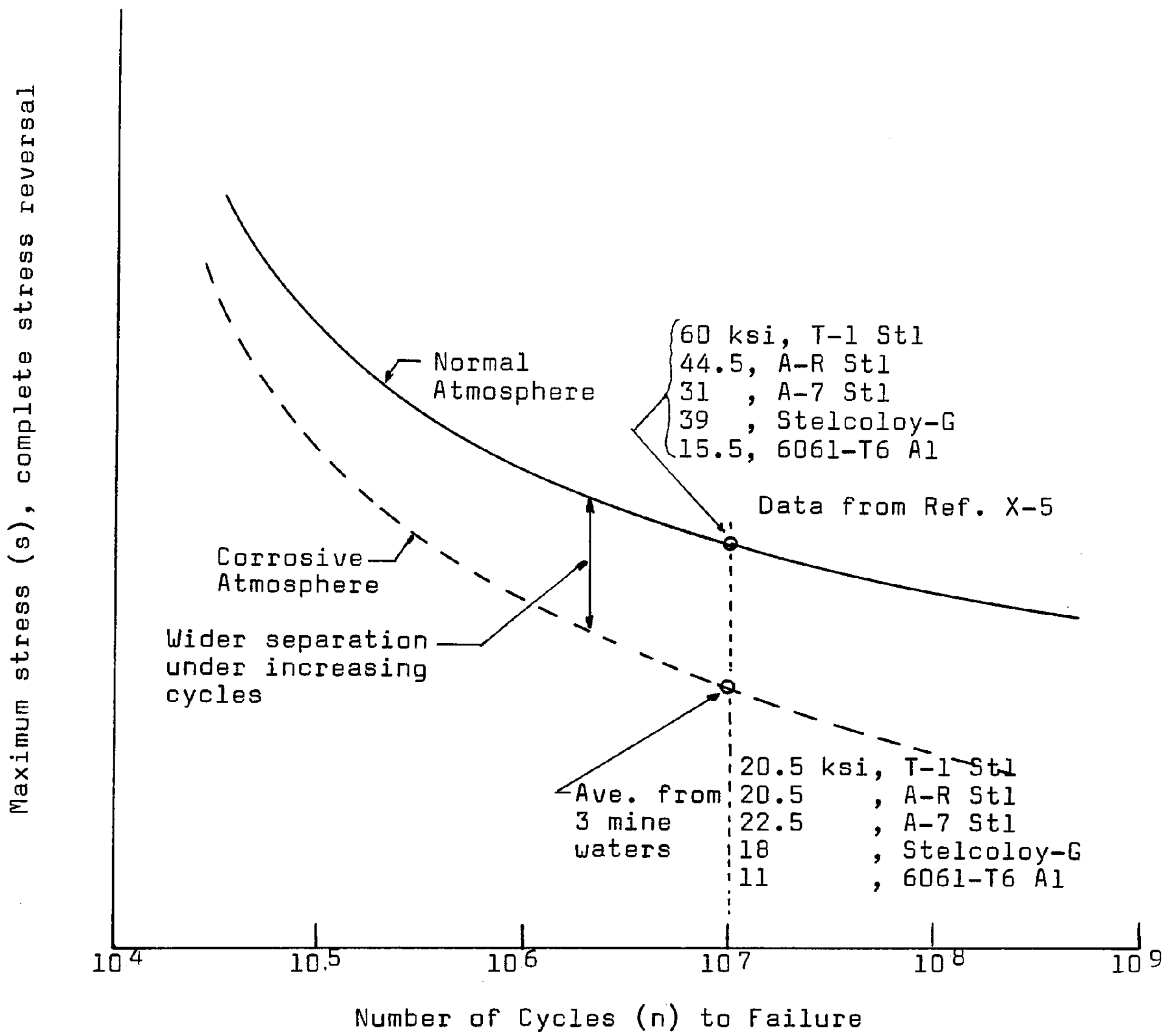


Figure X-10 s-n Diagram for Fatigue and Corrosion Fatigue adhesive wear, abrasive wear, erosive wear, corrosive wear, erosion-corrosion, and surface fatigue.

The predominant types of wear on a skip appear to be abrasive wear, the displacement of material by hard particles and corrosive wéar where corrosion products are removed by abrasion. Chemical reaction and abrasion may act to enhance each other (synergism).

Volume 10 of the Metals Handbook (Ref. X-4) devotes a chapter to wear failures. It is established there are few analytical tools to predict the amount of wear. There are, however, several generalities with regard to the abrasive wear of metals.

Gouging abrasion is the removal of large particles by rocks or sand sliding or rolling under pressure. The amount of material removed is directly proportional to the load and inversely proportional to the hardness of the softer material.

$$q \propto \frac{W}{p} \qquad \text{(X-1)}$$

where q = wear volume for unit length
W = applied load
p = hardness of softer material

Wear can be reduced by decreasing the load between the surfaces and/or increasing the hardness of the materials. This applies to skip bodies being filled and dumped and to the wear of guides and guide shoes.

Proper selection of guide rollers to keep guide shoes from rubbing the guides will decrease guide wear. The dynamic computer analysis, in Section V, was developed to select the guide roller systems that will keep the shoes off the guides. In addition, the guide rollers can be designed to decrease the dynamic forces between the skip and guides and still keep the shoes from rubbing.

Wear of skip bodies caused by loading and dumping is resisted by using abrasion-resistant liners. The common practice in the United States is to use abrasion-resistant steel alloys. Some United States mines are using abrasion-resistant rubber as liners.

Canadian (Ref. X-6) and South African (Ref. VIII-15) practice is to use rubber liners in areas of abrasion and impact and alloy steels for severe wear.

Another form of resisting abrasion is the use of cascade liners. These are shown as combination stiffeners and liners in this section.

See Section IX, "The Selection of Skip Materials", for details on the use of steels and elastomers for abrasion resistance.

B. STRUCTURAL ELEMENTS

A skip is made of a crosshead, bail, body, and door. These elements can be analyzed to find the magnitude and character of stress. The shaft environment causes fatigue, corrosion, wear either alone or in combination. Tensile stresses or full stress reversals enhance the degradation of materials. Therefore, having knowledge of the kind of and amount of stress can aid in design.

1. Crosshead

The crosshead can be represented as a simple beam loaded at the center, as shown in Figure X-11. Generally a pair of channels is used. Bending and shear stresses are greater for a full skip, less for an empty skip. The character of stress does not change. Tension remains tension and compression remains compression; however, the magnitudes increase and decrease.

Another load on the crosshead comes from guide reactions. This is a dynamic compressive load that can be as high as several thousand pounds. The dynamic computer program, SKIP, has been written to determine how large this force is for a given skip and shaft. The program can also be used to determine the guide straightness required to reduce this load to a given level. In addition, the program can be used to find the proper spring constant to keep guide forces below a specified value for a guide system that is not straight. See Section V for details.

2. Bail

The bail is made of a pair of members running from the crosshead downward to the door support. The bail provides: the pivot for swing-out body skips, the support for the door, the body stop, the attachment points for guide shoes, and the attachment for guide rollers. Stresses in the bail are bending, shear, and tension from the skip body applied statically and dynamically under repeated cyclic loading in a corrosive environment.

Loads on the bail from either a full or empty skip are shown in Figure X-12. Rope pull, pivot loads, body stop and door loads exist at the same time. Their magnitude changes for a full skip or empty skip and whether the skip is stopped or running. Also the loads change due to guide alinements and whether the skip is running with or without guide rollers. For guide rollers to be effective in reducing dynamic loads, they must be designed for the actual misalinement, the hoisting speed, the conveyance weight, the guide section and the bunton spacing.

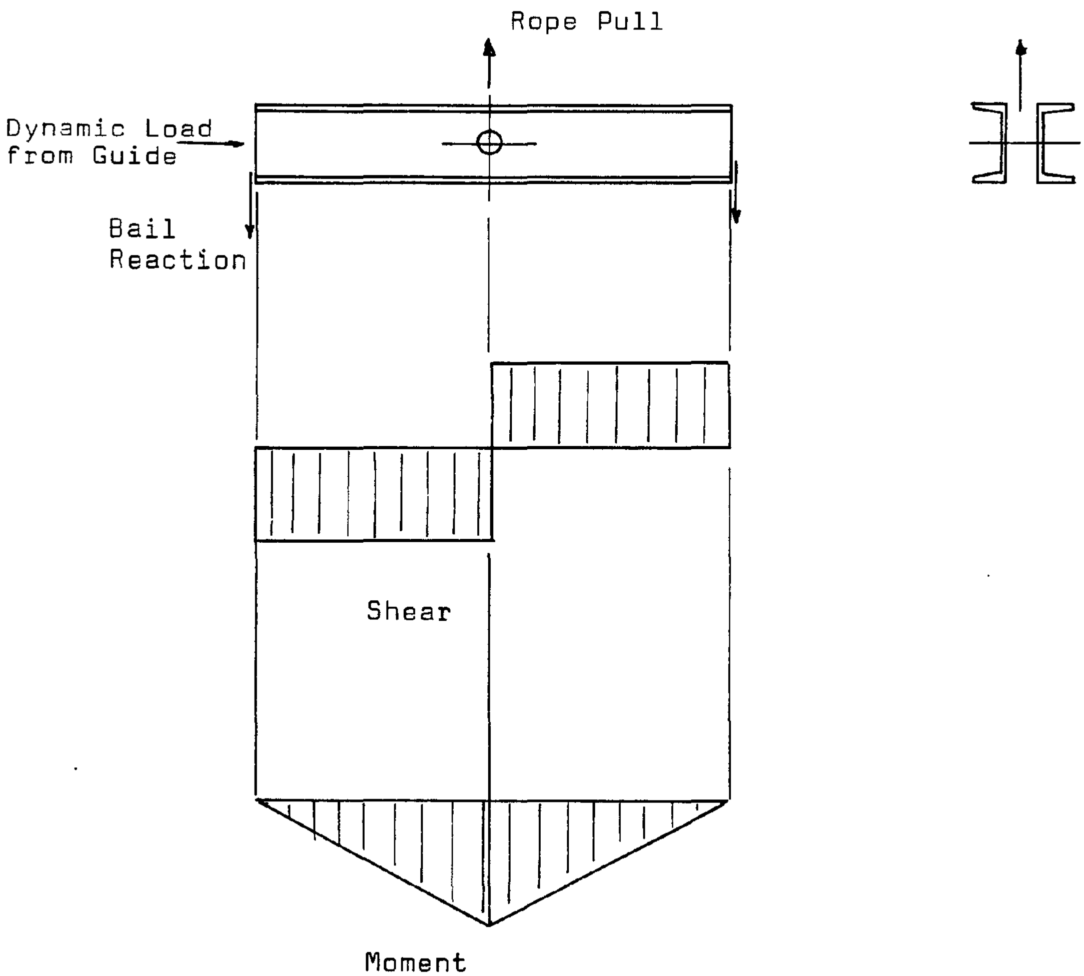


Figure X-11 Loads, Shears and Moments on Crosshead

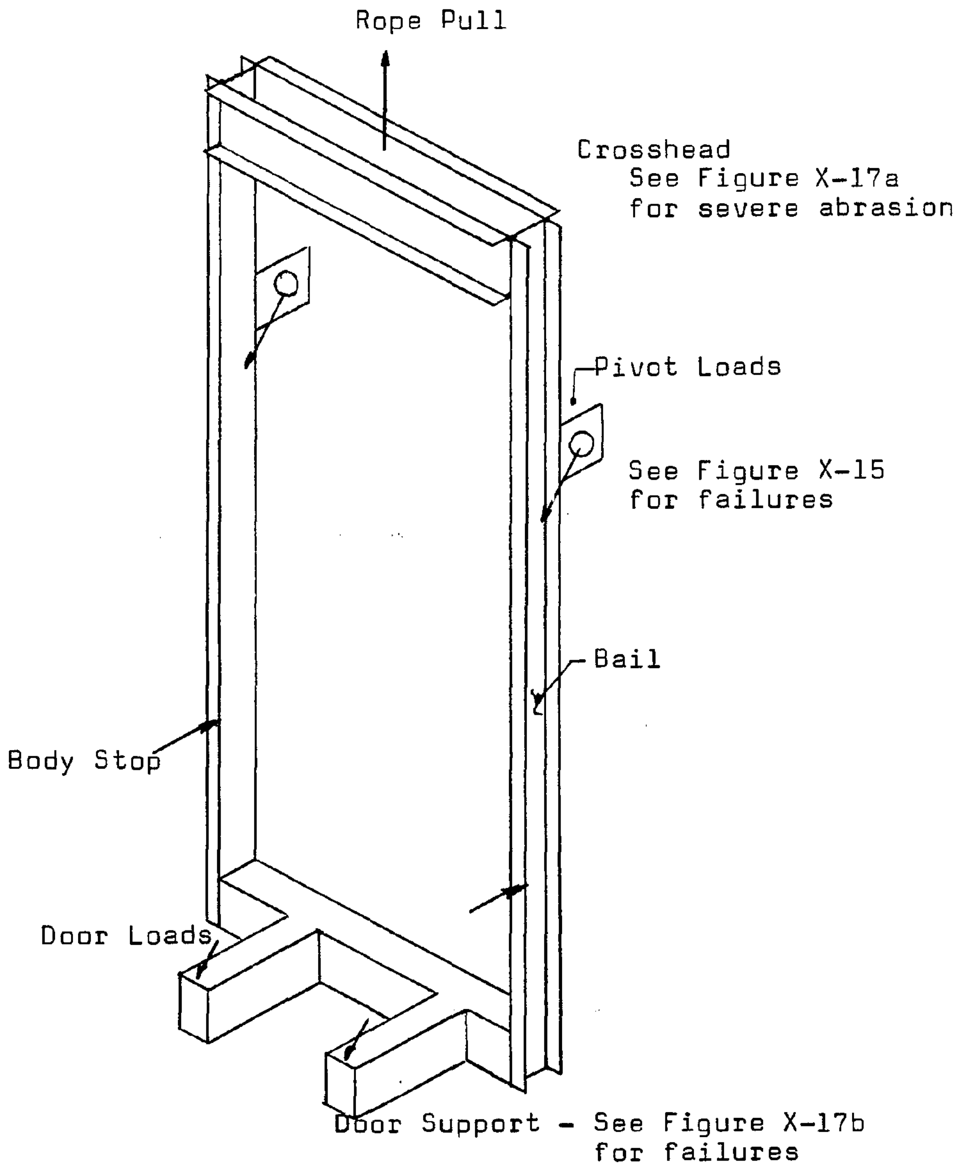


Figure X-12 Bail Loads Full or Empty Skip, Door Closed

Shear and moment diagrams for a full or empty skip with the door closed are shown in Figure X-13a. Quantitative values can be found from static equilibrium and dynamics. It is assumed the skip is balanced; therefore, it hangs vertically and does not rub against the guides.

Loads are also imposed on the skip bail during dumping. Body stop forces become zero while guide reactions are needed to restrain the bail. These are shown in Figure X-14. Values of these loads for a given skip depend on the position of the body in the swing and the amount of muck in the body.

Shears and moments in the bail during dumping are shown in Figure X13b. It should be noted that moment is plotted on the tension side of the bail.

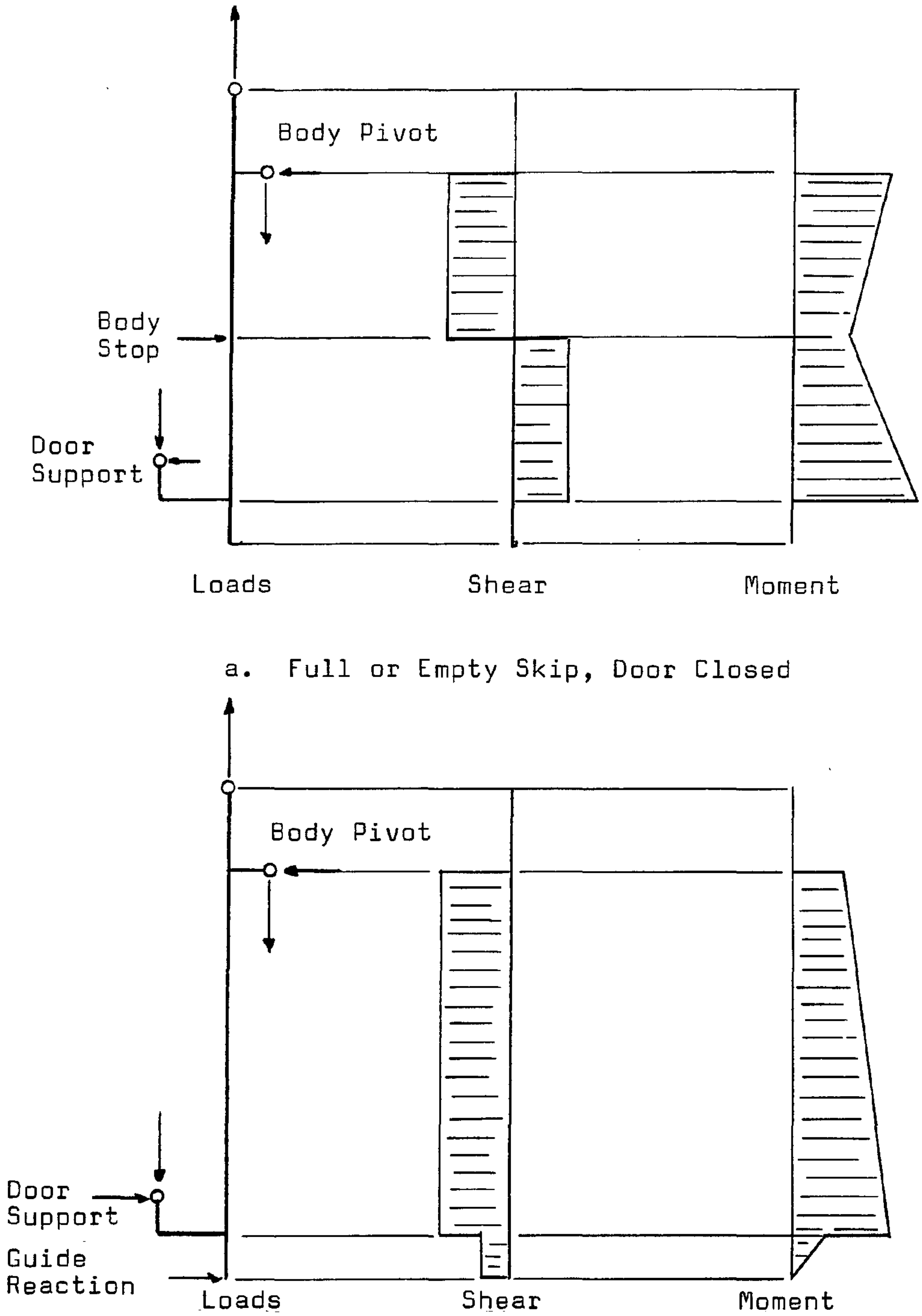
Experiences with bails have been recorded in the literature. Heather (Ref. IV-10) gives a good account of a bail that was modified three times to prevent cracking after cracks were discovered.

The original design is shown in Figure X-15a. The first failure occurred through a set of holes near the top of the channel section. The holes are located at a change in section. A pair of angles extends downward from the crosshead to a point just below the end of the bail. The bail has a different stiffness where these members join.

Several stress concentrations occur at the same place. Holes from riveted construction are stress raisers, as shown in Figure X-16a. A change in section in a tension member causes stress concentrations, illustrated by Figure X-16b. For a bending member stresses increase at a change in section, as demonstrated in Figure X-16b. These factors, acting together, are multiplied to find the highest stresses. Assuming each stress concentration to be about two, then the total factor becomes about six, a significant increase above a calculated nominal value. When coupled with a corrosive plus a fatigue environment, stress concentrations become very significant. A basic reference on stress concentration is Peterson, Reference X-7.

Further examination of the bail in Figure X-15 shows a crack following the areas adjacent to the change in section. In Figure X-15b, the angles from the crosshead were made heavier and were extended downward. A crack occurred at the end of the new angles, a change in section.

Figure X-15c shows a third crack, this one occurring in the flange of the channel; again reinforcing was added (two long angles along the bail) and a crack occurred at the end of the reinforcing. The second change in the design made a crack appear lower on the bail.



a. Full or Empty Skip, Door Closed

b. During Dumping

Figure X-13 Loads, Shears and Moment on Bail

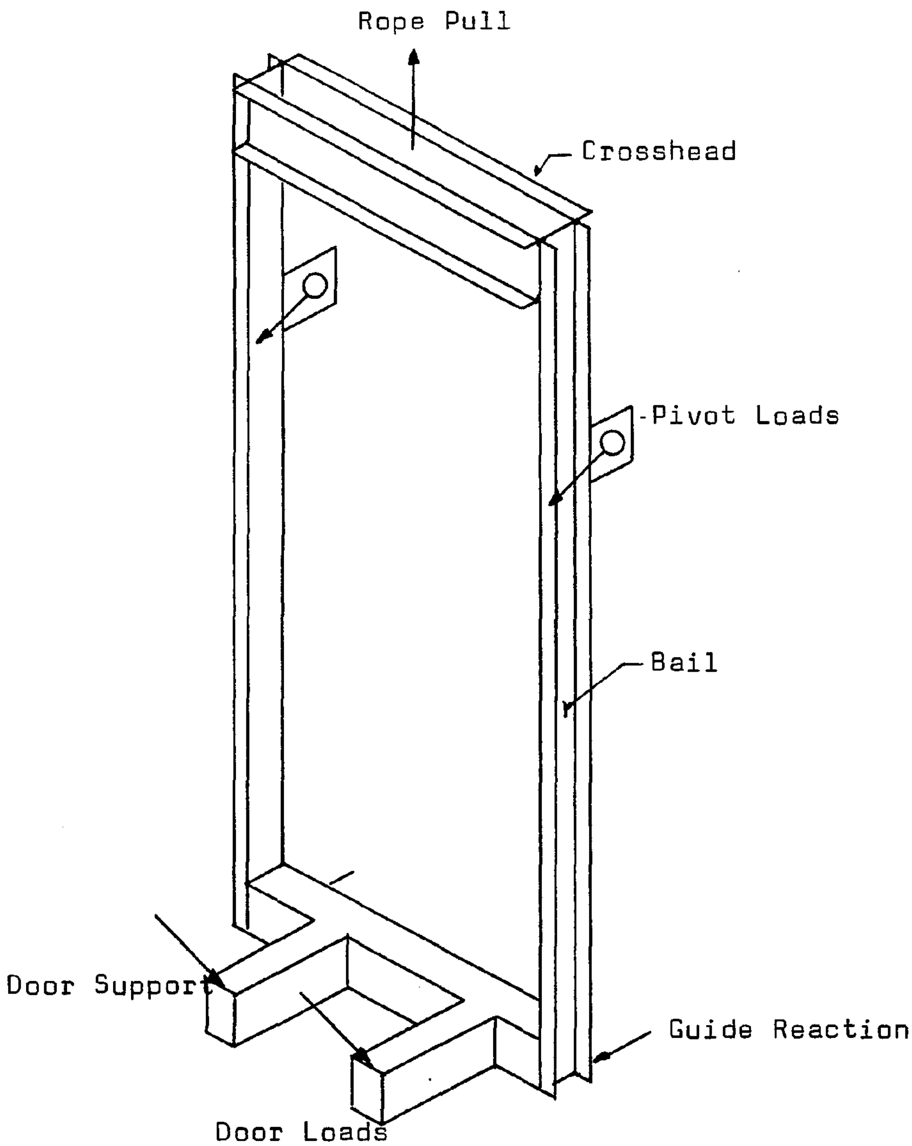
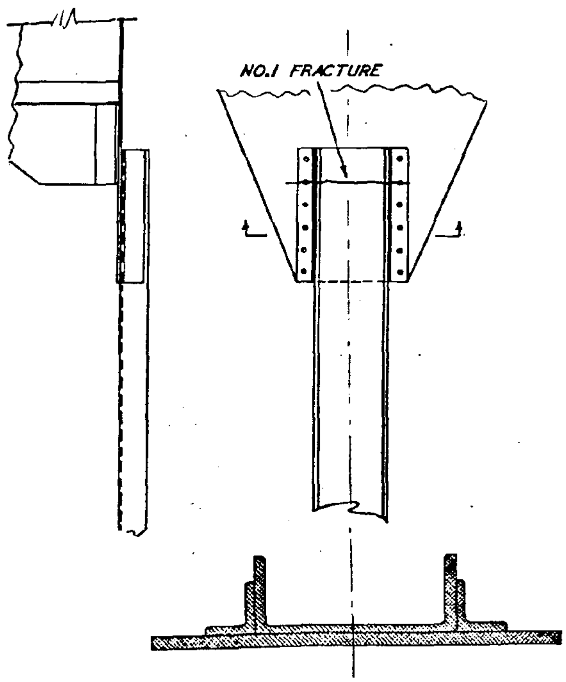
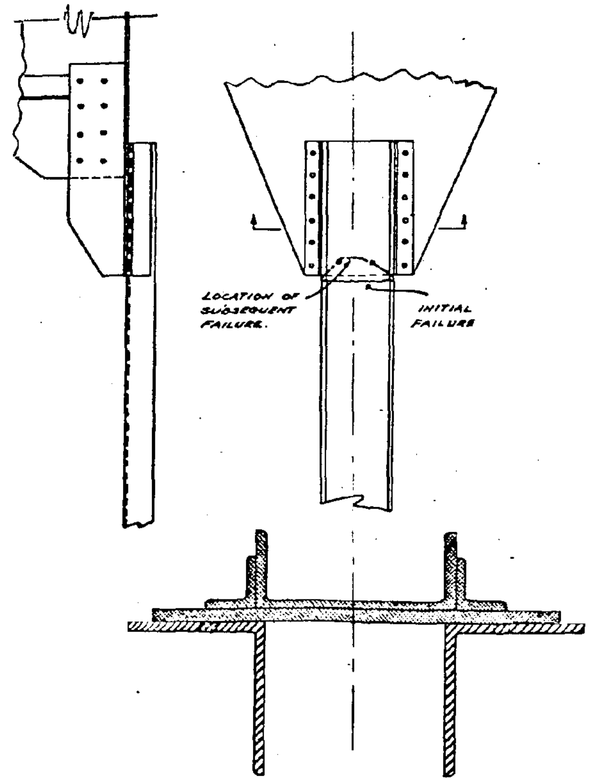


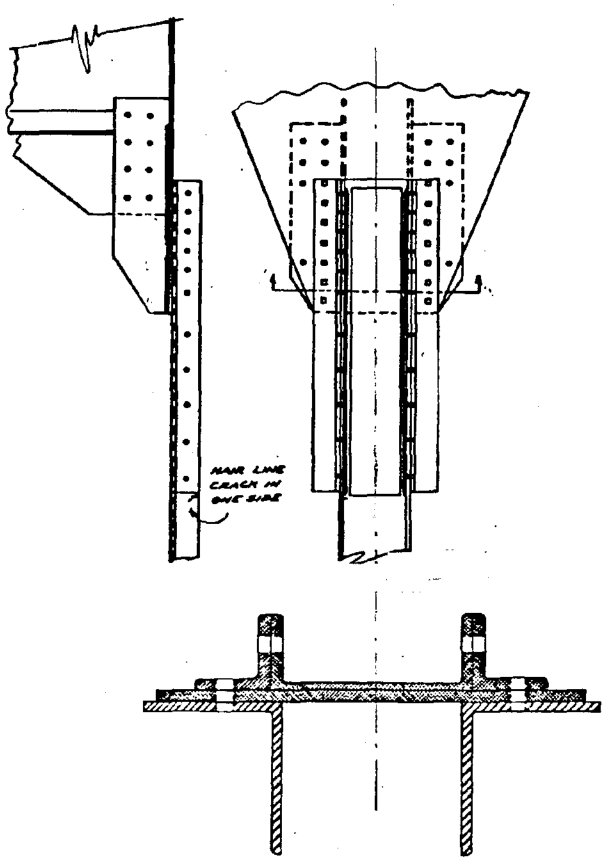
Figure X-14   Bail Loads During Dumping



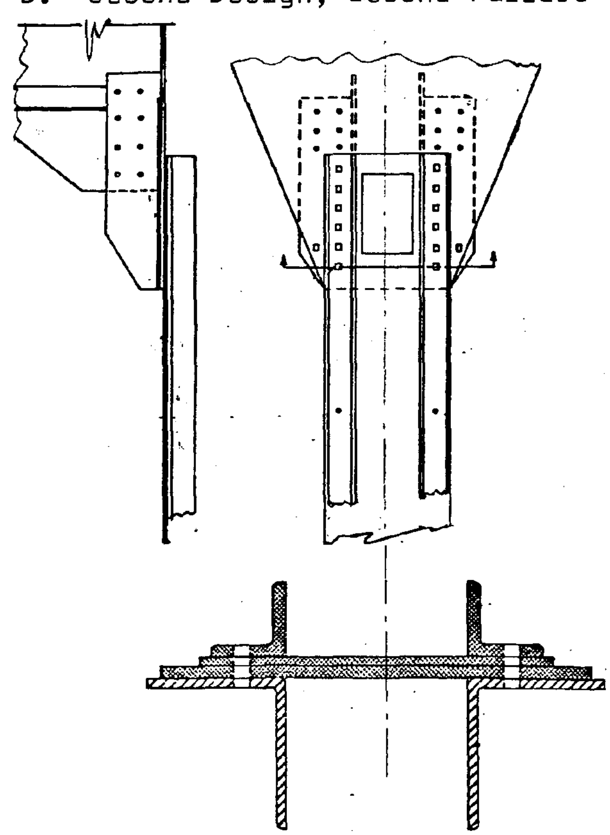
a. Original Design, First Failure



b. Second Design, Second Failure

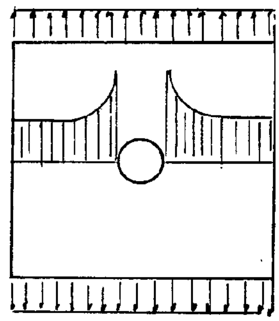


c. Third Design, Third Failure

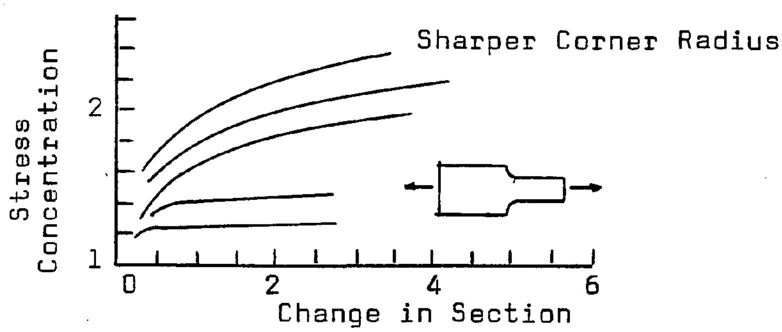


d. Final Design

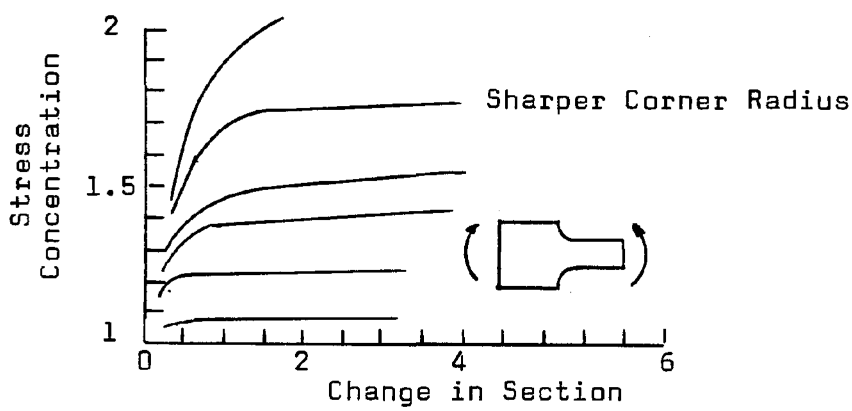
Figure X-15 Cracks in Bails at Madsen



a. Due to Holes



b. Due to Tension at Change of Section



c. Due to Bending at Change of Section

Figure X-16 Stress Concentration Factors

The final design, shown in Figure X-15d appears to have eliminated the change in section by providing long continuous members.

Examples of problem areas on the crosshead and on the door support bracket are shown in Figure X-17. In the upper figure, the flanges of the channels that form the crosshead have been severely worn away by muck striking the channels during loading. A straight edge shows the amount of distortion as well as the abrasion.

Failure of the door support bracket is pictured in Figure X-17b. The door loads were sufficient to cause the steel tube to tear open at the edge of the stiffeners. In retrospect, it would have been more appropriate for the stiffeners to be on the compression side, thereby eliminating the stress concentrations on the tension side.

3. Body

To keep the body in static equilibrium with the door closed, the forces acting are shown in Figure X-18. The external forces acting on the door and body can be found from the laws of statics: sum of horizontal and vertical forces equal zero, and the moments equal zero. Design of the bearing for the body pivot requires the body pivot load. When the skip is full, these forces will have their maximum values. Acceleration will add to the body, muck and door weights; and lateral forces from guide misalinement will periodically add to and subtract from the body stop force. The body pivot is located off center so that gravity keeps the skip closed.

During dumping the magnitude and direction of the external forces change. This is shown in Figure X-19. The same laws apply to determine the values of the external loads acting on the body. One design procedure, conservative but realistic, is to swing the body out to its maximum position and assume all the muck is still in the skip. This will produce the highest values of external loads. Such a condition can be found when the muck becomes frozen or when large rock forms a bridge and prevents muck from discharging.

A means of stiffening the body and providing liners for abrasion resistance is shown in Figure X-20. Cascade liners are formed of angles with their toes welded to the body, providing stiffening. On the forty-five degree surface, a replaceable, abrasion-resistant liner is fastened. Low weight, simplicity, plus internal stiffening (providing a larger cross section for a given clearance) are some of the advantages of using cascade liners.

a. Severe Abrasion on Crosshead

b. Failure of Door Support

Figure X-17 Examples of Problem Areas

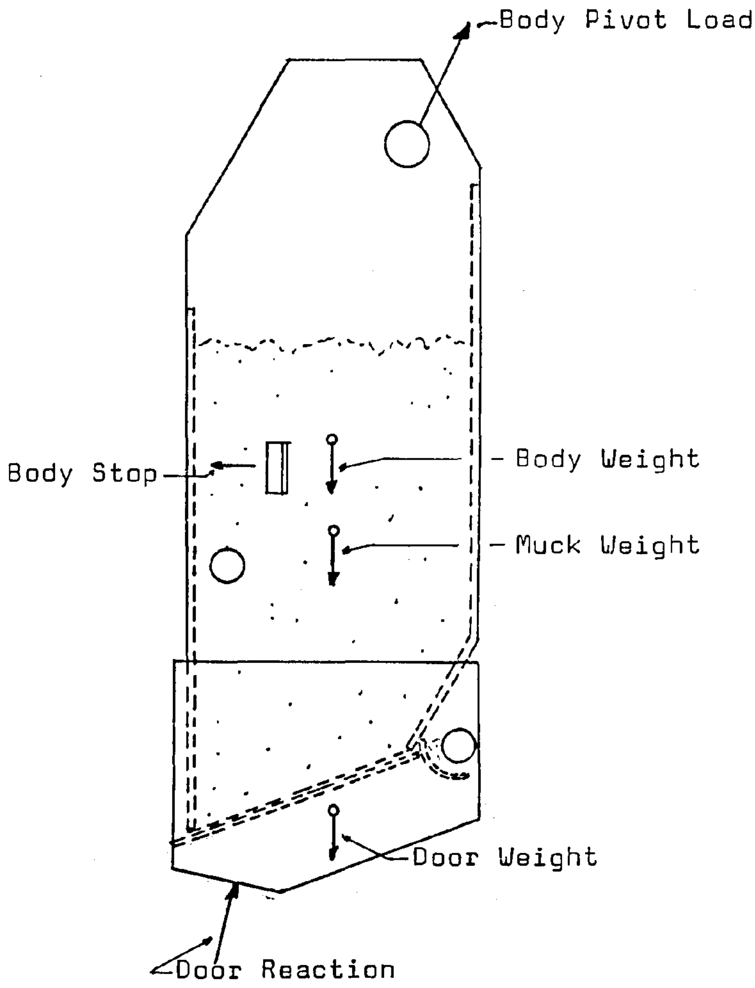


Figure X-18   Loads on and from Skip Body, Door Closed

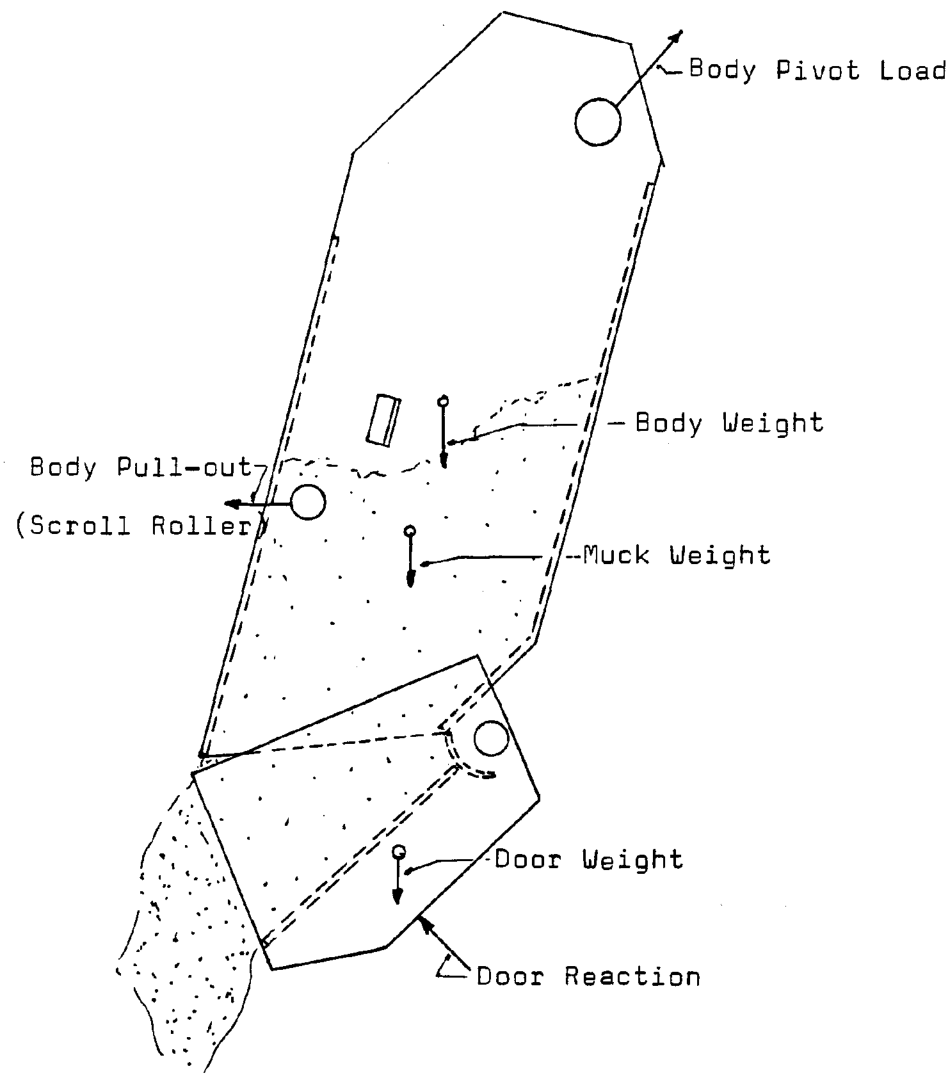


Figure X-19 Loads on and from Skip Body During Dumping

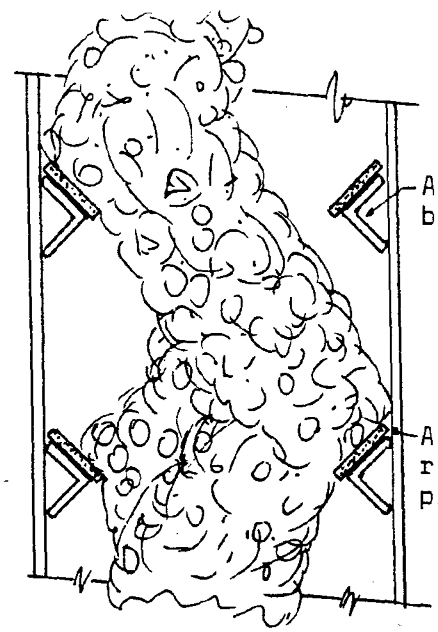
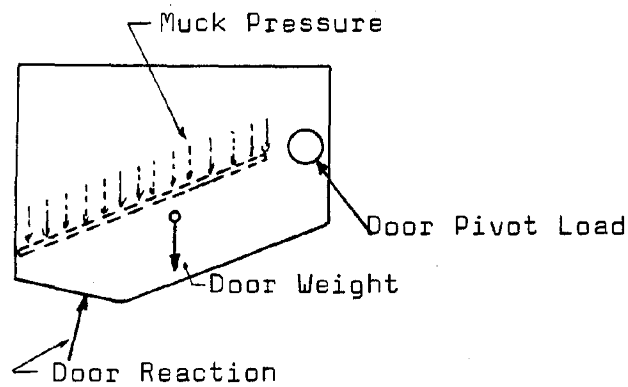


Figure X-20 Cascade Liners

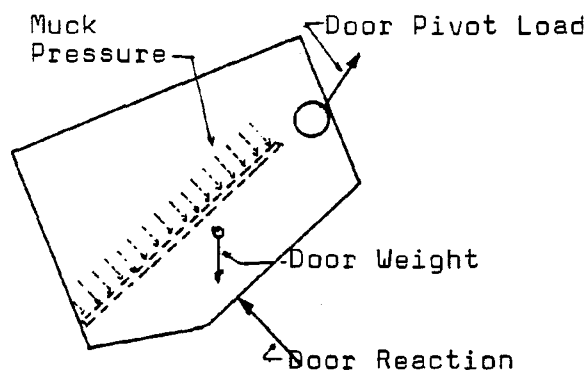
4. Door

In addition to the muck weight, the door must withstand the shock of material being dropped into the skip from a loading pocket. Figure X-21 shows the loads and reactions for a closed door and for an open door. Direction and magnitude of the door reaction and door pivot load can be found from the balance of forces and moments. Filling the skip will cause dynamic forces on the door. These can be found from Section IV for muck impinging on a surface. Hoisting accelerations also add to the door loads.

During dumping, a reasonable, conservative approach is to assume the skip full of muck with the door completely open. This will produce the highest values for loads and reactions in Figure X-21b. The door reaction loads are needed to design the door support. If the value of the load is not considered, distress, as shown in Figure X-17b may be the result. Also, bearing and roller design require the values of the door pivot load and door reaction.



a. Door Closed



b. Door Open

Figure X-21 Loads and Reactions on Door

Problems associated with the door are shown in Figures X-22 and 23. The nutcracker action in Figure X-22 can happen for two reasons. One is a very small clearance between the body plate and the door whereby muck buildup on the door will prevent the door from closing. The other reason for the nutcracker action is too small a dump angle on the door whereby the door does not clean properly. Even though the clearance between the body and door may be adequate, the muck remaining at the back of the door may jam between the body and the door. Very large forces applied to the door may result in distorting the skip and still not close the door. The dump angle must be increased to clean the skip.

As shown in Figure X-23, muck may be forced between the liner and the door and between the liner and the body. Although the dump angle may be sufficient to clean the door, fines may remain on the edge of the door liner shown by Figure X-23a. As the door closes, if the edge of the body liner is too close to the buildup of fines, the result will be a gradual forcing of material between the liner and the door and/or body. A sufficient amount of material can cause the liner to come off and/or to distort the body plates. A solution is to provide adequate clearance when the door is closed.

Rubber belting, such as conveyor belting, has been used to provide a seal at the back of the door. This appears to have limited success in preventing buildup. After some time the muck builds up under the belting and must be cleaned or the problem starts again. In addition to rubber belting, heavy canvas has been used to act as a flexible seal to keep muck out of joints. The same problems apply to muck getting in under the canvas.

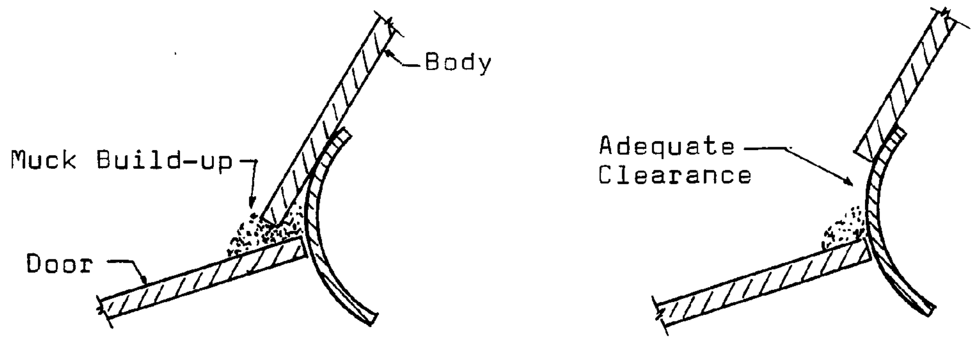
Good initial engineering design appears to be a better solution to preventing problems rather than relying on some means of fixing a troublesome area during mine use.

C. MECHANICAL ELEMENTS

Several mechanical elements of a skip are subject to the shaft environment. Guide shoes prevent the bail from rubbing against the guides. Guide rollers are designed to keep the guide shoes from rubbing against the guides. Bearings are needed at the upper body pivot, the scroll roller, the door pivot, and the door support.

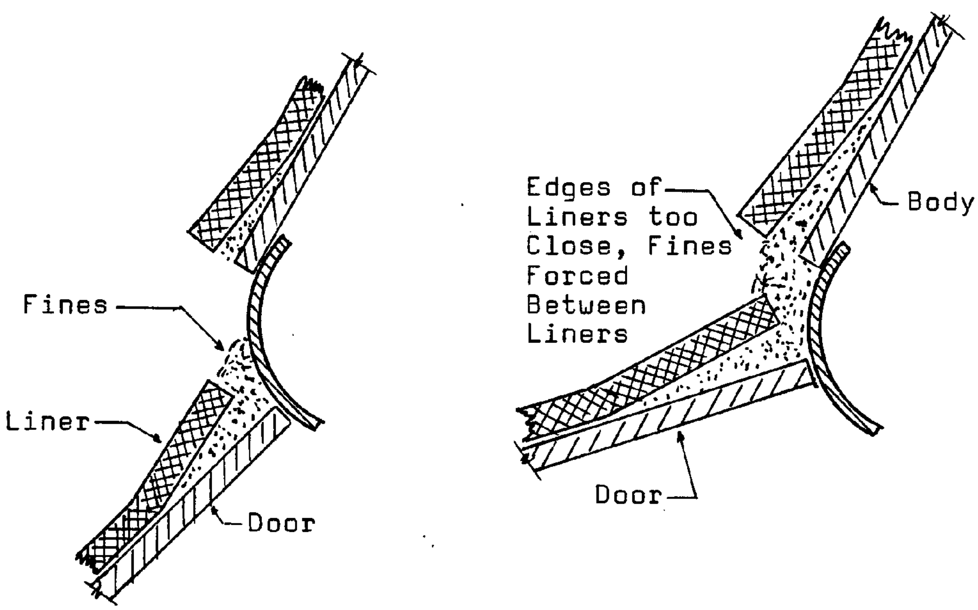
1. Guide Shoes

Guide shoes are considered expendable. There are many materials that have been used for guide shoes. Preference appears to be based on previous experiences. Some of the materials used are: cast iron, mild steel, hardened steel, abrasion-resistant steel, brass, or bronze. Shoes



a. Door will not close properly   b. Door closes properly

Figure X-22 Nutcracker Action Prevents Skip Door from Closing



a. Door opened   b. Door closed

Figure X-23 Muck Build-up Under Liners Tends to Pull Off the Liners used for wood, steel, or rope guides appear to have been made from a large variety of materials. At this time there does not appear to be any plastic material being used for guide shoes.

Wear of guide shoes can be predicted by Equation (X-1). The amount of wear is directly proportional to the load on the guide shoe and inversely proportional to the hardness of the softer material.

In a well-alined shaft with properly adjusted guide rollers the guide shoes do not touch the guides. At some mines any indication of shoes touching the guides calls for local realinement of the guides. It is felt that guides in such a system will never wear out, therefore will not be replaced. Some mines have used corrosion-resistant steels for guides. If the guides are not corroded nor abraded nor overstressed from pounding, they need not be replaced.

Guide shoes have been mounted on rubber to decrease the impact loads. Some designs have been for flexible mountings to allow rotation, thereby acting as self-alining shoes to accommodate guide irregularities. Very long guide shoes, equal to the length of the conveyance, give a smooth ride. This is due to the fact that the guides must be well alined or the conveyance cannot maneuver the shaft.

An item of concern in guide shoe design is the method of fastening the shoes to the bail. Generally the shoes are counter-sunk to receive flat headbolts. If the shoes wear badly, the bolt heads, which may be of hard material, can act as gouges, particularly on wood guides, and cause excessive guide wear. If the guide shoes and bolt heads wear away excessively, the shoes may fall off from the bolts.

An additional consideration is the problem of removing the bolts to replace the guide shoes. Access must be provided to put a wrench on the nuts. This may mean access holes on the sides and back of the bail. Also, the problem of keeping a flat head bolt from turning must be considered when replacing guide shoes. One of the solutions has been to use plow bolts or other special bolts designed to prevent rotation.

2. Guide Rollers

Many designs and many materials have been and are being used for guide rollers. The purpose of the roller is to keep the guide shoes off the guides, thereby keeping the skip from hitting the guides.

Several papers have reported the benefits from using guide rollers on skips and cages. The obvious benefit is the smoother ride, thereby decreasing the dynamic loads between the skip and the guides and the buntons. Another less obvious but significant benefit is longer rope life because lateral vibrations are reduced.

Reference X-8 describes a pneumatic tire guide roller system on wood guides that had been installed in 1941. Construction details and assemblies are given. Several advantages are listed such as smoother, cleaner skipping and more economical operations.

Another rubber-tired application is discussed in Reference X-9, which describes how the travel is much smoother. Tires were ten-inch diameter with one-inch axles.

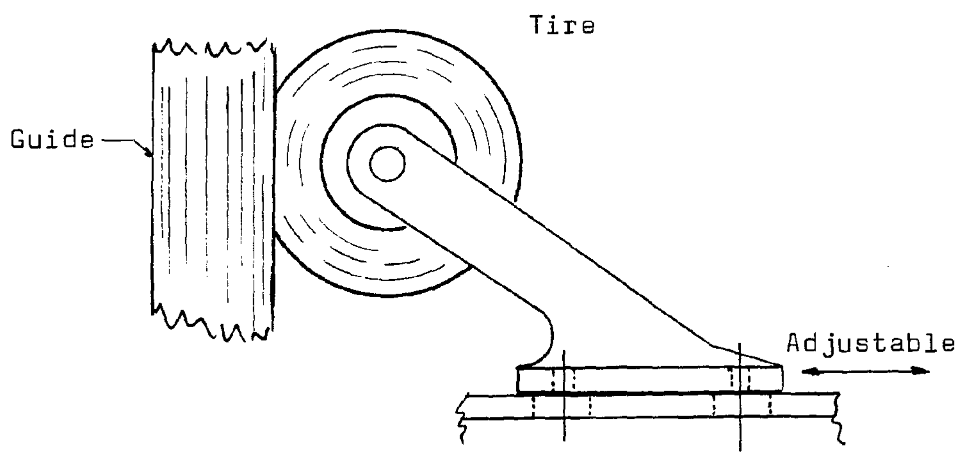
At a conference held in England in 1950, see Reference X-10, the Wire Rope Manufacturers of the United Kingdom reported on shaft guide reactions as well as other aspects of mining related to the use of wire rope. A double accelerometer was used to measure accelerations at 90 degrees to each other in the horizontal plane. Significantly lower dynamic forces were measured when guide rollers were introduced. Data are presented for the same shaft showing the change in horizontal g loading with and without pneumatic tired guide rollers.

Design of guide rollers and test data from accelerometers are presented in Reference X-11. Guide rollers were used when increased production was reqiured as part of an expansion program. Hoisting speed changed from 1800 feet per minute on a seven-ton skip to 2650 feet per minute with a 12-ton skip.

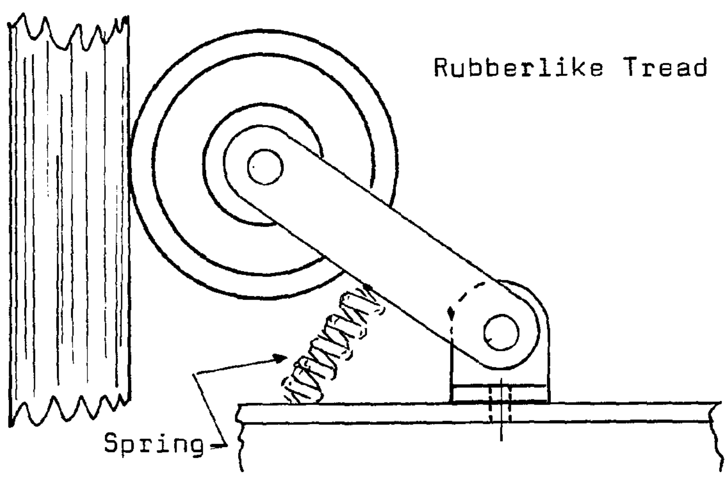
Some installations have kept the guide rollers away from the guides by about a half inch, the purpose being to prevent guide wear due to constant contact. Other mines with guide rollers preloaded against the guides have felt there was no appreciable guide wear from constant contact.

Two basic principles are used to keep the rollers in contact with the guides. One is to mount the roller on a rigid arm and rely on the flexibility of the tire to provide the springing action, as shown in Figure X-24a. The other system is to keep the roller in contact with the guide by a spring pushing the roller against the guide. This scheme is shown schematically by Figure X-24b. A combination of an external spring and tire flexibility is also used.

The tires on the guide rollers have been difficult to design. Generally there is a conflict between a large diameter tire, adequate shaft clearance, and largest cross section of the skip. Unless a decision is made to use a certain size of tire and let other dimensions be



a. Rigid Mount with Flexible Tire



b. Rigid Tire with Spring Loading

Figure X-24 Means of Keeping Guide Roller Against the Guide governed accordingly, the tire will be small. The range of tire sizes appears to be from a minimum of six-inch diameter to 12- or 14- inch diameter.

Smaller tires require more frequent replacement. Also, as tires wear, the tension must be adjusted. At this time there is no measure of how much force nor what the spring constant should be to have the "correct" guide roller system. Experience, judgement, maintenance, and using what works are what make some installations successful. Use of the SKIP program with actual guide geometry will be instrumental in designing the proper guide roller system.

Guide roller tires, solid or pneumatic, are subject to wear. Heat is the biggest problem in keeping good tire life. The temperature of a tire is directly dependent on the load on the tire, the speed at which it is travelling, and the temperature of the surrounding air. Tires are being rated on a load-mile per hour basis, which becomes lower as temperature increases.

For a given temperature if the speed is doubled, the load would have to be reduced by half, and so forth. This is the reason for the greater success with larger diameter guide rollers. When the air temperature increases from 60 to 120 degrees F, the load-mile per hour rating drops off by about 30 percent. A good cold weather guide roller system may not stand up in hot weather.

Guide rollers are called upon to take out the reaction caused by rope twist. Some mines with heavy duty guide rollers, high contact loads, and large diameter guide rollers have no problems with rope twist. Skips running without guide rollers will show signs of wear on one side as the guide shoes are forced against the guides by the twist. There have been cases of steel guides being worn through from constant dragging of the guide shoes on one side of the guide. Some mines, running without guide rollers, will install rope with the opposite lay when changing ropes to put the wear on the other side of the guide.

Many designs have been used to keep the rollers against the guides. The proper selection of spring force, spring constant, and damping is a function of velocity, mass, skip and bail size, bunton spacing, and guide flexibility as well as guide alinement. Analysis, using a program such as SKIP, is needed to make this determination.

3. Bearings

Several rotating points are incorporated in the swing-out bottom dump skip. The top bearing is the body pivot point. It supports the body weight, part of the door weight, and a portion of the muck weight. It must swing through an arc of several degrees when the skip dumps and returns.

The back of the door is supported by a bearing that must go through a rotation of 45 to perhaps 60 degrees or more. The front of the door is supported by either a set of rollers or a link, depending on design. The door rollers will undergo several revolutions as the door opens and closes. Links will go through a rotation of several degrees.

The scroll rollers, one on each side, pull the body out for dumping. They see many revolutions for each entry into and exit out of the scrolls.

Guide roller bearings may rotate for thousands of miles of travel during their lifetime.

For the most part, all the bearings except the guide rollers have a consistent set of repeating loads of the same magnitude and direction. Guide rollers may have changes in their load pattern, depending on their adjustment and changes in guide alinement.

This section will not attempt to incorporate the well established design formulas used for selecting bearings. The several bearing manufacturers have excellent technical sections as a part of their bearing catalogs. Nor will this section make any recommendations or suggestions for types of bearing to be used at the several bearing points. Several general comments, however, will be made regarding bearing design as related to skips.

Bearings have a basic static load rating and a dynamic rating. Loads on a bearing can be either radial loads or thrust loads or a combination of both. The basic static load is limited by permanent deformation of the load-carrying surfaces or by fracture of various parts of the bearing.

Dynamic loads on a bearing are identified by the speed, magnitude, and direction of the load. Also to be considered is a varying bearing load as opposed to a constant load. These factors - load, speed, loading history - affect the life of the bearing.

Relationships between load, speed, and bearing life are not linear. In general, if the load is doubled, the bearing life is reduced to about one-tenth. Conversely, if the load is reduced by one-half, the life is increased about ten times.

Doubling the speed reduces bearing life by one-half.

Reducing the speed by one-half doubles the bearing life.

Bearings are rated differently by the various manufacturers. When comparing bearings, the number of hours at a given RPM or a total number of revolutions must be considered. The basis for rating can be as different as one million revolutions to 90 million revolutions.

Where possible, bearings should be sealed. The shaft environment is very hostile to bearings.

Proper consideration for lubrication must be given to bearing design. Grease fittings in easily accessible locations will aid field maintenance which will improve bearing performance.

Bearing replacement must be considered in the design. It is fairly easy to weld a bearing into an assembly that later will require major surgery to remove and replace.

As with other elements of a skip, numerical analysis of loads and speeds can be used to select bearings that will perform for a given period. In addition to theory, the designer must use other information such as that gained from actual operating conditions. Where this information is not available, such as for a new design, then numbers must be generated and applied. In addition to establishing a basis for design by using loads, etc., the use of analysis may identify bearings that are over-designed and, as such, are more expensive. Economy in skip design can be an important consideration. On the other hand an underdesigned bearing is expensive when considering lost production and maintenance cost for a skip out of operation due to an early bearing failure.

XI. SELECTION OF GUIDES AND BUNTONS

Skip guides in vertical shafts are made of wood or steel or wire rope. The hoisting in vertical shafts can be by drum hoists or friction hoists, and friction hoists may be tower or ground mounted. Guides used for skips in newer installations tend to be either steel or rope in concrete-lined circular shafts. Practically all skips run on two guide systems when using wood or steel guides. Where skip-cage combinations are used in vertical shafts, the guides for drum hoists are wooden guides because of the need for safety devices. Where friction hoists are used with skip-cage combinations, there appears to be a tendency toward steel guides because safety catches are not required.

A. GUIDES

Guides are made from several species of wood, several kinds of steel shapes, and locked coil wire rope.

1. Wood

The current use of wood guides for skips appears to be in older shafts, shafts in bad ground, where skip-cage combinations are being used, and possibly short term operations.

Many species of wood are used, such as several kinds of fir, several kinds of pine, mahogany, oak, Apitong, Keruing, Karri, Tallowood and others, such as local woods. Selection appears to be influenced by previous experience. Quality material graded for mine guides is becoming more difficult to obtain and is relatively expensive. At some mines the selection of the wood is less demanding, and guides with knots and green wood are used.

Laminated wood guides are being used. Rather than having a single piece of timber of a given cross section and length, the guide is made of several layers of wood bonded together. There are advantages to this construction such that warpage and checking can be minimized. The need to preselect certain trees as suitable for guide material is not necessary. There also may be advantages to using more than one species when making a laminate to suit a particular shaft.

To keep wood from drying out, some means of storage under moist conditions is employed. At some locations outdoor sprinkler systems spray water on the piles of stacked wood guides. At some mines wood guides are stored outdoors with no provision to keep them moist.

There does not appear to be a uniform means of storing wood guides. The reasons for keeping guides moist is to prevent checking, warping and change in cross section and length.

Depending on the means of storing and the shaft conditions, guides may become either dry or moist after they are installed. To keep guides wet, water is sprayed on the guides in the shaft. Wet guides do not wear as readily as dry guides. Wood guides that become wet and dry periodically will tend to rot much faster than if they are always wet. If guides are kept moist prior to installation, then dry out after thay are installed, they may crack and warp. This condition affects guides in the headframe.

If local wood is used and installed while it is green, generally there is some cost saving and the warping after curing and the remoistening are avoided. Decisions must be made regarding the desirability or need for strength, straight grain, and other such wood properties.

One consideration for using wood guides appears to be the anticipated life of the mine. Generally if the use of the shaft is projected to be a short time, then wood guides may be more economical than other types. Shaft depth directly affects overall guide costs - the deeper the shaft, the greater the lineal feet of guide material.

When fastening wood guides to buntons for a skip-cage combination using a drum hoist, it is important to keep the bolts and bolt heads away from the teeth of the safety dogs. The size and number of fasteners needed for such guides is based on the weight of the skip-cage and the number of g's during stopping.

Spacing of guide sets for wood guides varies from three feet, four inches to 15 feet or more. Guide sizes range from 3 1/2 inches to 6 x 10 inches.

Where guide rollers are not used, guides show a decided wear pattern, becoming tapered on one side where the guide shoes rub. The rubbing happens because of rope twist when using a drum hoist. To keep the wear from becoming excessive, rope of the opposite lay may be used when changing ropes. The skip is then twisted in the opposite direction and wear patterns develop on the other face. Without guide rollers, one face of the guides and one guide shoe will tend to stay in contact and will wear.

For friction hoists there generally is an even number of ropes, half being right lay and the other half being lang lay.

Wood guides have joints, usually located at the center of guide sets. To keep the ends alined there are many splice details. Preferences are based on experience. To keep the ends from drying, metal plates may be nailed across the ends. Because wood will change dimensions due to moisture content, provision has to be made at the splices to allow for some end movement.

If the guide set spacing is small, then the guides may span across one or more sets between splices. Maintenance of the guides is required to assure the integrity of the bolts fastening the guide to the supporting sets. In some instances a steel guide-back is used where steel sets are installed. The guide is fastened to a continuous steel beam located behind the guide.

Wood guides have the advantage of being alined more easily than steel guides. Notching and shimming are standard woodworking operations.

The tendency for skip compartments is to use steel or rope guides.

2. Steel

Several shapes are used for steel guides. They include hollow steel tubing, rail, elevator tees, angle, and pipe. To keep the guide shoes from rubbing on the steel, the skips generally are equipped with guide rollers.

A steel guide system is considered to be more permanent than wood guides. Some installations use corrosion resistant steels (such as Cor-Ten). It is felt that steel guides in some shafts will never have to be replaced; there is no wear and there is no corrosion.

The design of the guide may be based on the expected stress from lateral forces between the skip and guide. Static values for this force have been estimated from about 10 to 20 percent of the total skip weight (skip plus muck). In addition, work in South Africa has suggested the maximum deflection of the guide be limited to less than one inch.

When steel guides are used, the buntons are generally steel. Provisions for adjusting the fact to face dimension and providing alinement are required in the design of the fastening. Shims are used as well as slotted connections to give adjustment in two horizontal directions, in and out as well as back and forth.

Engineering consultants have specified tolerances for hollow steel tube guides as shown in the following sketch, Figure XI-1:

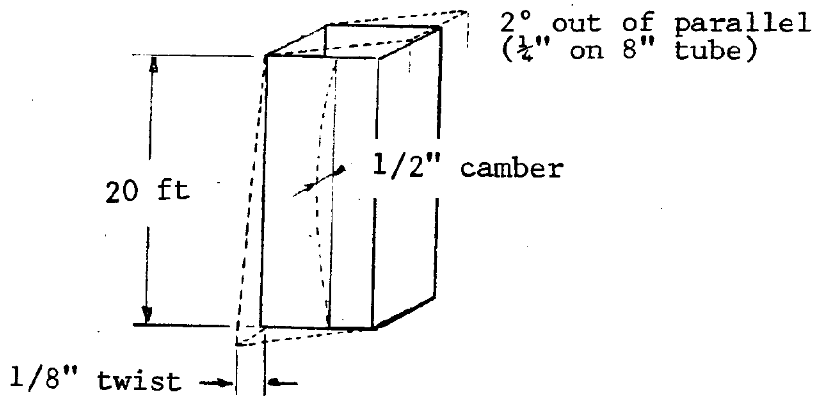


Figure XI-1 Allowable Tolerances Steel Tube

Splices are an important design detail for steel guides. Problems can arise from shapes such as elevator tees where the leg of the tee, projecting toward the center of the compartment, is not fastened. As the leg deflects from lateral load, the guide shoes and rollers follow the guide; however, at the splice the leg of the unloaded tee is straight and the rollers and shoes hit a bump. This can cause serious operational problems. There does not appear to be a simple way of joining the leg part of elevator tees.

It does not seem feasible to weld the ends of steel guides. Temperature changes in a shaft several hundred or perhaps several thousand feet deep would cause serious problems due to expansion and contraction.

Hollow structural tubes can be spliced by welding steel bars inside the tubes so that the tubes slip over the "fingers" and are kept positively alined. The design of such a splice should consider whether the joint can be taken apart in the shaft to replace a guide without having to remove many guides before the joint will separate. Also, the joint should be designed to make installing a new guide a simple procedure.

Splices on rail can be made more rigid than on elevator tees. The head of the rail provides the running and guiding surface leaving the web of the rail available for splice bars.

Angles and other shapes can be spliced to maintain transverse rigidity across the joint to prevent the guides from being misalined at the joint.

Where guide rollers have not been used on steel guides the guides have worn through from rope twist. Long sections of steel tube have had the tube wall worn away on one side. Wear and corrosion brought about the need for guide replacement in a relatively short time.

When selecting steel guides, it is necessary to consider the thickness of the thinnest part. Proper allowance must be made for corrosion. A steel tube having thin walls initially will not be usable after corrosion products reduce the walls to zero thickness.

3. Rope

A long established guide system is the use of rope guides. Under average operating conditions a 20- to 25-year rope life can be expected, based on British experience. A rope guide installation is simple and clean. The conveyance moves very smoothly along the ropes.

Rope guides are of locked coil construction, details of which are shown in Section VII, Guide Materials. The tensioning of rope guides is presented in Section II, Skip and Guide Systems.

Placing of rope guides and spacing of the conveyances to determine when to use rubbing ropes are shown in Figure XI-2, taken from Reference XI-1. Note that the sleeves are placed on one side of the conveyance rather than on four corners. Equally spaced symmetric shoes when running on parallel guides have a tendency to cause binding. Accelerometer tests have shown a smoother ride when the ropes are not at the corners. This type of behavior is also discussed in Section XII, Conceptual Design.

There are two ways of tensioning rope guides; one is by springs and the other by weights. The use of springs can lead to some difficulties. One problem is muck building up in the spring holding device preventing further tensioning and the conveyance runs on loose guides. The difference in behavior of the rope is noticeable. Another problem is not making proper adjustment for temperature changes. If the rope elongates, the guides will be loose. If the rope shrinks, the result can be a broken rope or badly damaged anchor points. A third problem is not knowing what the tension in each rope is because of spring relaxation or difficulty in measuring the spring length to determine the load. Still another problem is broken springs and the

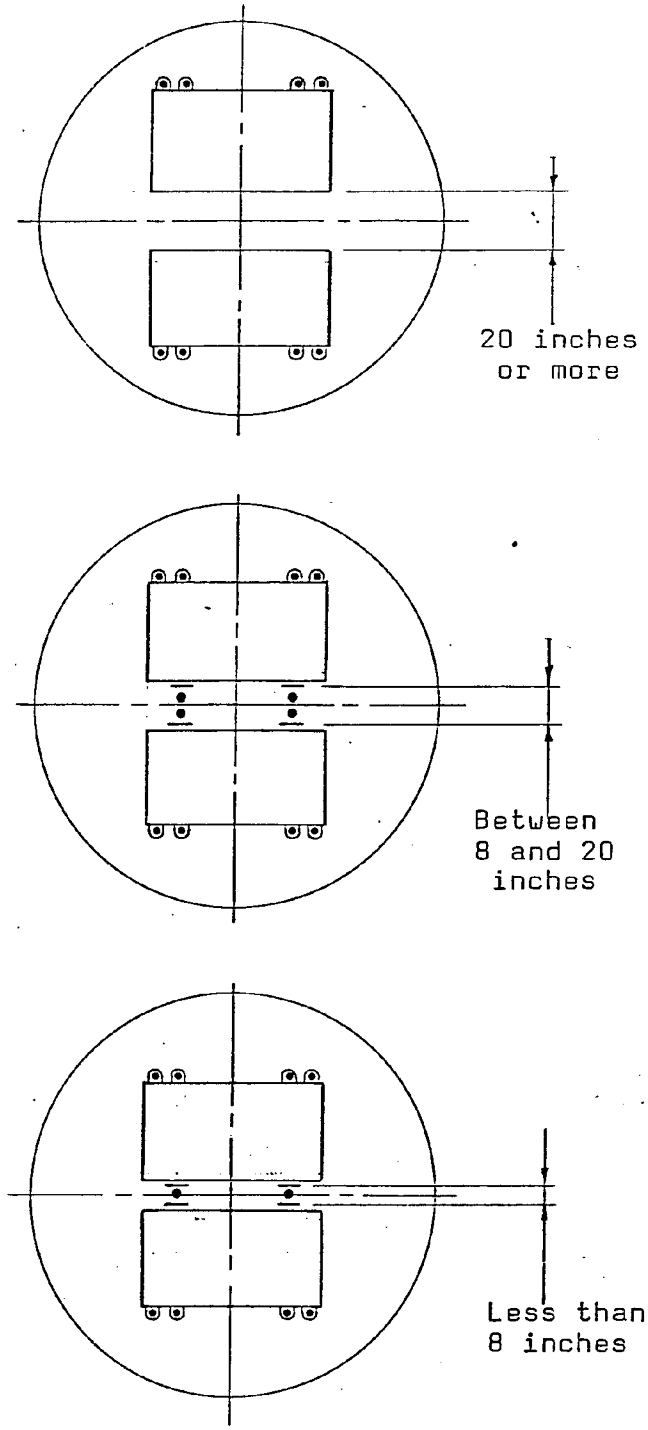


Figure XI-2 Arrangements of Rope Guides and Rubbing Ropes difficulty in detecting and replacing the broken one (ones).

Weights are far more trouble free. One reason for not using them is the additional excavation required at the bottom of the shaft.

Corrosion is an important factor when using rope guides. General corrosion of the rope, both on the surface and internally, must be considered. An additional consideration is corrosion of the rope at the fittings at the top and the bottom. Several kinds of rope terminations are used, some specifically designed for tensioning rope guides.

Design of a new system using rope guides appears to be based on adopting the same procedures used on a previous successful installation. Wind tunnel tests have been performed on the behavior of skips on rope guides to gain more knowledge. Points to consider are such factors as the behavior of the skips at the passing point and how soon any disturbances damp out after passing. Also there is the effect of cross ventilation in the shaft and how that affects the behavior of the skips. Further, there is the effect of the shaft wall on the skip and how close a skip can be located to the wall. A skip moving with the air flow will have low velocity air passing over it. Going against the air flow, skips will have high velocity air passing over them. The tendency to fly as an aerodynamic body must be considered as part of the design. Passing skips will travel in turbulent wakes. The shape of the skip body may be either blunt or streamlined. British practice favors square tops and bottoms.

A good discussion of the use of rope guides in deep shafts is given by Gray (Ref. II-3). For narrow crooked shafts, rope guides are not possible. For large circular concrete lined shafts, rope guides are being used. The design of the skip and the shaft have to be coordinated by considering the air flow requirements, tonnage requirements, hoisting depth, velocity, and other such factors.

B. BUNTONS

Wood or steel buntons or guide sets or shaft sets hold wood or steel guides in the shaft. In rectangular shafts the shaft sets are used for ground support as well as to position and hold the guides. A summary of sizes and arrangements is given in Reference XI-2, Section 5.1.3.

In rectangular shafts the set spacing is determined by the requirements for ground support. This spacing may change at different levels in the shaft. In circular shafts where the shaft walls are concrete-lined, the bunton spacing is constant and selected for convenience, strength, and economy.

Common spacing in rectangular shafts is about five feet, although the range of spacing can be from 3 feet, 4 inches to 10 or 12 feet. Air flow in the shaft is not affected appreciably after the bunton spacing is greater than 20 to 30 feet. This is shown in Figure VIII-1. Larger bunton spacing makes the guides more flexible to side forces.

For circular shafts, steel sets are used to support the guides. Generally the bunton spacing is greater than the spacing found in rectangular shafts with 15 feet being commonly used and some mines use 20 foot spacing.

Perhaps the spacing of the guide shoes and guide rollers should be coordinated with the spacing of buntons and guide splices. Although it has not been demonstrated at this time, it would seem that from an engineering standpoint, the spacings should not be the same. The matter of how different is still to be resolved. At present there does not appear to be any attempt to relate these two dimensions; the shaft designer designs the shaft and the skip fabricator designs the skip.

Aerodynamic shapes have been considered in Canada and South Africa for a number of years. Observations of new and planned installations in the United States do not appear to be concerned with improving the air flow in shafts by using aerodynamic shapes. A summary of this subject is given in Section VIII A 3 of this report.

It would seem that simple triangular caps added to the buntons of existing mines would increase air flow and/or decrease power requirements. Where there is muck spillage in a downcost shaft, the tops of the bunton are covered by muck that forms itself into what appears to be an aerodynamic shape.

Bunton protection against corrosion is of some concern. A number of protective coatings have been used. These include primer and paint, epoxy coatings, galvanizing, tar, and other such materials. The most severe corrosion problems appear to be associated with wide flange beams placed with their webs horizontal. The two flanges form a deep trough.

For well alined shafts the guides, supported by adjustable buntons, are kept to within one-eighth of an inch in 100 feet, noncumulative; and top to bottom of shaft centerlines are within one-quarter of an inch in thousands of feet.

XII. CONCEPTUAL DESIGN

Shaft layout and skip design for various mines have followed a somewhat repetitive pattern. Shafts have been either rectangular or more recently circular, and skips have been consistently rectangular boxes. Three basic skips have developed: the Kimberly, or overturning skip; the bottom dump with swing-out body; the bottom dump with fixed body and door bottom. The latter two are the subject of this report.

There are other considerations for skip design and performance. From a weight consideration, a cylindrical skip is a more efficient structure when compared to a reinforced rectangular box. Still greater weight efficiency can be found by reinforcing a cylindrical structure using composite materials. Cylindrical skips present some interesting shaft layouts whereby the number of guides and the amount of guide set material can be reduced.

Some wheeled vehicles such as railroad cars are inherently stable by using tapered treads. There are some concepts that will permit a skip to be center seeking on the guides.

For a mechanism travelling on a pair of parallel guides, the supports on the guides must be offset to prevent the supports from grabbing and/or chattering.

A. CYLINDRICAL SKIP

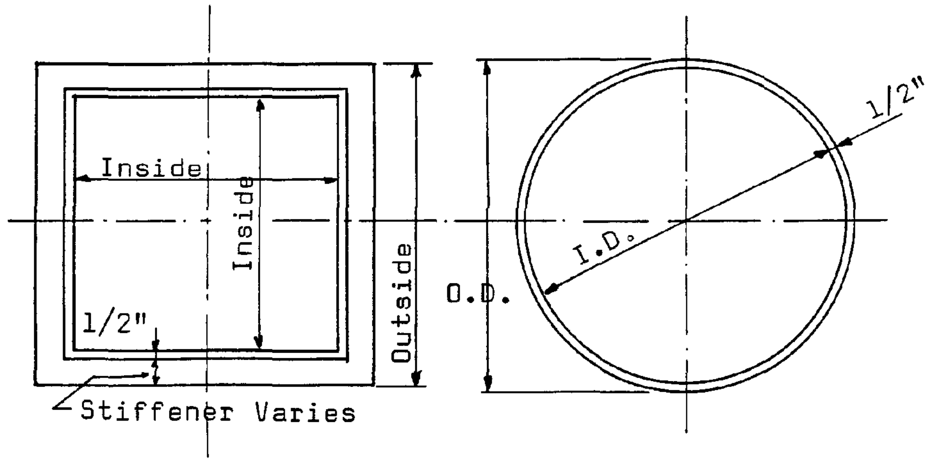
A cylindrical skip presents two new areas in design. One is the skip body and door, and the other is the shaft layout.

1. Skip Design

To consider a cylindrical skip body versus a conventional rectangular body, there must be some advantages. It would appear that a cylindrical skip does not use the cross section as efficiently as a square or rectangular skip.

When looking at the stiffening required for a rectangular body, it becomes apparent the cross sectional area of the inside and the overall outside dimensions of both constructions are similar. This is apparent in Figure XII-1. For a range of inside areas from 4 to 42 square feet, the overall cross sectional dimensions vary such that the cylinder is one inch smaller overall to two inches larger overall.

Another aspect of cylinders compared to box-type design is the much lighter construction possible for the same stress level. For a four-foot square box compared to a



Inside Area (ft^2)	Rectangular			Cylindrical		Cylinder is	
	Inside	Stiff	Outside	I.D.	O.D.	Smaller	Larger
4.00	2'-0"	2"	2'-5"	2'-3"	2'-4"	1"	-
6.25	2'-6"	2"	2'-11"	2'-10"	2'-11"	-	-
9.0	3'-0"	2"	3'-5"	3'-4"	3'-5"	-	-
12.25	3'-6"	3"	4'-1"	3'-11½"	4'-0½"	½"	-
16.00	4'-0"	3"	4'-7"	4'-6"	4'-7"	-	-
20.25	4'-6"	3"	5'-1"	5'-1"	5'-2"	-	1"
25.00	5'-0"	4"	5'-9"	5'-7½"	5'-8½"	½"	-
30.25	5'-6"	4"	6'-3"	6'-2½"	6'-3½"	-	½"
36.00	6'-0"	4"	6'-9"	6'-9½"	6'-10½"	-	1½"
42.25	6'-6"	4"	7'-3"	7'-4"	7'-5"	-	2"

Figure XII-1 Comparison of Cross-sectional Area for Square and Circular Skip Bodies four-foot-six-inch-diameter cylinder, the difference in weight is more than 7 1/2 to 1; but, as shown in Figure XII-1, the inside area is the same. Conversely, for the same thickness of material the stresses in the body are 7 1/2 to 1 less in the cylinder. As shown in Section X, low stresses are important when designing to resist fatigue, corrosion, wear, and stress concentrations.

Some models have been made to investigate the details in designing a cylindrical skip. Construction is considerably simpler; a single longitudinal weld must be compared to four corner welds plus welded stiffeners, to form the body. Figure XII-2 shows a cylindrical skip body. A stiffening ring at the top of the body maintains the circular shape at the free edge. Body pivot is located similarly to the pivot on a rectangular skip body. The bottom of the skip is a conical section requiring one longitudinal weld, a circumferential weld to the body and to the flange. The door pivot is supported on reinforcement placed on the outside of the cone.

Figure XII-3 shows details of the door for a cylindrical skip. The door diameter is the same as the skip body, thereby allowing the cylindrical skip to be closer to the face of the guide by several inches.

The body pullout roller is shown on the door rather than on the body itself. This permits additional reinforcing on the door rather than adding to the body.

The back of the door is shown closed, thereby preventing spills down the shaft. If required, a seal can be used on the bottom of the flange. Wet muck may be handled without excessive leakage. A liner on the door can be used to prevent excessive wear during dumping and to absorb some of the shock during loading. Other liners can be placed in the skip as needed to protect the body from wear.

A construction worth considering is shown in Figure XII-4. Two intersecting cylinders connected by a single flat plate can weigh less than two individual cylinders. This same principle has been used in optimizing the weight of spherical pressure vessels.

The flat plate provides the radial component of force required to keep the intersection in equilibrium. There is some shear and local bending at the intersection; however, these are very local. The use of this type of construction is illustrated in the following portion of this section showing shaft layouts.

2. Shaft Layouts

The question of how a cylindrical skip fits into a shaft

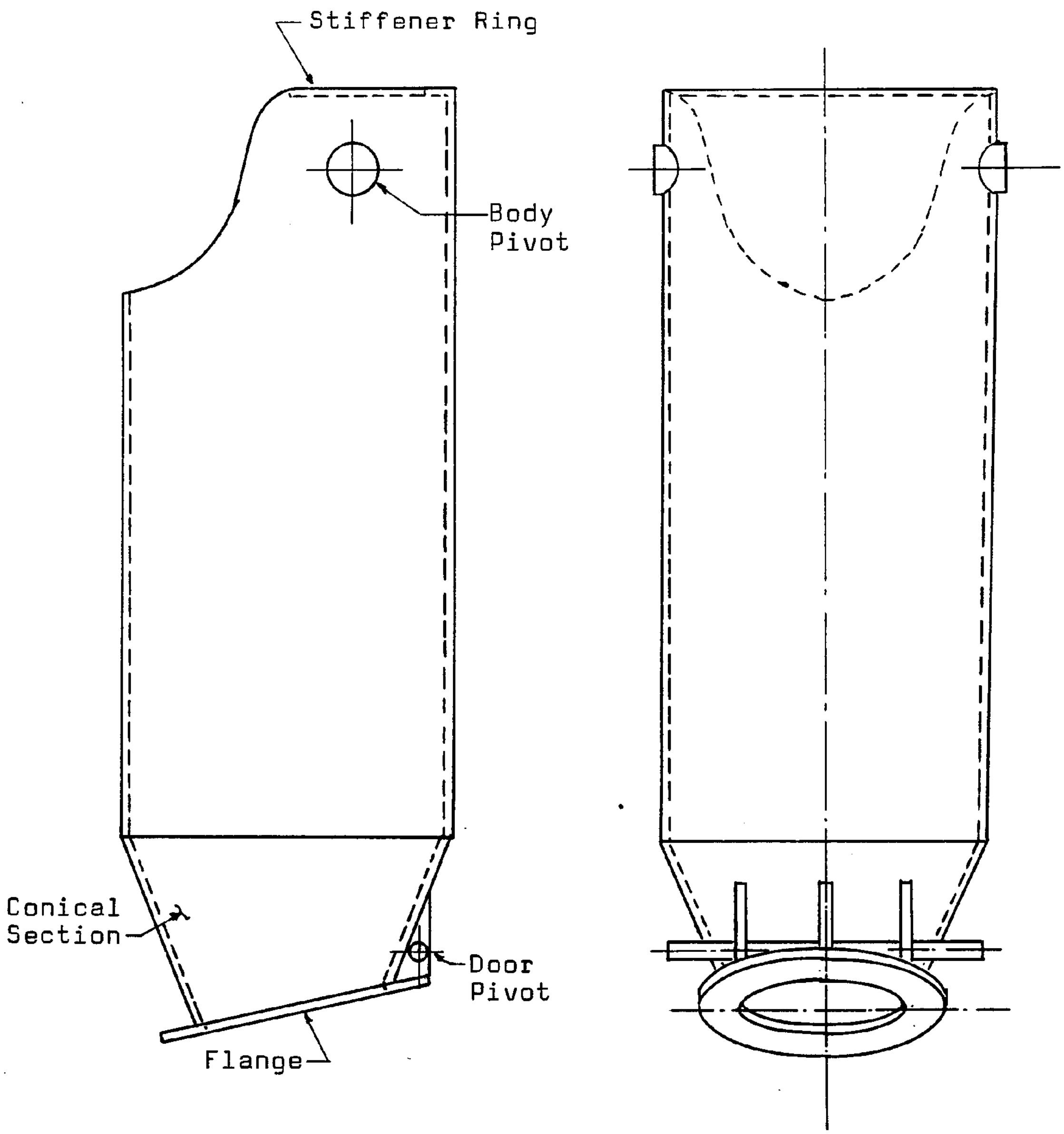


Figure XII-2 Cylindrical Skip Body

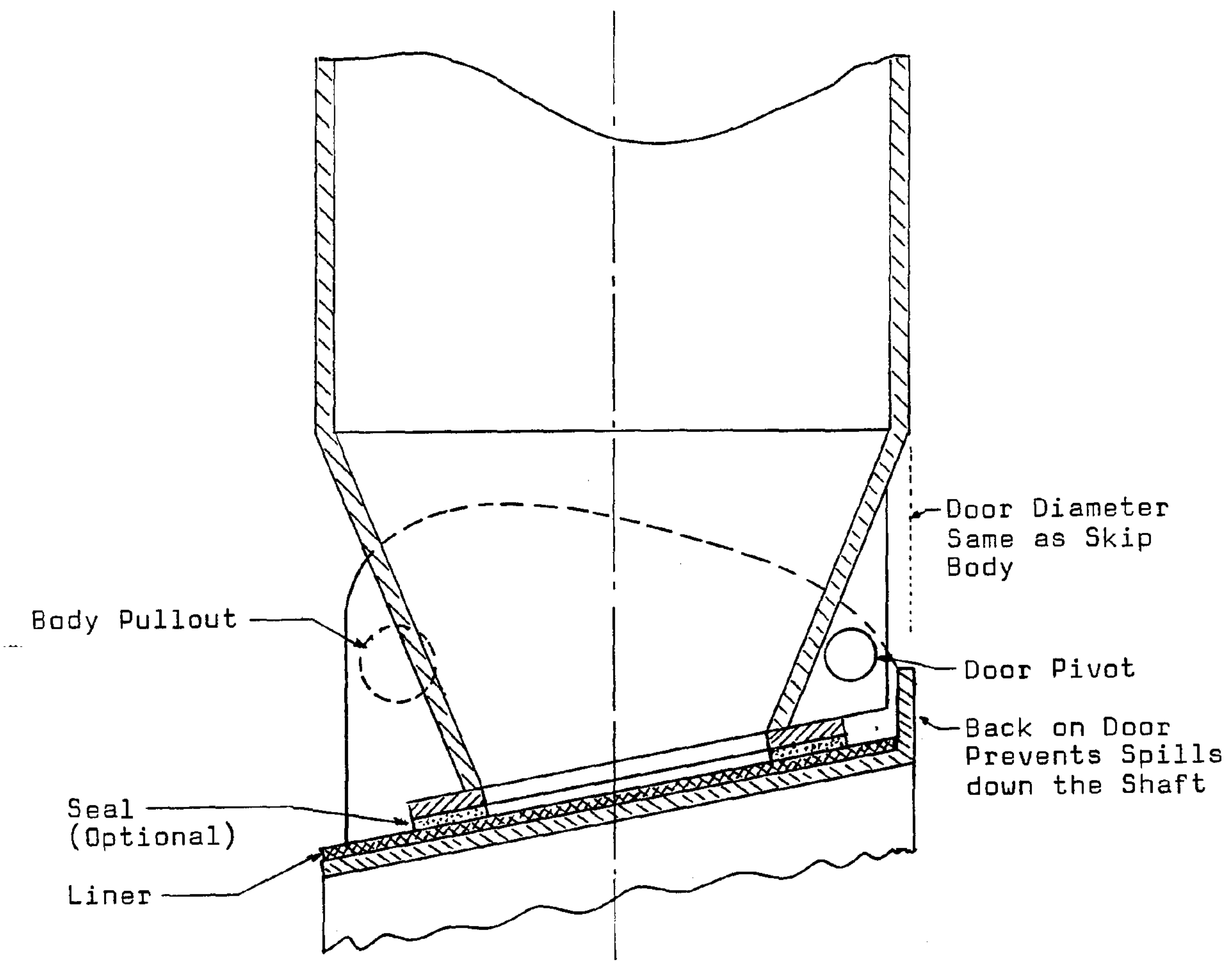


Figure XII-3 Detail of Door for Cylindrical Skip

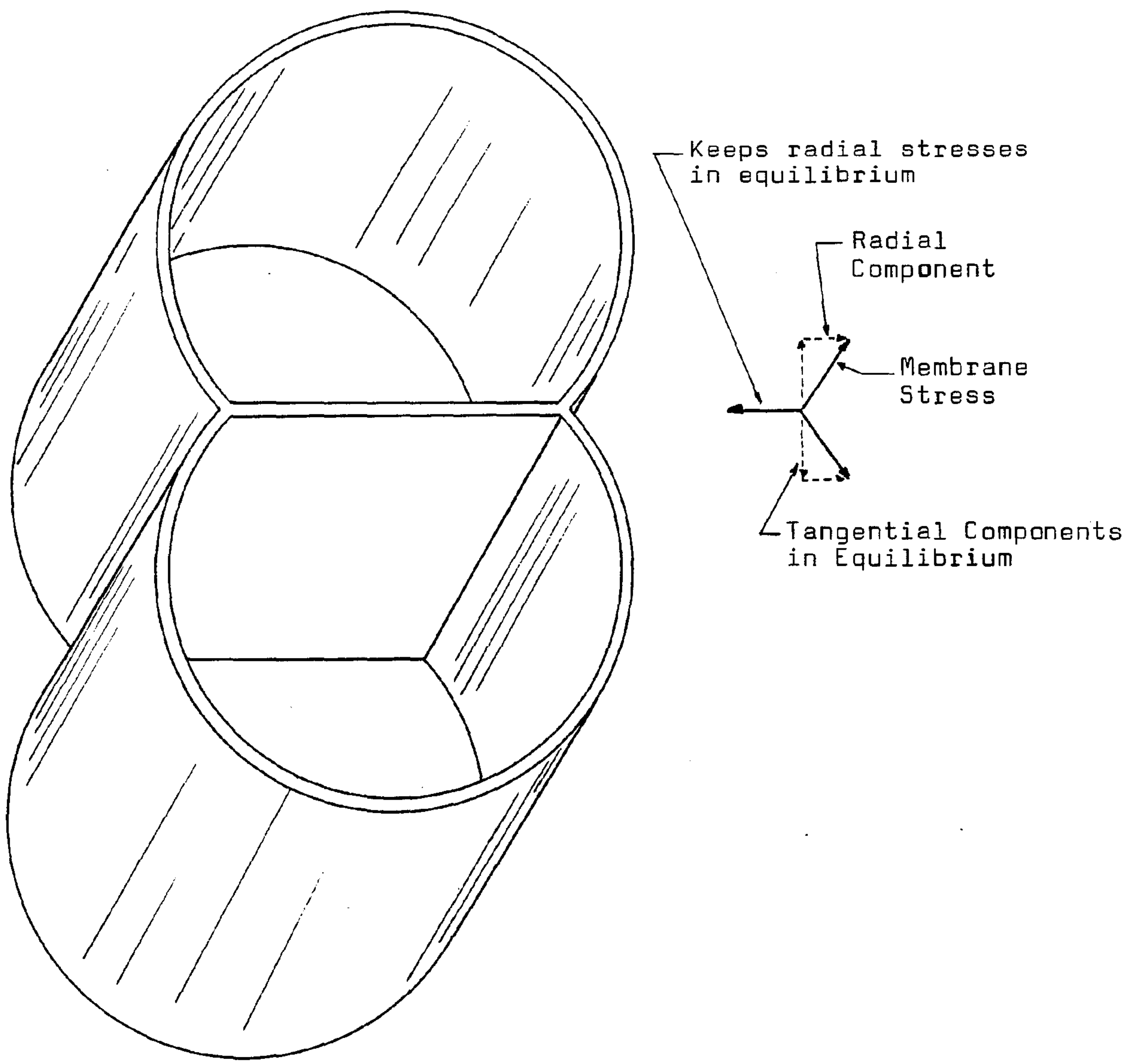


Figure XII-4 Minimum Weight Intersecting Cylinders requires some thought. Each shaft and each skip are designed individually; however, some generalities can be established.

In Figure XII-5 is shown a three-compartment rectangular timbered shaft. Each compartment is five by five feet clear inside dimension. The left-hand compartment shows a rectangular skip with an inside area of 14.9 square feet using a four-inch guide and holding a two-inch clearance. Keeping the guides the same, a cylindrical skip (shown in the center) has a cross-sectional area of 12.1 square feet. However, the clearance has become six inches; therefore, the shaft could be made eight inches narrower to keep the clearance at two inches.

The right hand compartment shows the guides in the corners. With a two-inch clearance, the cylindrical skip has a cross-sectional area of 16.5 square feet, an increase of about eleven percent over the rectangular body. Note the bail is triangular for the right-hand compartment. Also, the number of guide rollers for the right-hand side is eight, compared to twelve for the other two compartments. In addition, the guide rollers form a center seeking suspension, the subject of a later section. Dumping the skip with guides oriented at 45 degrees to the centerline of the headframe is different from today's practices. There do not appear to be any major technical problems to prevent such guide orientation.

Looking at circular shafts, an example of a circular and rectangular skip body is illustrated in Figure XII-6.

For a 24-foot-diameter shaft the left-hand compartment shows a rectangular body with a 54.0-square-foot cross-sectional area. The right-hand compartment has a cylindrical skip and the cross-sectional area is 63.6 square feet. The guide orientation is at 45 degrees.

A more noticeable difference in shaft layout and skip area is given in Figure XII-7. Two cylindrical skips are shown in a 24-foot-diameter shaft. Each body has an area of 70.9 square feet. More noticeable is the shaft steel arrangement, comparing Figures XII-6 and -7. Figure XII-7 shows a far more open arrangement, giving better ventilation. By eliminating the several long structural steel members, the cost of the shaft sets become noticeably less. Also, instead of using four guides for two skips, Figure XII-7 shows three guides for two skips.

Figures XII-6 and -7 show a 24-foot-diameter shaft for illustration purposes only. The same general trends for skip areas, amount of shaft steel, and number of guides apply to other diameters. Many arrangements are possible, and no attempt is being made to make final recommendations.

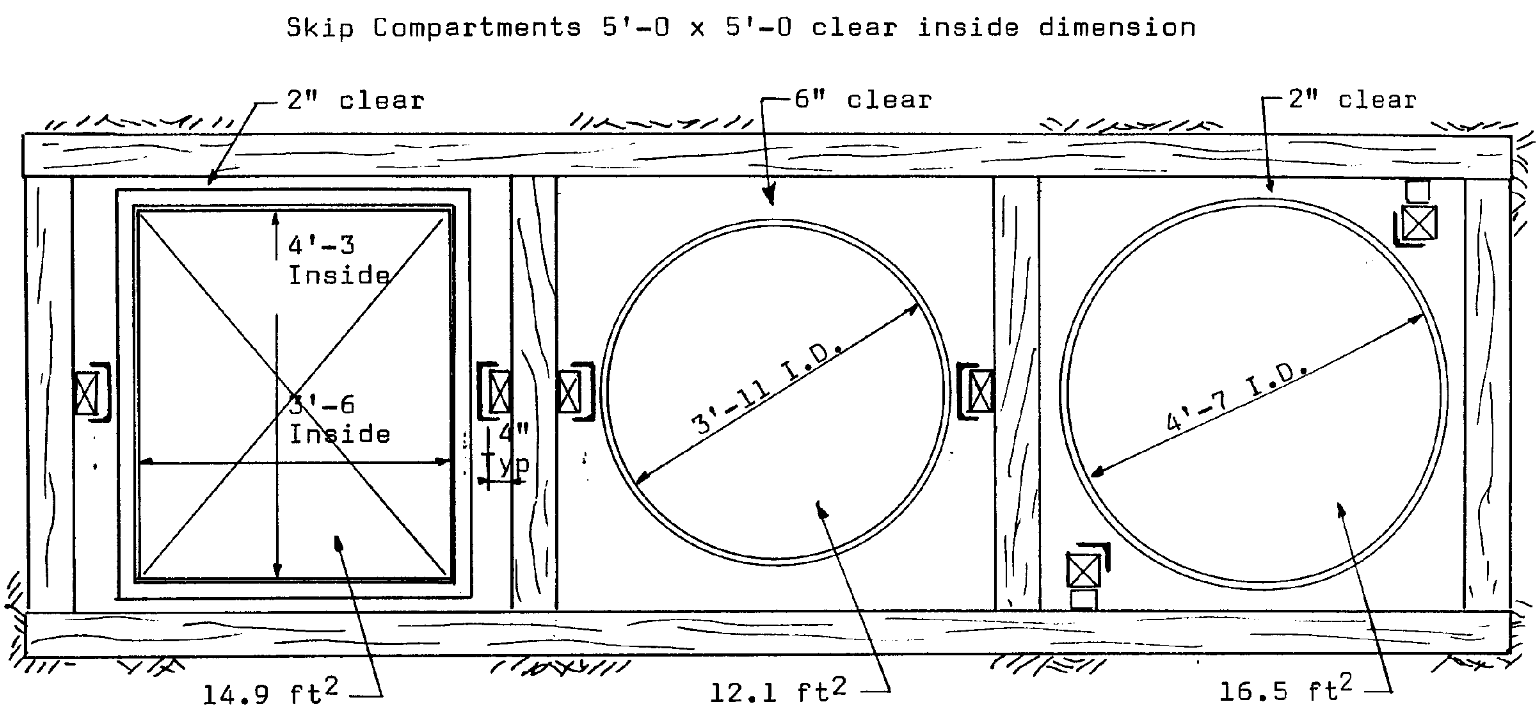


Figure XII-5 Three Skip Configurations in a Rectangular Shaft

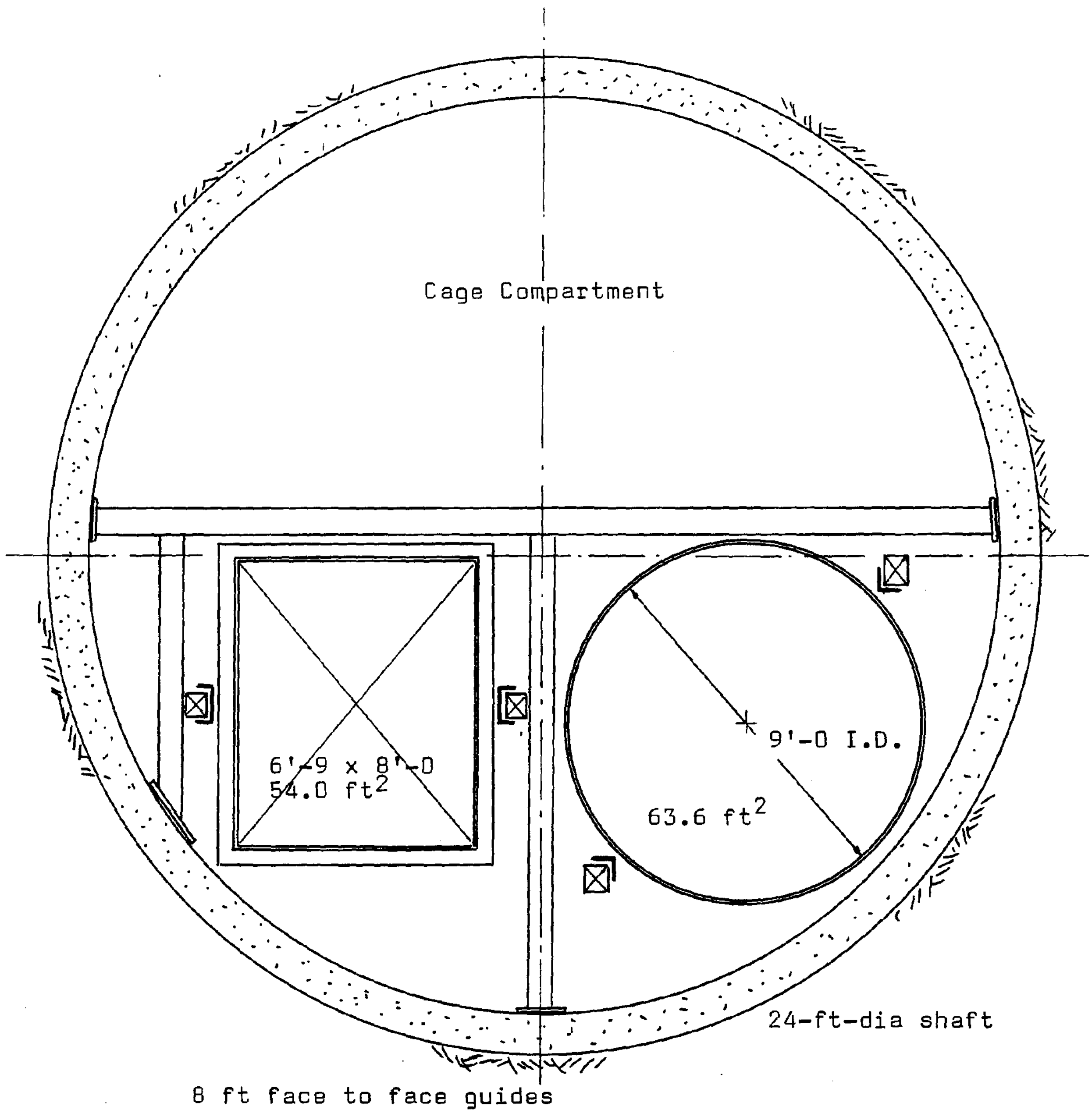


Figure XII-6   Two Skip Configurations in 24-ft Dia. Shaft

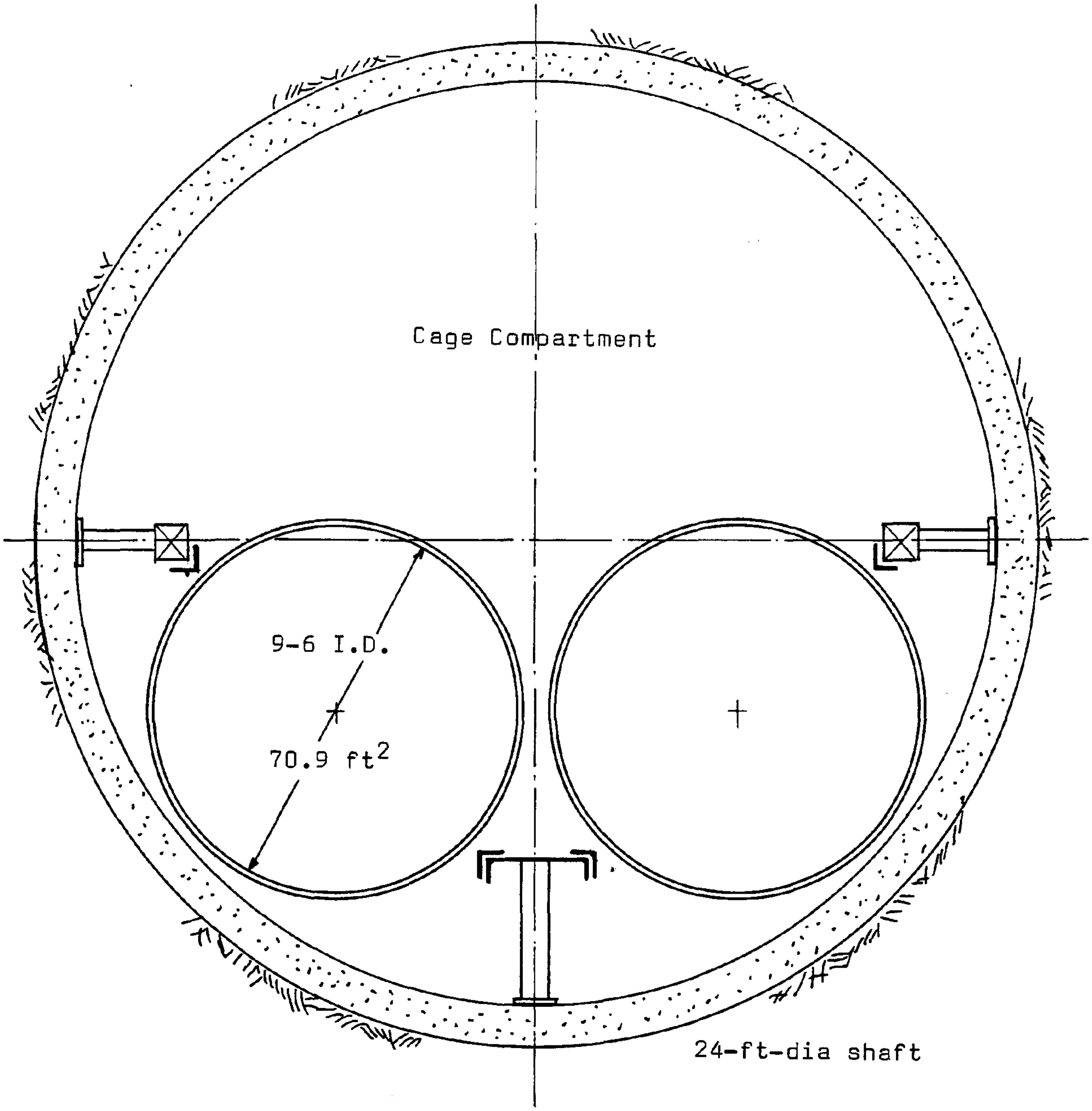


Figure XII-7 Cylindrical Skips in 24-ft-dia Shaft

As stated earlier, each shaft and skip are separate design problems. This section considers other ways of arranging and designing more effective shafts.

Figure XII-8 is a conceptual design for a circular shaft using intersecting cylinders whereby several optimum conditions are shown. As mentioned earlier, the intersecting cylinders provide a lower weight body than two separate cylinders for the same cross-sectional area. Another optimization comes from better shaft utilization by putting circles inside of a circle rather than rectangles inside of a circle (see Figures XII-6 and -7).

Further optimization is found by using three guides rather than four. This concept does not use conventional shaft steel but rather a system of rod trusses shown in Figure XII-9. Several advantages can be seen in the figure, such as: inexpensive rod construction; turnbuckles for adjustment and alinement; simple connections to the shaft wall and to the guide; and very little interference with air flow.

Guide rollers are center seeking to stabilize the skip, and eight rather than twelve rollers are needed for each skip.

Although this is presented as a concept and many details require clarification, from an engineering point of view the several optimizations suggest serious consideration to this type of construction.

B. CENTER SEEKING GUIDE ROLLERS

An example of the usual guide roller arrangement with three rollers at each corner located at 90 degrees to each other is shown in Figure XII-10. Rollers are kept against the guides by various springing systems. The guides are rectangular, and the rollers are perpendicular to three of the four faces. The skip is kept centered by the spring forces acting at each guide roller.

A self-centering tracked wheel system is shown in Figure XII-11. Railroad type wheels have tapered treads. When the tapers are projected toward the center, a pair of cones is formed such that the tendency is for the assembly to cwnter itself. Springing is not required to keep this construction centered as it rolls along the track.

If the tread were not tapered, the wheels would wander on the rails until the flanges hit the rail. Such behavior is similar to the present usual guide roller system. As shown in Figure XII-10, the wheel motions are perpindicular to each other; therefore, the action of one set of wheels does not affect (try to center) the other set of wheels.

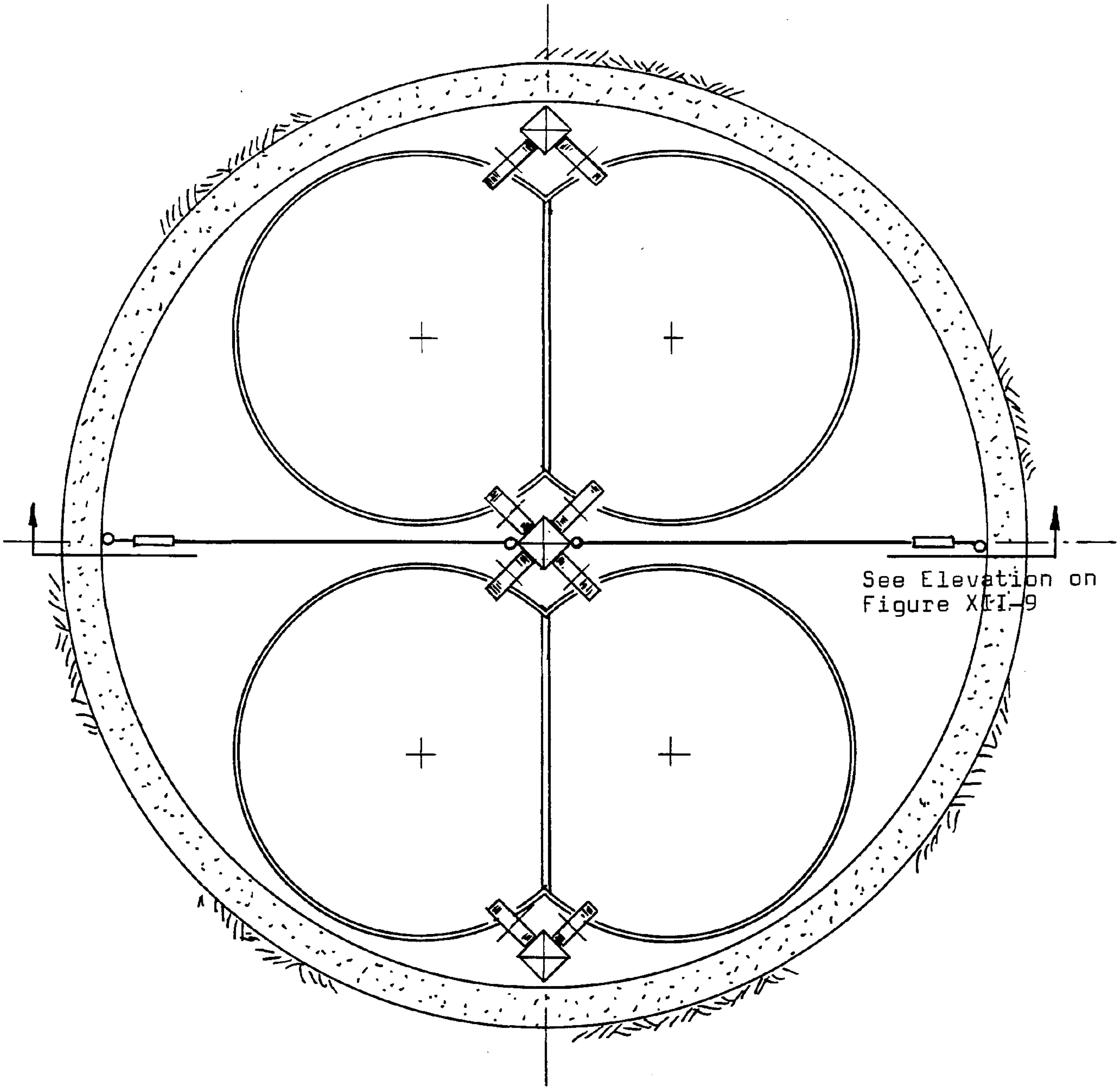


Figure XII-8 Intersecting Cylinders Used for Skip Bodies

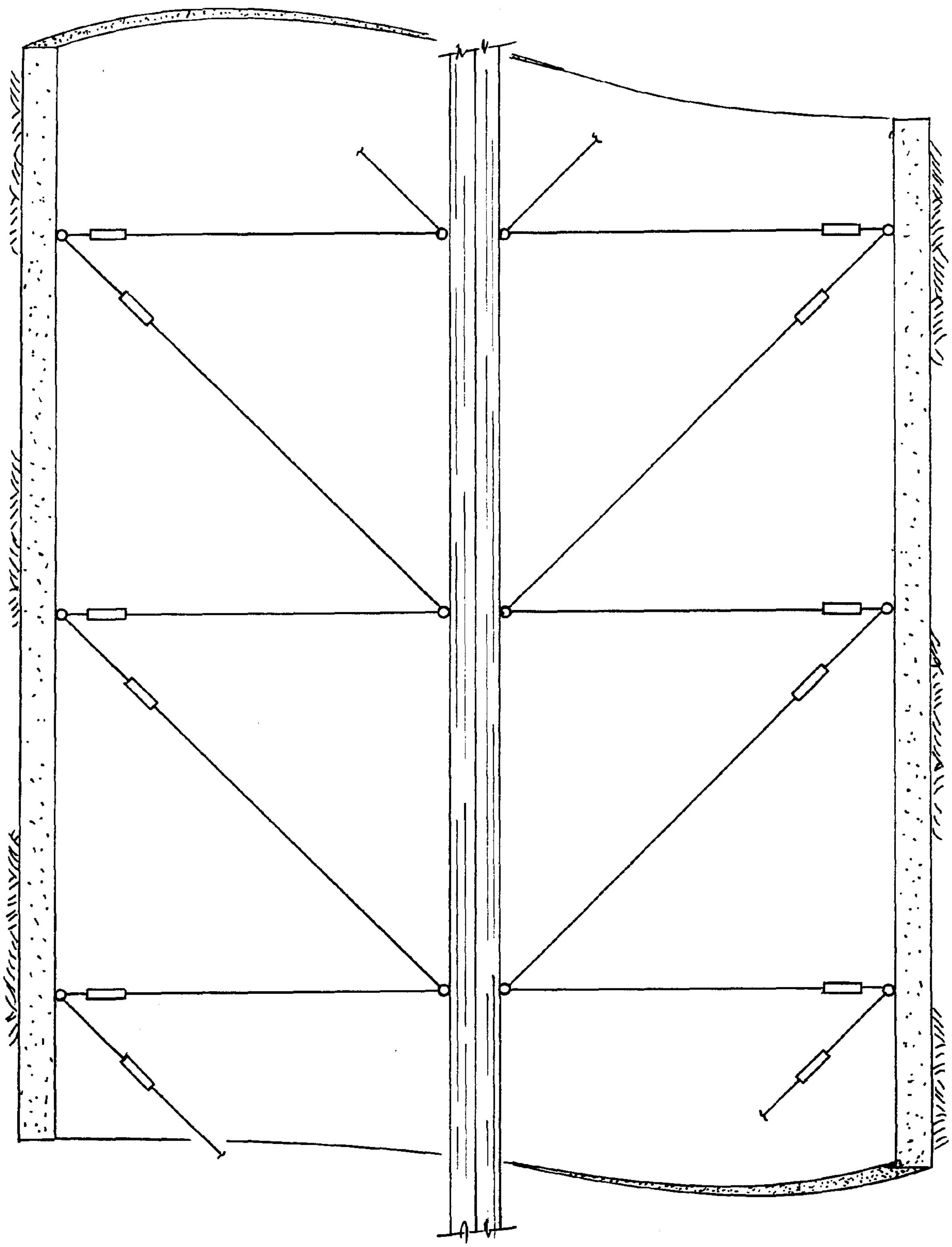


Figure XII-9 Elevation Showing Center-Guide Support

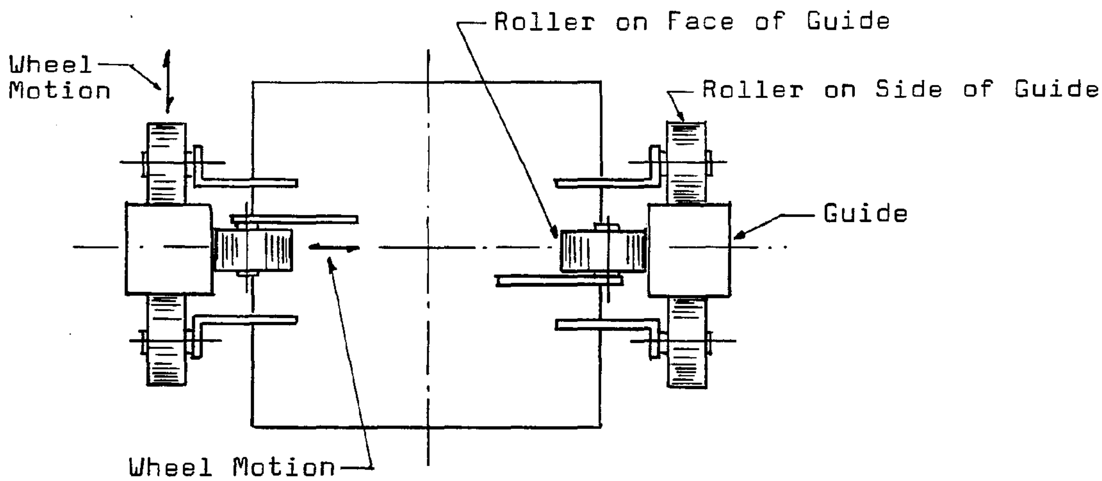


Figure XII-10 Usual Guide Roller Arrangement

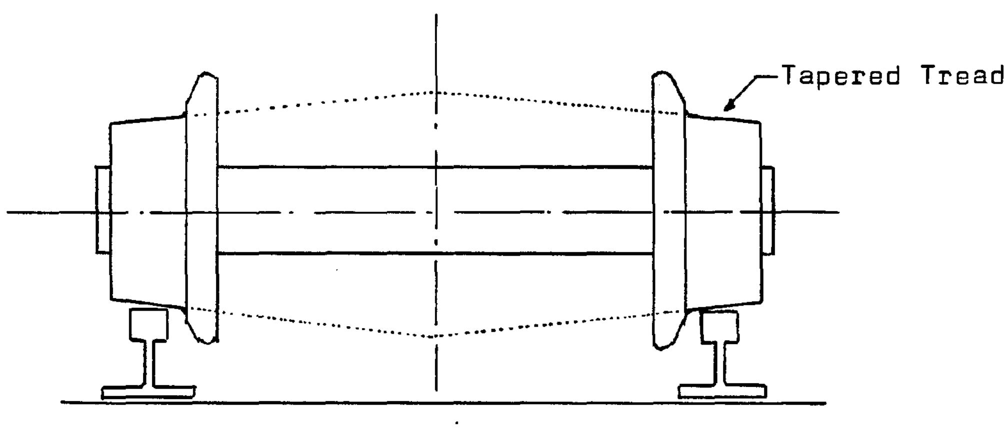


Figure XII-11 Self-centering Railroad Type Wheels

Some concepts for center-seeking systems using three rollers at each corner are illustrated in Figure XII-12. These concepts use both the geometry of the system and springs to help keep the skip centered. A system similar to using railroad flanged wheels is sketched in Figure XII-12a. The guides are rectangular with the self-centering being provided by the tapered treads. When considering wear on the guides and the wheels, this may not have general application for high speeds and large loads.

Figure XII-12b shows tapered faces on the guides and conical wheels. This is a center-seeking system with the side wheels providing an inward restoring force. This type of wheel will scrubthe guides and cause excessive wear at high speeds and large loading. Tapered cross section guides appear not to have been used; and if attempted, may not be practical from many considerations.

A system that does not scrub the guides is illustrated in Figure XII-12c. The wheels are canted on tapered guides. As mentioned above, the tapered guides may present other problems.

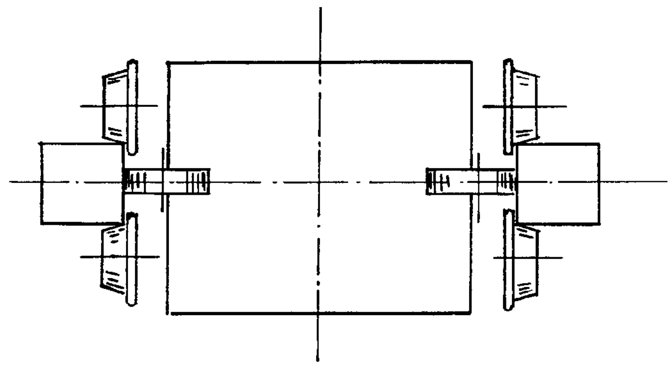
If the guides are tapered to 45 degrees, easily done by rotating a square to 45 degrees or by using a structural angle, two rollers can be used at each corner instead of three. This makes a difference of eight guide rollers per skip as compared to twelve per skip. This system, shown in Figure XII-13a, using angle guides, has been installed in some mines. The system is center-seeking because of its geometry and is assisted by springs. This provides an additional stability not found in the more usual system shown in Figure XII-10. With the axles parallel to the faces of the guides, there is no scrubbing between wheels and guide faces.

The simplest center-seeking system used one wheel at each corner, for a total of four wheels per skip. Figure XII-13b shows guide faces oriented at 45 degrees. A double flanged wheel rides on two faces of the guide. There is a tendency for this type of wheel to scrub. To decrease the scrubbing action, the flanges have to be smaller, thereby increasing wheel pressures and wear.

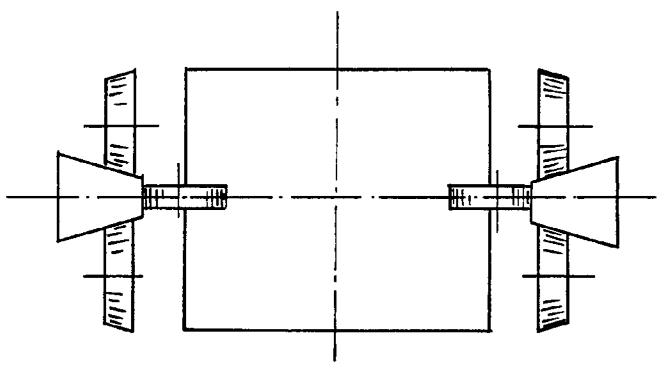
The technology developed for vehicles running horizontally has some application for mine conveyances running vertically in a shaft. The problem of suitable suspensions to decrease dynamic load and the amount of wear can be solved by using basic engineering principles and some fundamental analysis.

C. OFFSET GUIDE ROLLERS AND SHOES

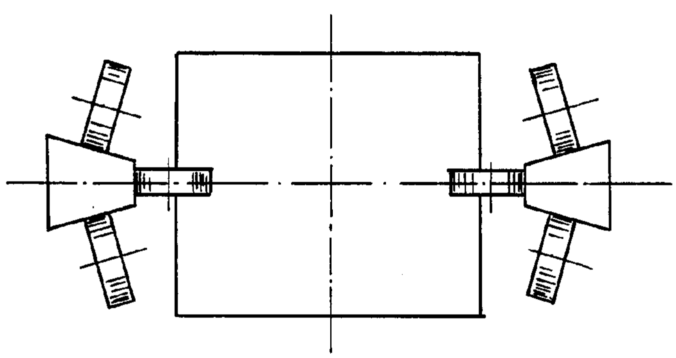
Another means of improving the ride characteristics of a



a. Rectangular Guides, Tapered Flanges

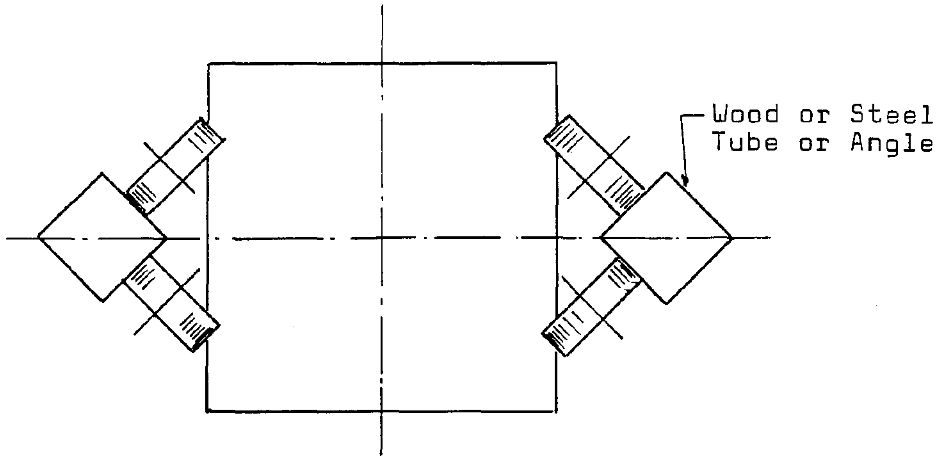


b. Tapered Guides, Conical Wheels

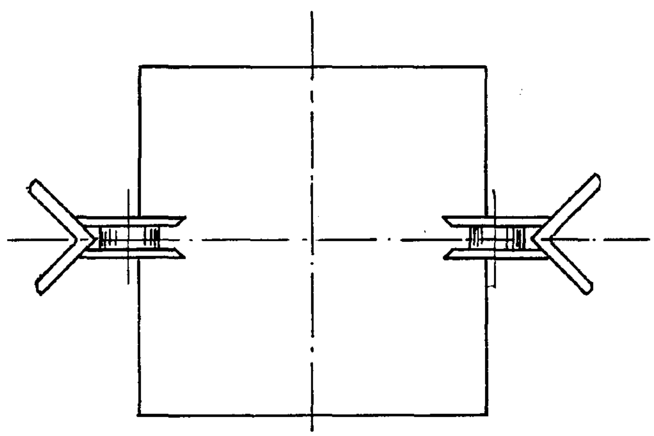


c. Tapered Guides, Wheels Canted

Figure XII-12 Center-seeking Systems, Three Rollers at Each Corner



a. Two Rollers at Each Corner



b. One Roller at Each Corner

Figure XII-13 Center-seeking Systems with Less than Three Rollers at Each Corner skip appears to be a nonsymmetric suspension system. Reviewing the response curves in Section V shows a repeating set of oscillations on perfectly alined guides as well as misalined guides. The computer model is representative of current skip construction in which the guide rollers and guide shoes are placed symmetrically on the bail. This is shown schematically in Figure XII-14a.

One way to change the symmetry of that construction is to use different spring constants (hence different restoring forces) for the guide rollers. There are several spring constant possibilities such as: each roller is different; some are alike; the top ones are larger than the bottom; the bottom are larger than the top; the side ones are the same on one side but different from the other side, etc.

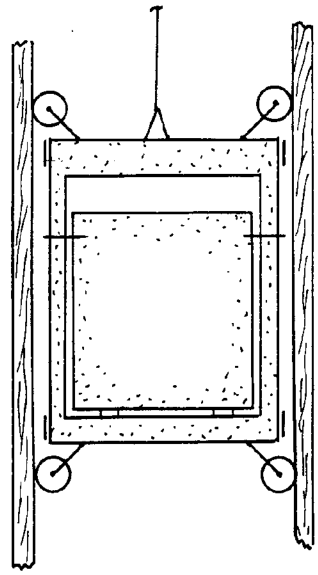
If a guide survey were available, the best combinations could be found by running the dynamic computer program of Section V. One criterion for selecting springs for the suspension system could be the combination that produces the smallest dynamic force on the guides at the desired hoisting speed. Another criterion could be to set the highest allowable value of lateral dynamic force and find the fastest hoisting speed.

Until a parametric study is made, there are numerous unanswered questions when deciding the best system. The several things to be considered are:

1. Weight of the system
   (Empty and full skips react differently)
2. Hoisting Speed
3. Length of the skip
4. Location of the center of gravity
5. Location of the guide rollers
   (symmetric or offset)
6. Properties of the guides
7. Bunton spacing
8. Spring constants of the guide rollers
   (same or different)
9. Clearance between the guides and guide shoes

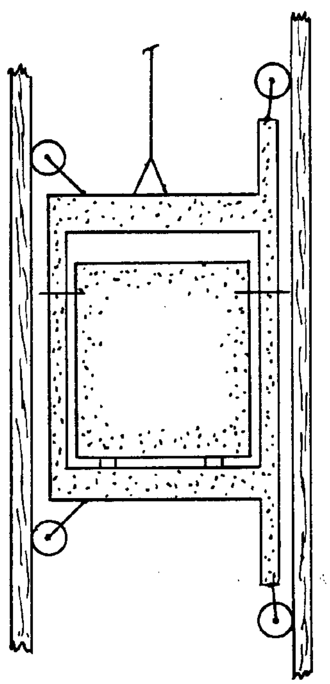
For a given shaft and skip the main adjustments for providing a smooth ride are the location and springing of the guide rollers and the clearance of the guide shoes. If the shoes are very close to the guides, the full benefit of a springing system may not be realized because the shoes will hit the guides in fractions of an inch.

It appears that clearances between guides and guide shoes have remained the same for skips with and without guide rollers. Perhaps additional clearances are needed to let



Spring Constant, k;
All Equal
All Different
Some Equal

a. Symmetrically Placed Guide Rollers and Shoes



Spring Constant, k;
All Equal
All Different
Some Equal

b. Offset Guide Rollers and Shoes

Figure XII-14 Symmetric and Offset Guide Rollers and Shoes the springing system become effective. Also to be considered in further studies is the use of shock absorbers designed for the skip conditions at any particular mine.

D. THE USE OF COMPOSITES

Many advances have been made in the use of composite materials for lightweight structures. As yet composites have not been used to any extent in the design of skip bodies and bails. One use of filamentary composites would be winding the high-strength thread-like material around the cylinder building up layers of filaments.

Hoop stresses (the stresses going around the cylinder) are twice as high as longitudinal stresses (the stresses parallel to the axis of the cylinder) in a pressurized cylinder. In a skip body the pressures are highest at the bottom; therefore, the hoop stresses vary from a maximum at the bottom to zero at the top.

The use of composite overwrap for a cylindrical skip body is shown in Figure XII-15. A skip has triangular pressure distribution on the sides. The hoop stress varies with the pressure; therefore, the amount of overwrap varies, more required at the bottom than at the top.

In an overwrapped cylinder the strains in both materials are essentially the same. To find the plate thickness and the amount of overwrap that will keep the stresses in the metal and the composite within their allowable limits, we find the ratio of hoop stresses as a function of the elastic moduli such that,

$$\frac{\sigma_1}{E_1} = \frac{\sigma_2}{E_2} \qquad \text{(XII-1)}$$

where σ_1 = hoop stress in metal
E_1 = elastic modulus of metal
σ_2 = stress in filaments
E_2 = elastic modulus of overwrap

We can also show, using the notation of Figure XII-15, that for a pressure, p, at a depth:

$$pr = \gamma r \ell = \sigma_1 t_1 + \sigma_2 t_2 \qquad \text{(XII-2)}$$

where γ = unit density of material

Using Equations XIII-1 and -2, we are able to write the expression for the hoop stress in the metal, σ_1, and in the composite overwrap, σ_2, as

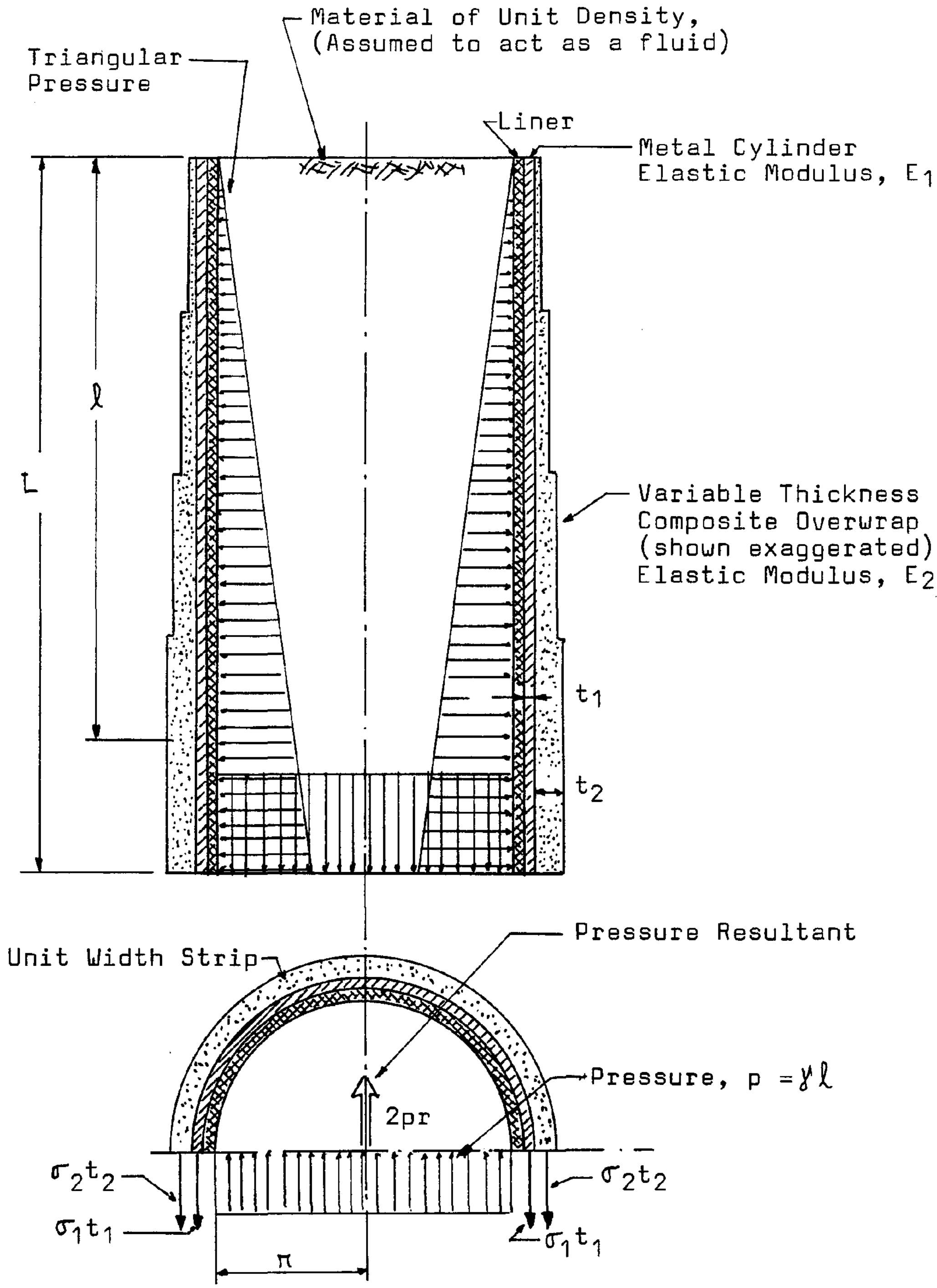


Figure XII-15 Overwrapped Cylinder

$$\sigma_1 = \gamma r \ell \frac{E_1}{E_1 t_1 + E_2 t_2} \qquad \text{(XII-3)}$$

$$\sigma_2 = \gamma r \ell \frac{E_2}{E_1 t_1 + E_2 t_2}$$

The maximum stresses occur at the bottom when $\ell = L$.

Values of ultimate stress and elastic modulus for several metals and filamentary materials are given in Table XII-1. Ultimate strengths must be factored to allowable stresses.

Note that in solving for stresses in Equation XII-3, all units must be consistent so that the radius r, depth ℓ, thickness t_1 or t_2 should be in inches. The elastic moduli E_1 and E_2 should be in pounds per square inch and the density of the material in pounds per cubic inch (equals pounds per cubic foot divided by 1728). This will give the hoop stresses in pounds per square inch.

It will be apparent from the densities in pounds per cubic inch (pci) in Table XII-1 that using an overwrap such as glass-epoxy on aluminum will be lighter than an all aluminum skip or an all steel skip. It must be remembered that the ultimate tensile strengths given in Table XII-1 need to be modified by service factors. These are discussed in Section X.

The longitudinal stress in the metal cylinder (the overwrap is assumed not to work longitudinally) for a unit width strip is

$$\sigma_\ell = \frac{\text{Wt of muck}}{\text{Area of metal cylinder wall}}$$

$$\sigma_\ell = \frac{\gamma \pi r^2 L}{2 \pi r t_1}$$

$$\sigma_\ell = \frac{\gamma r L}{2 t_1} \qquad \text{(XII-4)}$$

The metal liner of thickness t_1 is in a biaxial state of stress, σ_1 in the hoop direction and σ_ℓ in the longitudinal direction. Whereas the hoop stress varies along the axis with the depth ℓ being a maximum at full depth, L, the longitudinal stress σ_ℓ is constant regardless of depth and depends on the overall depth, L.

Table XII-1 Typical Material Properties*

Material	Ult Tens (ksi)	Mod Elast (10^6 psi)	Density (pci)	Spec Str (10^6 in)	Spec Mod (10^8 in)
Magnesium	40	6.5	.064	0.6	1.0
Steel	220	29	.280	0.8	1.0
Aluminum	80	10	.098	0.8	1.0
Titanium	180	16	.165	1.1	1.0
Beryllium	80	42	.067	1.2	6.3
Glass-Epoxy	190	7	.062	3.1	1.1
Graphite-Epoxy	200	18	.057	3.6	3.1
Boron-Epoxy	205	33	.070	2.9	4.7
Boron-Aluminum	230	34	.093	2.5	3.7

* Higher and lower values can be found from many similar references. Ratio of matrix and filament affects values.

XIII. FIELD TESTS AND INSTRUMENTATION

This test program outlines tests that can be used to evaluate improvements in new skip systems determined by improved design criteria. Measurements made during such tests can be used to correlate the response of the dynamic computer model to the response for a full scale skip.

The test zone would be conducted at a shaft permitting hoisting velocities up to 2000 to 3000 feet per minute. The guides should be of wood or steel in good condition. Muck must be available for testing of the loaded skip configuration.

A. TEST DESCRIPTION

Several different skip component configurations should be evaluated for improvement and correlation with the dynamic model. The different component configurations are 1) with or without guide rollers, 2) light, medium or hard spring rate of guide roller when used, and 3) empty or full load in the skip. The number of test series required to evaluate all these component configurations is twelve. Table XIII-1 outlines the test series matrix and indicates the component configuration for the specific test series. The test series has been set up to minimize change of component down time so that the entire test sequence can be performed in a smooth and efficient manner. The Table XIII-2 gives a schedule of events beginning with checkout testing and evaluation. As shown, ten working days are required to perform the twelve test series and accomplish the necessary component changeover between tests.

Each test series consists of four test runs. A test run is performed by running the skip in the desired component configuration either up or down the shaft at the required hoisting speed. The four test runs for each test series will be performed at different hoisting speeds; namely 1/4, 1/2, 3/4, and full speed.

Three test series are scheduled to be performed on one day. The difference in the configurations is one is with a loaded skip going up, the other an empty skip going up and down. By using the normal shaft loading and dumping mechanism, that loading and unloading can occur in a short period of time, allowing a three test series to be performed in one day. The changing of the springs and addition of the guide rollers will take more time and as shown in the schedule, one and two days respectively have been allotted for these changeover tasks.

Table XIII-1 Test Series Matrix

Test Series	Guide Roller		Spring Constant			Load		Direction	
	Yes	No	Lt.	Med.	Hard	Empty	Full	Up	Down
1		x				x		x	
2		x				x			x
3		x					x	x	
4	x		x			x		x	
5	x		x			x			x
6	x		x				x	x	
7	x			x		x		x	
8	x			x		x			x
9	x			x			x	x	
10	x				x	x		x	
11	x				x	x			x
12	x				x		x	x	

Note: Each series to be run at 1/4, 1/2, 3/4, and full hoisting speed.

Table XIII-2 Test Schedule

Task	Day 1	2	3	4	5	6	7	8	9	10
Checkout Tests	→									
Test Series 1, 2		→								
Test Series 3		→								
Install Guide Rollers			→	→						
Test Series 4, 5					→					
Test Series 6					→					
Install Med. Springs						→				
Test Series 7, 8							→			
Test Series 9							→			
Install Hard Springs								→		
Test Series 10, 11									→	
Test Series 12									→	
Contingency Test Day										→

Parameters to be measured are the guide roller forces, guide shoe contact, skip body acceleration and displacement, and skip body position and speed. All the transducers will be calibrated and/or proper operation verified before and after each day's testing. The transducer signals will be conditioned by units located on the skip. The signals will be recorded by an FM recorder also mounted to the skip. The recorder and signal conditioners will be shock mounted to the skip.

Once calibration is complete and all components are ready for testing, the data recording system will be started manually; and the skip will then make a pass at the desired speed down and then up the shaft. When the skip returns to the start position, the recorder will be turned off manually. Each test run will be repeated in this manner. If it is possible for someone to ride the conveyance, this would prove beneficial in that any anomalies could be noted verbally on the recorder sound track at the time of occurrence.

Before successive test runs are made, quantity of tape and spot checking of transducer outputs will be checked. All events and recording information will be noted on the test and recording test logs.

B. INSTRUMENTATION

The twelve guide shoes will be instrumented to determine contact with the guide. When the guide shoes contact the guide, the small displacement of the shoe will be detected by a microswitch. The guide shoes will be held in place by springs and holding clamps. Two micro-switches, one at either end of the shoe, will activate a logic circuit which indicates contact for the duration of the contact. Either micro-switch or both will activate the logic circuit. Figure XIII-1 illustrates the guide shoe instrumentation. The twenty-four micro-switches will be powered and ganged in pairs to provide the contact, non-contact situation of each of the twelve guide shoes.

The contact force of the twelve guide rollers will be measured by strain gage type load cells which measure the moment about two axes. Due to the design of the roller wheel and arm, the normal force at the guide roller is resolved into a moment at the guide roller arm. Figure XIII-2 shows how a load cell located at this juncture will measure the guide roller force as a moment. Since the arm rotates and the axis of sensitivity changes, the outputs of the mutually perpindicular axis will be added vectorily to measure the actual force.

The lateral and longitudinal acceleration of the skip body will be measured at four locations. The eight piezo-

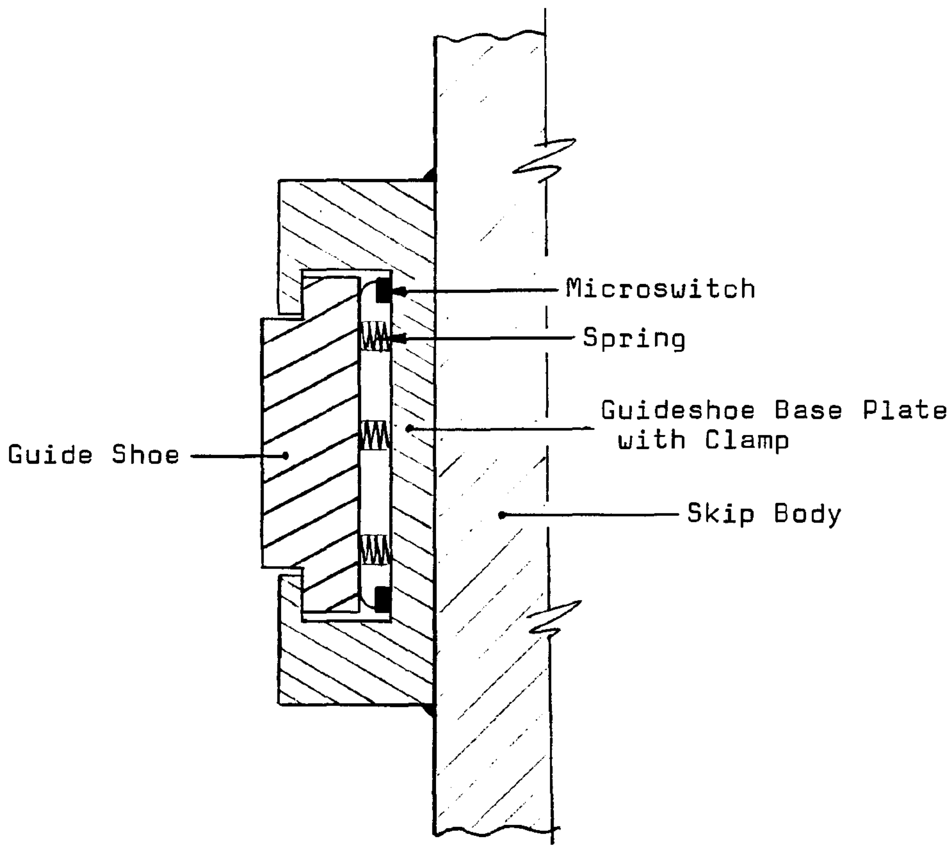


Figure XIII-1 Schematic of Guide Shoe Instrumentation

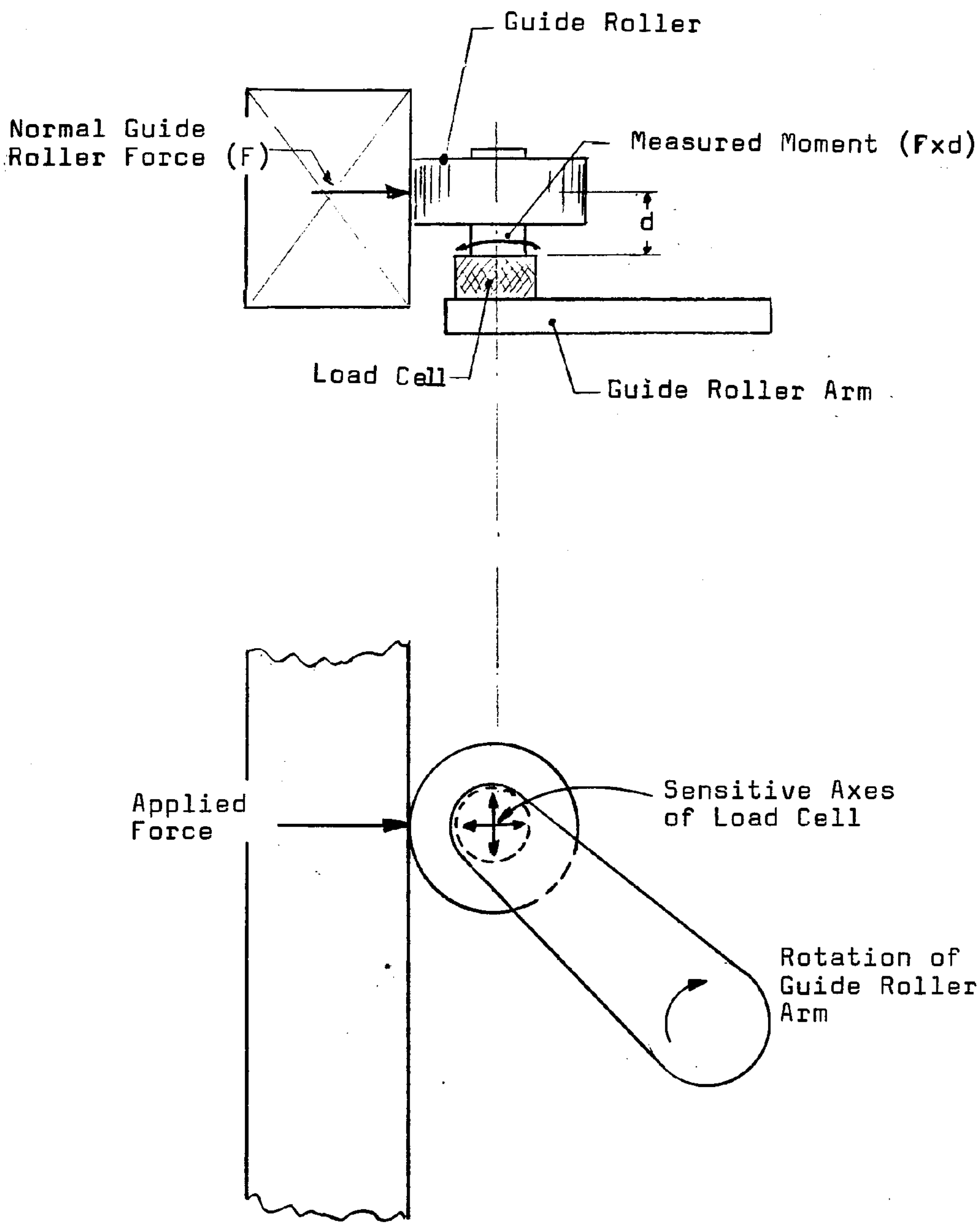


Figure XIII-2 Schematic of Guide Roller Instrumentation electric transducers will be mounted to flat plate brackets fastened to the skip. Each of the transducer signals will be conditioned by an appropriate strain gage type amplifier. The signal will then be split and recorded as is and double integrated to yield displacement and subsequent recordings. Standard operational amplifiers will be used for the integration process. Figure XIII-3 illustrates the path of the accelerometer signal.

The position and speed of the skip in the shaft will be measured by an optical encoder attached to a trailing fifth wheel. The encoder output of pulses per unit distance will be accumulated to determine position and differentiated with respect to time to determine speed.

The net resultant of this set of instrumentation system will be forty-two data channels suitable for recording. To review, the twenty-four guide shoe micro-switches will yield twelve outputs which indicate contact or non-contact between the guide and guide shoe. Twelve load cells will have twenty-four outputs that will be combined vectorily to indicate the force at the guide roller interface of the twelve guide rollers.

The output of eight accelerometers mounted to the skip body will indicate the acceleration at those locations. The double integration of those accelerometer signals will yield the inertial displacement of those locations.

The raw output of an optical encoder is used to determine the vertical position of the skip in the shaft. This same output differentiated with respect to time will yeild the speed of the skip.

Table XIII-3 reviews the instrumentation and the parameters to be measured.

All of the transducers will be electronically calibrated or checked for proper operation before and after each day's testing. The load cells will be calibrated by a shunt calibration which switches a known resistance into a leg of the strain gage bridge. The signal conditioning can then be adjusted for desired output. Each of the load cell moment axes will be shunt calibrated at eight points. The load cell calibration sheet will be used to record the output voltages.

The piezo accelerometers will also be calibrated using a shunt calibration. Only four points will be examined, as given on the accelerometer calibration sheet.

The remaining transducers will be checked for proper operation. The micro-switches will be pressed to determine if they are operating properly. The zero offsets

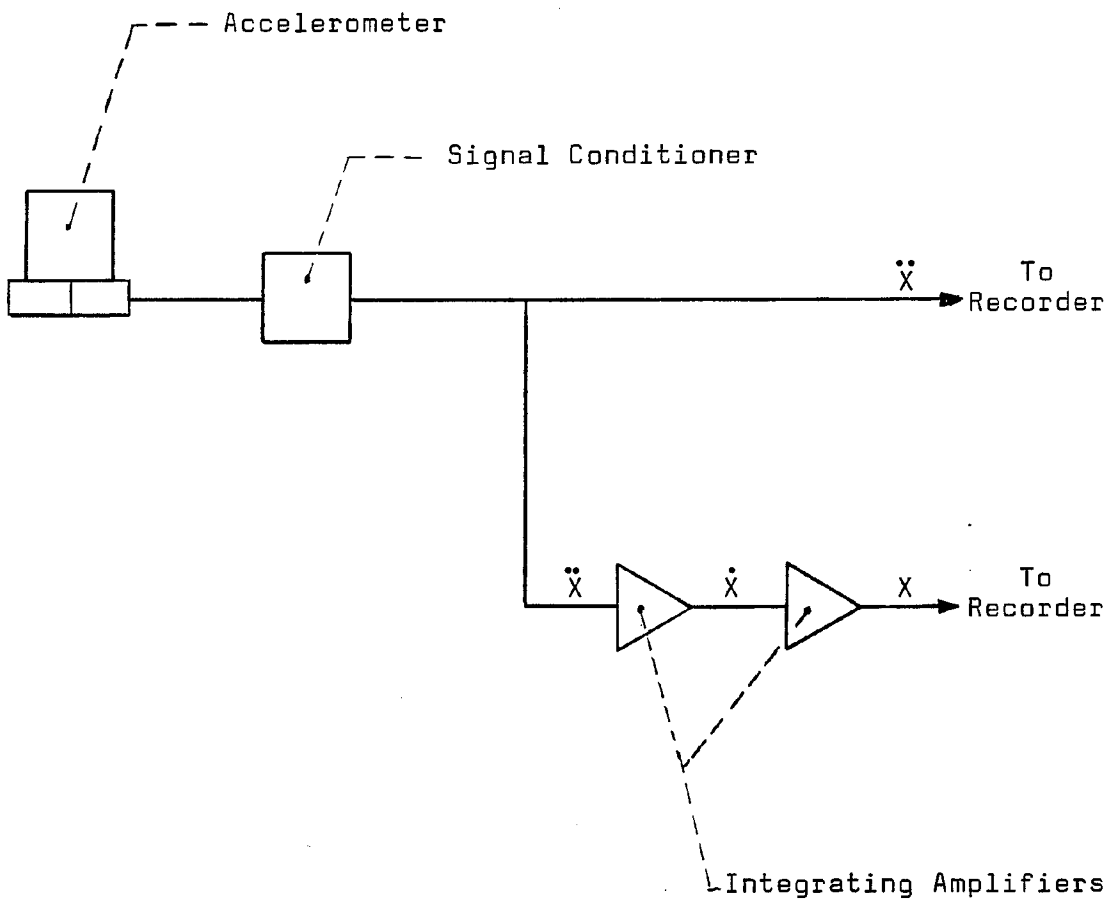


Figure XIII-3 Schematic of Accelerometer Signal Path

Table XIII-3 Description of Instrumentation

CH. NO.	PARAMETER	TRANSDUCERS	SPECIAL CONDITIONING
1-12	Guide - Guide Shoe Contact (12 Locations)	Microswitches (2 Per Location)	Either/or Logic
13-24	Guide Roller Force (12 Locations)	Strain-gage Load Cell Measuring Two Moments at Each Location	Vector Sum of Two Moments Measured at Each Location
25-28	Skip Body Acceleration Lateral (4 Locations)	Piezoelectric Accelerometer	Standard
29-32	Skip Body Acceleration Longitudinal (4 Locations)	Piezoelectric Accelerometer	Standard
33-36	Skip Body Displacement Lateral (4 Locations)	Piezoelectric Accelerometer	Double Integration of Lateral Body Accelerometer Outputs
37-40	Skip Body Displacement Longitudinal (4 Locations)	Pizeoelectric Accelerometer	Double Integration of Longitudinal Body Accelerometer Outputs
41	Skip Body Position	Optical Encoder	Accumulating Register
42	Skip Body Speed	Optical Encoder	Derivative of Position Signal WRT Time

of the skip body displacement double integrating amplifiers will be set to zero. The tachometer fifth wheel will be rotated once to determine that the proper number of counts is being registered. The tachometer wheel will then be spun to be sure that the speed electronics are working properly. All of these items will be checked off on the calibration checklist sheet.

At the beginning and completion of the entire test program all transducers will be calibrated by applying known physical inputs such as force or acceleration. This task will be performed by an acceptable testing laboratory whose measurement standards are traceable to National Bureau of Standards.

C. DATA RECORDING AND REDUCTION

The data will be recorded using a frequency modulated (F.M.) tape recorder having at least four channels. The first forty data channels, guide shoe contact, guide roller force, skip body acceleration and displacement will be multiplexed to form a single recorded channel. The skip position and speed data will be recorded on two separate recording channels. The fourth channel will be used as a voice record of specific events taking place.

The recorder as well as the signal conditioning will be located to the skip body by a shock mounted platform. The recorder and all electronics will be powered by a portable, rechargeable battery pack.

The recorder will use one-half-inch tape and operate at a speed of 15/16 inches per second.

A series of logs and check sheets are provided for recording portions of the test data. Table XIII-4 is a log of all tests. Table XIII-5 records the data on each tape. A checklist prior to a test is given in Table XIII-6. Load cell and accelerometer calibrations are shown in Tables XIII-7 and -8, respectively.

The data from each channel will be played back from the tape recorder through the multiplexer and displayed on an X-Y plotter. The data will appear on the Y-axis and elapsed time on the X-axis. If desired, distance can be played out on the X-axis instead of time. The data plot versus time is similar to the dynamic model output and therefore direct comparison is possible. Also, a power spectral density plot of the accelerometer data will be generated by feeding the played back data into a spectral analyzer. This information can be used to determine the characteristic frequency content of the skip body accelerations and those frequency components containing the most energy.

Table XIII-4 TEST LOG

TEST TITLE/DESCRIPTION ______________________________

PURPOSE OF TEST ______________________ DATE __________

GEOGRAPHIC TEST AREA ______________________________

TEST DIRECTOR ____________________ TEST DAY NO. __________

RECORDED BY____________________

TIME	SERIES NO.	RUN NO.	SPD	REMARKS

Table XIII-5 TEST RECORDING LOG

TEST TITLE/DESCRIP____________________ DATE__________

TEST DIRECTOR____________________ TAPE NO. __________

RECORDED BY ____________________ REEL _____ OF __________

TEST SERIES NUMBER	TEST RUN NUMBER	SPEED	BEGIN FOOTAGE	END FOOTAGE	REMARKS

Table XIII-6 TEST AND CALIBRATION CHECKLIST SHEET

TEST SERIES ____________________ DATE ____________________

PERFORMED BY ____

1. __________ All load cells calibrated.
   Remarks: ______________________________________

2. __________ All accelerometers calibrated.
   Remarks: ______________________________________

3. __________ All displacement integrating amplifiers set to 0.00.
   Remarks: ______________________________________

4. __________ All microswitches operating properly.
   Remarks: ______________________________________

5. __________ Number of tach pulses per revolution(1000).
   Remarks: ______________________________________

6. __________ Speedometer working properly.
   Remarks: ______________________________________

7. __________ Recording system operational.
   Remarks: ______________________________________

Table XIII-7 LOAD CELL CALIBRATION SHEET

TEST SERIES____________ DATE______________ PERFORMED BY___________________

CH / SER	VOLTAGE	AMP. ZERO	OUTPUT ZERO	1/4 SCALE		1/2 SCALE		3/4 SCALE		FULL SCALE	
					+		+		+		+
					−		−		−		−
					+		+		+		+
					−		−		−		−
					+		+		+		+
					−		−		−		−
					+		+		+		+
					−		−		−		−
					+		+		+		+
					−		−		−		−
					+		+		+		+
					−		−		−		−
					+		+		+		+
					−		−		−		−
					+		+		+		+
					−		−		−		−
					+		+		+		+
					−		−		−		−
					+		+		+		+
					−		−		−		−
					+		+		+		+
					−		−		−		−
					+		+		+		+
					−		−		−		−

Table XIII-8 ACCELEROMETER CALIBRATION SHEET

TEST SERIES__________ DATE __________ PERFORMED BY__________

CH.	SER.	VOLTAGE	AMP. ZERO	OUTPUT ZERO	1/2 SCALE		FULL SCALE	
						+		+
						−		−
						+		+
						−		−
						+		+
						−		−
						+		+
						−		−
						+		+
						−		−
						+		+
						−		−
						+		+
						−		−
						+		+
						−		−

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APPENDIX A

LISTING OF "SKIP" COMPUTER PROGRAM

```
      PROGRAM SKIP(TAPE1=512, INPUT=256, OUTPUT=512, TAPE2=INPUT,
     1             TAPE3=OUTPUT)
      REAL KR,KX(4),KZ(4)
      REAL KX11,KX22,KX33,KX44,KX13,KX24,KZ11,KZ22,KZ33,KZ44,KZ13,KZ24
      DIMENSION Y(10,1),DY(10,1),Y1(10,1),Y2(10,1),Y3(10,1)
      DIMENSION ZO(100),XPO(100),X(4),Z(4),XP(4),ZP(4),FX(4),FZ(4),
     1C(5,3),XB(4),ZB(4),ZII(4),XII(4),  IZ(4),AX(4),AZ(4),YZ(6),YX(6),
     1IX(4)
      COMMON/A/XPOL(100),ZOL(100)
      COMMON Y,DY,KT,HH,XX,N,JQ, IMAX,IFREQ,XOUT,MQ
      COMMON X1,X2,X3,Y1,Y2,Y3
      COMMON D, D2,A1,A2,B1,B2,X,Z,XP,ZP,FX,FZ,KX,KZ,XB,ZB
      COMMON WT,XBAR,YBAR,ZBAR,HX,HY,HZ,TORS,V,AM,AIX,AIZ,YCM,AIY,XOO
      COMMON CR,KR,NBUNT,FL,E,GI,GI2,ZO,XPO,LLL,YCMP,IX,IY,IZ,ZLIM,XLIM
      COMMON DZ,DPHI,DX,DT,DPSI,AX,AZ,YZ,YX
      COMMON F1,F2,F3,F4,F5,C
      KT=1
      N=10
      IN=2
      IOUT=3
      LLL=1
      READ(IN,1)F
    1 FORMAT(A6)
      READ(IN,2)AIX,AIY,AIZ
    2 FORMAT(3F20.7)
      READ(IN,3)WT,XBAR,YBAR,ZBAR,HX,HY,HZ,CR,KR,TORS
    3 FORMAT(F20.7,3F10.3/3F10.3/3F20.5)
      READ(IN,4)FL,NBUNT,E,GI,GI2,ZLIM,XLIM
    4 FORMAT(F10.3,I5,E20.7,2F20.7/2F10.7)
      DO 9 LB = 1,NBUNT
      XPOL(LB)=0.0
      ZOL(LB)=0.0
      XPO(LB) = 0.0
    9 ZO (LB) = 0.0
      READ (IN,10) KBUNT
   10 FORMAT(I5, 4F10.5)
      IF(KBUNT .LE. 0) GO TO 13
      DO 12 LB = 1,KBUNT
      READ  (IN,10) LC, XPO(LC), ZO(LC), XPOL(LC), ZOL(LC)
   12 CONTINUE
   13 CONTINUE
      READ(IN,6)TI,T,D,D2,V,I
    6 FORMAT(F10.5,F20.5/2E20.7/F20.7/I3)
      WRITE(IOUT,7)I,F,WT,HX,HY,HZ,XBAR,YBAR,ZBAR,CR,KR,TORS
    7 FORMAT(1H1,15X,≠RUN NUMBER ≠,I3/5X,≠SKIP IS ≠2X,A6/5X,
     1≠WEIGHT=≠2X,E15.7,2X,≠LBS,≠5X≠HX= ≠F7.3≠HY= ≠F7.3≠HZ= ≠,
     2F7.3/5X,≠OFF-CENTER OF CM≠,2X≠XBAR=≠,F7.3,≠ YBAR=≠,F7.3,
     32X,≠ZBAR=≠,F7.3/5X,≠ROLLER STIFFNESSES≠,2X,≠Z-≠,F10.3,2X,≠LBS/IN X
     1-≠,F10.3,2X,≠LBS/IN≠,5X≠APPLIED ROPE TORQUE≠,2X,F10.3)
      WRITE(IOUT,8)FL,E,GI,GI2,NBUNT
    8 FORMAT(5X,≠DIST. BETW. BUNTONS ≠,F7.3,≠ E=≠,E10.3,≠ IX=≠,F7.3,
     1≠IZ=≠,F7.3,≠ NUMBER OF BUNTONS≠,I5)
      E=E*144.
      GI=GI/20736.
      GI2=GI2/20736.
      CR=CR*12.
      KR=KR*12.
      V=V/60.
      J=T/TI
```

```
      AM=WT/32.2
      YINC=V*TI
      A1=HZ/2.-ZBAR
      A2=HZ/2.+ZBAR
      B1=HY/2.-YBAR
      B2=HY/2.+YBAR
      YCM=B2
      YCMP=YCM
      XX=0.
      XOO=XX
      CALL GUIDE
      YYY=YZ(1)*YZ(3)-YZ(5)*YZ(5)
      KZ33=YZ(3)/YYY
      KZ13=YZ(5)/YYY
      KZ11=YZ(1)/YYY
      YYY=YZ(4)*YZ(2)-YZ(6)*YZ(6)
      KZ44=YZ(4)/YYY
      KZ24=YZ(6)/YYY
      KZ22=YZ(2)/YYY
      YYY=YX(1)*YX(3)-YX(5)*YX(5)
      KX33=YX(3)/YYY
      KX13=YX(5)/YYY
      KX11=YX(1)/YYY
      YYY=YX(4)*YX(2)-YX(6)*YX(6)
      KX44=YX(4)/YYY
      KX24=YX(6)/YYY
      KX22=YX(2)/YYY
      C(1,1)=KX33+KX11+KX22+KX44-2.*KX13-2.*KX24
      C(1,2)=A1*(KX33+KX11-2.*KX13)+A2*(2.*KX24-KX44-KX22)
      C(1,3)=B1*(KX13+KX24-KX33-KX44)+B2*(KX11+KX22-KX24-KX13)
      Q1=XB(1)*(KX33-KX13)+XB(2)*(KX44-KX24)+XB(3)*(KX11-KX13)
     1+XB(4)*(KX22-KX24)
      C(2,1)=KZ11+KZ22+KZ33+KZ44-2.*KZ13-2.*KZ24
      C(2,2)=B1*(KZ33+KZ44-KZ13-KZ24)-B2*(KZ11+KZ22-KZ13-KZ24)
      Q2=ZB(3)*(KZ11-KZ13)+ZB(1)*(KZ33-KZ13)+ZB(2)*(KZ44-KZ24)+ZB(4)*
     1(KZ22-KZ24)
      C(3,1)=KX33+KX11-KX44-KX22-2.*KX13+2.*KX24
      C(3,2)=A1*(KX33+KX11-2.*KX13)+A2*(KX22+KX44-2.*KX24)
      C(3,3)=B2*(KX11-KX22-KX13+KX24)-B1*(KX33-KX44-KX13+KX24)
      Q3=XB(1)*(KX33-KX13)+XB(2)*(KX24-KX44)+XB(3)*(KX11-KX13)
     1+XB(4)*(KX24-KX22)
      C(4,1)=KX11+KX22-KX33-KX44
      C(4,2)=A1*(KX11-KX33)-A2*(KX22-KX44)
      C(4,3)=B1*(KX33+KX44+KX13+KX24)+B2*(KX11+KX22+KX13+KX24)
      Q4=XB(3)*(KX11+KX13)+XB(4)*(KX22+KX24)-XB(1)*(KX33+KX13)
     1-XB(2)*(KX44+KX24)
      C(5,1)=KZ33+KZ44-KZ11-KZ22
      C(5,2)=B1*(KZ33+KZ44+KZ13+KZ24)+B2*(KZ11+KZ22+KZ13+KZ24)
      Q5=ZB(1)*(KZ33+KZ13)+ZB(2)*(KZ44+KZ24)-ZB(3)*(KZ11+KZ13)
     1-ZB(4)*(KZ22+KZ24)
      DET=C(1,1)*(C(3,2)*C(4,3)-C(3,3)*C(4,2))+C(1,2)*(C(3,3)*C(4,1)
     1-C(3,1)*C(4,3))+C(1,3)*(C(3,1)*C(4,2)-C(3,2)*C(4,1))
      Y(1,1)=(C(5,2)*Q2-C(2,2)*(Q5+WT*ZBAR*2./HY))/
     1(C(2,1)*C(5,2)-C(2,2)*C(5,1))
      Y(2,1)=0.
      Y(3,1)=(C(2,1)*(Q5+WT*ZBAR*2./HY)-C(5,1)*Q2)/
     1(C(2,1)*C(5,2)-C(2,2)*C(5,1))
      Y(4,1)=0.
      Y(5,1)=Q1*(C(3,3)*C(4,1)-C(3,1)*C(4,3))+(Q3+2*TORS/HZ)*
```

```
   1(C(1,1)*C(4,3)-C(1,3)*C(4,1))+(Q4-2.*WT*XBAR/HY)*
   2(C(3,1)*C(1,3)-C(3,3)*C(1,1))
    Y(6,1)=0.
    Y(7,1)=Q1*(C(3,2)*C(4,3)-C(3,3)*C(4,2))+(Q3+2.*TORS/HZ)*
   1(C(1,3)*C(4,2)-C(1,2)*C(4,3))+(Q4-2.*WT*XBAR/HY)*
   2(C(1,2)*C(3,3)-C(1,3)*C(3,2))
    Y(8,1)=0.
    Y(9,1)=Q1*(C(3,1)*C(4,2)-C(4,1)*C(3,2))+(Q3+2.*TORS/HZ)*
   1(C(4,1)*C(1,2)-C(4,2)*C(1,1))+(Q4-2.*WT*XBAR/HY)*
   2(C(1,1)*C(3,2)-C(1,2)*C(3,1))
    Y(10,1)=0.
    Y(5,1)=Y(5,1)/DET
    Y(7,1)=Y(7,1)/DET
    Y(9,1)=Y(9,1)/DET
    CALL XANDZ
    CALL GUIDE
    CALL FORCE
    DO 19JJ=1,4
    XII(JJ)=X(JJ)*12.
 19 ZII(JJ)=Z(JJ)*12.
    WRITE(IOUT,20)
 20 FORMAT(//2X,≠TIME≠,7X≠Z1/FZ1≠6X≠Z2/FZ2≠6X≠Z3/FZ3≠6X≠Z4/FZ4≠6X≠X1/F
   1X1≠6X≠X2/FX2≠,6X≠X3/FX3≠6X≠X4/FX4≠6X≠Y≠)
    WRITE(IOUT,21)XX,(ZII(JJ),JJ=1,4),(XII(JJ),JJ=1,4),YCM,
   1(FZ(JJ),JJ=1,4),(FX(JJ),JJ=1,4)
 21 FORMAT(/2X,F7.3,8(3X,F9.4),3X,F8.2/9X,8(2X,F10.3))
    NVAR = 14
    WRITE (1) NVAR, J
    WRITE (1) XX,  Y(1), Y(3), Y(5), Y(7), Y(9), FZ, FX
    XI=XX-TI
    DO 50 II=1,J
 14 XI=XI+TI
    XOUT=XI+TI
    XX=XI
    XOO=XX
    HH=.05*TI
    IFREQ=3
    J9=1
    M9=1
    JMAX=1
    CALL RNKUT
    CALL XANDZ
    CALL GUIDE
    CALL FORCE
    DO 30 JJ=1,4
    XII(JJ)=X(JJ)*12.
 30 ZII(JJ)=Z(JJ)*12.
    YCM=YCM+YINC
    WRITE(IOUT,21)XX,(ZII(JJ),JJ=1,4),(XII(JJ),JJ=1,4),YCM,
   1(FZ(JJ),JJ=1,4),(FX(JJ),JJ=1,4)
    WRITE (1) XX,  Y(1), Y(3), Y(5), Y(7), Y(9), FZ, FX
    DO 36 JJ=1,4
    IF(IZ(JJ))33,33,31
 31 WRITE(IOUT,32)JJ
 32 FORMAT(10X,≠Z(≠,I1,≠)GUIDE SHOE CONTACT≠)
 33 IF(IX(JJ))36,36,34
 34 WRITE(IOUT,35)JJ
 35 FORMAT(10X,≠X(≠,I1,≠)GUIDE SHOE CONTACT≠)
 36 CONTINUE
```

```
C
C     IF Z, PHI OR X TOO LARGE, STOP.
      Y8=ABS(Y(1,1))-.1667
      IF(Y8)23,23,29
   29 WRITE(IOUT,22)
   22 FORMAT(15X,≠Z TOO LARGE≠)
      GO TO 26
   23 Y8=ABS(Y(3,1))-.1
      IF(Y8)43,43,24
   24 WRITE(IOUT,25)
   25 FORMAT(15X,≠PHI TOO LARGE≠)
      GO TO 26
   43 Y8=ABS(Y(7,1))-.1667
      IF(Y8)50,50,27
   27 WRITE(IOUT,28)
   28 FORMAT(15X,≠X TOO LARGE≠)
      GO TO 26
   50 CONTINUE
   26 CALL EXIT
      END
      SUBROUTINE DIFEQ
C
C     THIS SUBROUTINE CALCULATES ALL NEW VALUES OF PARAMETERS
C     NECESSARY TO CALCULATE NEW ≠DY≠ ARRAY VALUES.
      REAL KR,KX(4),KZ(4)
      DIMENSION Y(10,1),DY(10,1),Y1(10,1),Y2(10,1),Y3(10,1)
      DIMENSION ZO(100),XPO(100),X(4),Z(4),XP(4),ZP(4),FX(4),FZ(4),
     1C(5,3),      XB(4),ZB(4),       IX(4),IZ(4),AX(4),AZ(4),YZ(6),YX(6)
      COMMON Y,DY,KT,HH,XX,N,JQ, JMAX,IFREQ,XOUT,MP
      COMMON X1,X2,X3,Y1,Y2,Y3
      COMMON D, D2,A1,A2,B1,B2,X,Z,XP,ZP,FX,FZ,KX,KZ,XB,ZB
      COMMON WT,XBAR,YBAR,ZBAR,HX,HY,HZ,TORS,V,AM,AIX,AIZ,YCM,AIY,XOO
      COMMON CR,KR,NBUNT,EL,E,GI,GI2,ZO,XPO,LLL,YCMP,IX,IY,IZ,ZLIM,XLIM
      COMMON DZ,DPHI,DX,DT,DPSI,AX,AZ,YZ,YX
      COMMON F1,F2,F3,F4,F5,C
      CALL XANDZ
      CALL GUIDE
      CALL FORCE
      DY(1,1)=Y(2,1)
      DY(2,1)=F2/AM
      DY(3,1)=Y(4,1)
      DY(4,1)=-AM*YBAR*DY(2,1)/AIX+WT*ZBAR/AIX+HY*F5/(2.*AIX)
      DY(8,1)=F1/AM
      DY(5,1)=Y(6,1)
      DY(6,1)=TORS/AIY+AM*(XBAR*DY(2,1)-ZBAR*DY(8,1))/AIY+HZ*F3/(2.*AIY)
      DY(7,1)=Y(8,1)
      DY(9,1)=Y(10,1)
      DY(10,1)=AM*YBAR*DY(8,1)/AIZ-WT*XBAR/AIZ+HY*F4/(2.*AIZ)
      RETURN
      END
      SUBROUTINE XANDZ
      REAL KR,KX(4),KZ(4)
      DIMENSION Y(10,1),DY(10,1),Y1(10,1),Y2(10,1),Y3(10,1)
      DIMENSION ZO(100),XPO(100),X(4),Z(4),XP(4),ZP(4),FX(4),FZ(4),
     1C(5,3),      XB(4),ZB(4),       IX(4),IZ(4),AX(4),AZ(4),YZ(6),YX(6)
      COMMON Y,DY,KT,HH,XX,N,JQ, JMAX,IFREQ,XOUT,MP
      COMMON X1,X2,X3,Y1,Y2,Y3
      COMMON D, D2,A1,A2,B1,B2,X,Z,XP,ZP,FX,FZ,KX,KZ,XB,ZB
      COMMON WT,XBAR,YBAR,ZBAR,HX,HY,HZ,TORS,V,AM,AIX,AIZ,YCM,AIY,XOO
```

```
      COMMON CR,KR,NBUNT,FL,E,GI,GI2,Z0,XP0,LLL,YCMP,IX,IY,IZ,ZLIM,XLIM
      COMMON DZ,DPHI,DX,DT,DPST,AX,AZ,YZ,YX
      COMMON F1,F2,F3,F4,F5,C
      X(1)=Y(7,1)+A1*Y(5,1)-B1*Y(9,1)
      X(2)=Y(7,1)-A2*Y(5,1)-B1*Y(9,1)
      X(3)=Y(7,1)+A1*Y(5,1)+B2*Y(9,1)
      X(4)=Y(7,1)-A2*Y(5,1)+B2*Y(9,1)
      Z(1)=Y(1,1)+B1*Y(3,1)
      Z(2)=Y(1,1)+B1*Y(3,1)
      Z(3)=Y(1,1)-B2*Y(3,1)
      Z(4)=Y(1,1)-B2*Y(3,1)
      XP(1)=Y(8,1)+A1*Y(6,1)-B1*Y(10,1)
      XP(2)=Y(8,1)-A2*Y(6,1)-B1*Y(10,1)
      XP(3)=Y(8,1)+A1*Y(6,1)+B2*Y(10,1)
      XP(4)=Y(8,1)-A2*Y(6,1)+B2*Y(10,1)
      ZP(1)=Y(2,1)+B1*Y(4,1)
      ZP(2)=Y(2,1)+B1*Y(4,1)
      ZP(3)=Y(2,1)-B2*Y(4,1)
      ZP(4)=Y(2,1)-B2*Y(4,1)
      RETURN
      END
      SUBROUTINE FORCE
      REAL KR,KX(4),KZ(4)
      DIMENSION Y(10,1),DY(10,1),Y1(10,1),Y2(10,1),Y3(10,1)
      DIMENSION Z0(100),XP0(100),X(4),Z(4),XP(4),ZP(4),FX(4),FZ(4),
     1C(5,3),     XB(4),ZB(4),     IX(4),IZ(4),AX(4),AZ(4),YZ(6),YX(6)
      COMMON Y,DY,KT,HH,XX,N,JP, JMAX,IFREQ,XOUT,MP
      COMMON X1,X2,X3,Y1,Y2,Y3
      COMMON D, D2,A1,A2,B1,B2,X,Z,XP,ZP,FX,FZ,KX,KZ,XB,ZB
      COMMON WT,XBAR,YBAR,ZBAR,HX,HY,HZ,TORS,V,AM,AIX,AIZ,YCM,AIY,XDO
      COMMON CR,KR,NBUNT,FL,E,GI,GI2,Z0,XP0,LLL,YCMP,IX,IY,IZ,ZLIM,XLIM
      COMMON DZ,DPHI,DX,DT,DPST,AX,AZ,YZ,YX
      COMMON F1,F2,F3,F4,F5,C
      F1=0.
      F2=0.
      F3=0.
      F4=0.
      F5=0.
      FX(1)=(-(X(1)-XB(1))*YX(3)+(X(3)-XB(3))*YX(5))/
     1(YX(1)*YX(3)-YX(5)*YX(5))
      FX(2)=(-(X(2)-XB(2))*YX(4)+(X(4)-XB(4))*YX(6))/
     1(YX(2)*YX(4)-YX(6)*YX(6))
      FX(3)=((X(1)-XB(1))*YX(5)-(X(3)-XB(3))*YX(1))/
     1(YX(1)*YX(3)-YX(5)*YX(5))
      FX(4)=((X(2)-XB(2))*YX(6)-(X(4)-XB(4))*YX(2))/
     1(YX(2)*YX(4)-YX(6)*YX(6))
      FZ(1)=(-(Z(1)-ZB(1))*YZ(3)+(Z(3)-ZB(3))*YZ(5))/
     1(YZ(1)*YZ(3)-YZ(5)*YZ(5))
      FZ(2)=(-(Z(2)-ZB(2))*YZ(4)+(Z(4)-ZB(4))*YZ(6))/
     1(YZ(2)*YZ(4)-YZ(6)*YZ(6))
      FZ(3)=((Z(1)-ZB(1))*YZ(5)-(Z(3)-ZB(3))*YZ(1))/
     1(YZ(1)*YZ(3)-YZ(5)*YZ(5))
      FZ(4)=((Z(2)-ZB(2))*YZ(6)-(Z(4)-ZB(4))*YZ(2))/
     1(YZ(2)*YZ(4)-YZ(6)*YZ(6))
      DO 1 JJ=1,4
      F1=F1+FX(JJ)-D2*DX*(XP(JJ)-AX(JJ))
    1 F2=F2+FZ(JJ)-D*DZ*(ZP(JJ)-AZ(JJ))
      DO 2 JJ=1,2
      JK=2*JJ-1
```

```
      JL=JJ+2
      F3=F3+FX(JK)-D2*DT*(XP(JK)-AX(JK))
      F4=F4+FX(JL)-D2*DPSI*(XP(JL)-AX(JL))
    2 F5=F5+FZ(JJ)-D*DPHI*(ZP(JJ)-AZ(JJ))
      DO 3 JJ=1,2
      JK=2*JJ
      JL=JJ+2
      F3=F3-FX(JK)+D2*DT*(XP(JK)-AX(JK))
      F4=F4-FX(JJ)+D2*DPSI*(XP(JJ)-AX(JJ))
    3 F5=F5-FZ(JL)+D*DPHI*(ZP(JL)-AZ(JL))
      RETURN
      END

      SUBROUTINE GUIDE
      REAL KBX(4),KBZ(4),KSX(4),KSZ(4)
      REAL KP,KX(4),KZ(4)
      DIMENSION ZO(100),XPO(100),X(4),Z(4),XP(4),ZP(4),FX(4),FZ(4),
     1C(5,3),      XB(4),ZB(4),      IX(4),IZ(4),AX(4),AZ(4),YZ(6),YX(6)
      DIMENSION Y(10,1),DY(10,1),Y11(10,1),Y22(10,1),Y33(10,1)
      COMMON/A/XPOL(100),ZOL(100)
      COMMON Y,DY,KT,HH,XX,N,JP,JMAX,IFREQ,XOUT,MP
      COMMON X1,X2,X3,Y11,Y22,Y33
      COMMON D, D2,A1,A2,B1,B2,X,Z,XP,ZP,FX,FZ,KX,KZ,XB,ZB
      COMMON WT,XBAR,YBAR,ZBAR,HX,HY,HZ,TORS,V,AM,AIX,AIZ,YCM,AIY,X00
      COMMON CR,KR,NBUNT,EL,E,GI,GI2,ZO,XPO,LLL,YCMP,IX,IY,IZ,ZLIM,XLIM
      COMMON DZ,DPHI,DX,DT,DPSI,AX,AZ,YZ,YX
      COMMON F1,F2,F3,F4,F5,C
      TI=XX-X00
      YINCD=TI*V
      YCMG=YCM+YINCD
      YI=YCMG-YBAR
      Y1=YI+HY/2.
      Y2=Y1
      Y3=YI-HY/2.
      Y4=Y3
      IY1=IFIX(Y1/EL)
      Y1D=Y1-IY1*EL
      IY1=IY1+1
      Y2D=Y1D
      IY2=IY1
      IY3=IFIX(Y3/EL)
      Y3D=Y3-IY3*EL
      IY3=IY3+1
      Y4D=Y3D
      IY4=IY3
      AX(1)=(XPO(IY1+1)-XPO(IY1))/EL
      AX(2)=(XPOL(IY2+1)-XPOL(IY2))/EL
      AX(3)=(XPO(IY3+1)-XPO(IY3))/EL
      AX(4)=(XPOL(IY4+1)-XPOL(IY4))/EL
      AZ(1)=(ZO(IY1+1)-ZO(IY1))/EL
      AZ(2)=(ZOL(IY2+1)-ZOL(IY2))/EL
      AZ(3)=(ZO(IY3+1)-ZO(IY3))/EL
      AZ(4)=(ZOL(IY4+1)-ZOL(IY4))/EL
      XB(1)=XPO(IY1)+AX(1)*Y1D
      XB(2)=XPOL(IY2)+AX(2)*Y2D
      XB(3)=XPO(IY3)+AX(3)*Y3D
      XB(4)=XPOL(IY4)+AX(4)*Y4D
      ZB(1)=ZO(IY1)+AZ(1)*Y1D
      ZB(2)=ZOL(IY2)+AZ(2)*Y2D
```

```
      ZB(3)=ZO(IY3)+AZ(3)*Y3D
      ZB(4)=ZOL(IY4)+AZ(4)*Y4D
      DO 1 JJ=1,4
      AX(JJ)=AX(JJ)*V
    1 AZ(JJ)=AZ(JJ)*V
      SI=0.
      IF(IY1-IY3)2,2,3
    2 SI=-1.
    3 CONTINUE
      AB=3*E*GI*FL
      DO 25 IJ=1,4
      IX(IJ)=0
      IZ(IJ)=0
      KSX(IJ)=KR
   25 KSZ(IJ)=CR
      IF(Y1D)30,30,35
   30 KBZ(1)=1.E10
      KBX(1)=1.E10
      GO TO 40
   35 KBZ(1)=AB/(Y1D*Y1D*(EL-Y1D)**2)
      KBX(1)=KBZ(1)*GI2/GI
   40 IF(Y2D)45,45,50
   45 KBZ(2)=1.E10
      KBX(2)=1.E10
      GO TO 55
   50 KBZ(2)=AB/(Y2D*Y2D*(EL-Y2D)**2)
      KBX(2)=KBZ(2)*GI2/GI
   55 IF(Y3D)60,60,65
   60 KBZ(3)=1.E10
      KBX(3)=1.E10
      GO TO  70
   65 KBZ(3)=AB/(Y3D*Y3D*(EL-Y3D)**2)
      KBX(3)=KBZ(3)*GI2/GI
   70 IF(Y4D)75,75,80
   75 KBZ(4)=1.E10
      KBX(4)=1.E10
      GO TO 85
   80 KBZ(4)=AB/(Y4D*Y4D*(EL-Y4D)**2)
      KBX(4)=KBZ(4)*GI2/GI
   85 CONTINUE
      IF(LLL)86,86,87
   86 LLL=0
      IF(ABS(X(1)-XB(1))-XLIM)95,90,90
   90 IX(1)=1
      KSX(1)=1.44E9
   95 IF(ABS(X(2)-XB(2))-XLIM)105,100,100
  100 IX(2)=1
      KSX(2)=1.44E9
  105 IF(ABS(X(3)-XB(3))-XLIM)115,110,110
  110 IX(3)=1
      KSX(3)=1.44E9
  115 IF(ABS(X(4)-XB(4))-XLIM)125,120,120
  120 IX(4)=1
      KSX(4)=1.44E9
  125 IF(ABS(Z(1)-ZB(1))-ZLIM)135,130,130
  130 IZ(1)=1
      KSZ(1)=1.44E9
  135 IF(ABS(Z(2)-ZB(2))-ZLIM)145,140,140
  140 IZ(2)=1
```

```
      KSZ(2)=1.44E9
  145 IF(ABS(Z(3)-ZB(3))-ZLIM)155,150,150
  150 IZ(3)=1
      KSZ(3)=1.44E9
  155 IF(ABS(Z(4)-ZB(4))-ZLIM)87,160,160
  160 IZ(4)=1
      KSZ(4)=1.44E9
   87 CONTINUE
      LLL=0
      YZ(1)=1./KPZ(1)+1./KSZ(1)
      YZ(2)=1./KPZ(2)+1./KSZ(2)
      YZ(3)=1./KPZ(3)+1./KSZ(3)
      YZ(4)=1./KPZ(4)+1./KSZ(4)
      YZ(5)=SI*(FL-Y1D)*Y3D*((FL-Y1D)**2.-EL*EL+Y3D*Y3D)/(AB*2.)
      YZ(6)=SI*(FL-Y2D)*Y4D*((FL-Y2D)**2.-EL*FL+Y4D*Y4D)/(AB*2.)
      YX(1)=1./KPX(1)+1./KSX(1)
      YX(2)=1./KPX(2)+1./KSX(2)
      YX(3)=1./KPX(3)+1./KSX(3)
      YX(4)=1./KPX(4)+1./KSX(4)
      YX(5)=YZ(5)*GI/GI2
      YX(6)=YZ(6)*GI/GI2
      DO 170 IJ=1,4
      KX(IJ)=1./YX(IJ)
  170 KZ(IJ)=1./YZ(IJ)
      DX=2.*SQRT(AM*(KX(1)*KX(2)+KX(3)+KX(4)))
      DZ=2.*SQRT(AM*(KZ(1)+KZ(2)+KZ(3)+KZ(4)))
      DPHI=SQRT(AIX*2.*HY*(B2*(KZ(3)+KZ(4))+B1*(KZ(1)+KZ(2))))
      DT=SQRT(AIY*2.*HZ*(A1*(KX(1)+KX(3))+A2*(KX(2)+KX(4))))
      DPSI=SQRT(AIZ*2.*HY*(B2*(KX(3)+KX(4))+B1*(KX(1)+KX(2))))
      RETURN
      END
      SUBROUTINE RNKUT
      DIMENSION Y(10,1),DY(10,1),Y1(10,1),Y2(10,1),Y3(10,1)
      COMMON Y,DY,KT,HH,X,N,J,JMAX,IFREQ,XOUT,M
      COMMON X1,X2,X3,Y1,Y2,Y3
      INDE9=0
      CALL AJSTP
      IF(J-JMAX)4040,4040,4100
 4040 INDE9=INDE9+1
      CALL INTPL
      IF(J-JMAX)4043,4043,4100
 4043 CALL STEP
      X1=X2
      X2=X3
      X3=X
      DO 4050 K=1,KT
      DO 4050 I=1,N
      Y1(I,K)=Y2(I,K)
      Y2(I,K)=Y3(I,K)
 4050 Y3(I,K)=Y(I,K)
      IF (INDE9 - IFREQ) 4040, 4060, 4060
 4060 INDE9     = 0
      CALL AJSTP
      IF(J-JMAX) 4040,4040,4100
 4100 RETURN
      END
      SUBROUTINE AJSTP
      DIMENSION Y(10,1),DY(10,1),Y1(10,1),Y2(10,1),Y3(10,1)
      COMMON Y,DY,KT,HH,X,N,J,JMAX,IFREQ,XOUT,      M
```

```
      COMMON X1,X2,X3,Y1,Y2,Y3
      KSL=0
      HFACT      = 1.0 E+31
      HFCT1=1.0E+30
      GO TO (2001, 3001), M
3001  H1         = HH
      HH         = 2.0 * HH
      X          = X1
      DO 3000 K=1,KT
      DO 3000 I = 1, N
3000  Y(I,K)=Y1(I,K)
      GO TO 2021
2001  KSL=1
2002  H1         = HH
      XXX        = X
      DO 2000 K=1,KT
      DO 2000 I = 1, N
2000  Y1(I,K)=Y(I,K)
      X1         = X
      CALL INTPL
      IF(J-JMAX) 2003,2003,2120
2003  CALL STEP
      DO 2010 K=1,KT
      DO 2010 I = 1, N
2010  Y2(I,K)=Y(I,K)
      X2         = X
      CALL INTPL
      IF(J-JMAX) 2013,2013,2120
2013  CALL STEP
      DO 2020 K=1,KT
      DO 2020 I = 1, N
      Y3(I,K)=Y(I,K)
2020  Y(I,K)=Y1(I,K)
      X3         = X
      X          = XXX
      HH         = 2.0 * HH
2021  CALL STEP
      DO 2050 K=1,KT
      DO 2050 I = 1, N
      DELY=ABS (Y (I,K)-Y3(I,K))/30.0
      IF(DELY-ABS (Y2(I,K))*1.0E-05)2030,2022,2022
2022  IF(ABS (Y2(I,K)) - 1.0E-04)  2030,2040,2040
2030  HFRST=1.0E+30
      GO TO 2050
2040  HFRST=(ABS(Y2(I,K))*1.0E-5/DELY)**0.2
2050  CONTINUE
2060  HFACT=AMIN1(HFACT,HFRST)
      IF(HFCT1-HFACT)2062,2062,2067
2062  HH         = 2.0 * H1
      GO TO (2002, 2100), M
2067  HH         = H1 * HFACT
      GO TO (2070, 2100), M
2070  IF(KSL) 2095,2095,2080
2080  KSL=0
      IF(ABS (HH)-ABS (H1)) 2082, 2095, 2095
2082  CONTINUE
      DO 2090 K=1,KT
      DO 2090 I = 1, N
2090  Y(I,K)=Y1(I,K)
```

```
      X              = XXX
      GO TO 2002
 2095 KSL=0
      M              = 2
 2100 CONTINUE
      DO 2110 K=1,KT
      DO 2110 I = 1, N
 2110 Y(I,K)=Y3(I,K)
 2120 RETURN
      END
      SUBROUTINE STEP
      DIMENSION Y(10,1),DY(10,1),Y1(10,1),P1(10,1)
      COMMON Y,DY,KT,HH,X,NDE, J ,JMAX,IFREQ,XOUT,       M
      DO 11 K=1,KT
      DO 11 I=1,NDE
   11 Y1(I,K)=Y(I,K)
      X1=X
      CALL DIFEQ
      DO 12 K=1,KT
      DO 12 I=1,NDE
      P1(I,K)=DY(I,K)*HH
   12 Y(I,K)=Y1(I,K)+P1(I,K)*0.5
      X=X1+0.5*HH
      CALL DIFEQ
      DO 13 K=1,KT
      DO 13 I=1,NDE
      P1(I,K)=P1(I,K)+2.0*HH*DY(I,K)
   13 Y(I,K)=Y1(I,K)+0.5*HH*DY(I,K)
      CALL DIFEQ
      DO 14 K=1,KT
      DO 14 I=1,NDE
      P1(I,K)=P1(I,K)+2.0*HH*DY(I,K)
   14 Y(I,K)=Y1(I,K)+HH*DY(I,K)
      X=X1+HH
      CALL DIFEQ
      DO 15 K=1,KT
      DO 15 I=1,NDE
   15 Y(I,K)=Y1(I,K)+(P1(I,K)+DY(I,K)*HH)/6.0
      RETURN
      END
      SUBROUTINE INTPL
      DIMENSION Y(10,1),DY(10,1)
      COMMON Y,DY,KT,HH,X,N   ,J ,JMAX,IFREQ,XOUT,       M
 5004 IF(ABS (XOUT    - X)-ABS (HH)) 5001, 5001, 5020
 5001 HH             = XOUT-X
      CALL STEP
      J              = J + 1
 5020 RETURN
      END
```

APPENDIX B

TABLES OF PROPERTIES OF STEEL

Table B-1 Properties of A-36 Steel

MECHANICAL PROPERTIES:

Y.S.	36 ksi
U.S.	58-80 ksi
Elong. % (in 2 in.)	23%
Red. of Area	
Hardness	
Fatigue Strength	
Toughness	Charpy V-notch 15 ft-lbs at 20°F
Transition Temperature	20°F

PHYSICAL PROPERTIES:

Modulus of Elasticity	29×10^6 psi
Shear Modulus	11×10^6 psi
Poisson's Ratio	0.30
Density	.284 lbs/in^3
Coefficient of Expansion	6.5×10^{-6} in/in/°F (68°F to 212°F)

Chemistry*

C	Mn	P	S	Si	Cu					
.25/.29	.60/1.20	.04 max	.05 max	.00/.30	.00/.20					

HEAT TREATMENT:

WELDABILITY: Good

ATMOSPHERIC CORROSION RESISTANCE:

MACHINABILITY: Good

FORMABILITY: Good

GAS CUTTING: Good

MATERIAL SPECIFICATIONS: ASTM A36

AVAILABILITY: Plate, shapes, bars

GENERAL: A-36 is the most common structural steel used.

*Chemistry given is only general. Actual ranges are dependent on section.

(Refs. VI-16,17,18)

Table B-2. Compositions, Properties, and Producers of High Strength Steels

I. Columbium or Vanadium Group

Name	Producer Code	C	Mn	P	S	Si	Cu Min	Cb Min	V Min	Yield, 1,000 Psi	Tensile, 1,000 Psi	Elongation in 2 In., %	ASTM or SAE Specification No.
		Composition, %								Mechanical Properties			
AWHF-45	AW	(0.07)	(0.75)	(0.01)	(0.02)	(0.04)	Opt	(0.035)	—	45	—	—	A572, J410c
50	AW	(0.07)	(0.75)	(0.01)	(0.02)	(0.04)	Opt	(0.035)	—	50	—	—	
AWX-42	AW	(0.12)	(0.45)	(0.02)	(0.03)	(0.04)	Opt	(0.02)	—	42	60	20(a)	
45	AW	(0.12)	(0.45)	(0.02)	(0.03)	(0.04)	Opt	(0.02)	—	45	60	19(a)	A572
50	AW	(0.14)	(0.50)	(0.02)	(0.03)	(0.04)	Opt	(0.02)	—	50	65	18(a)	
55	AW	(0.16)	(0.55)	(0.02)	(0.03)	(0.04)	Opt	(0.02)	—	55	70	17(a)	
AW-Ten	AW	(0.18)	(0.75)	(0.02)	(0.03)	(0.04)	(0.25)	(0.02)	—	50	70	18(a)	
Armco High Strength C (b) (Plates, shapes, bars)													
C-42	ARM	0.21	1.35	0.04	0.05	0.30	Opt	0.005-0.05(c)		42	60	20(a)	
C-45	ARM	0.22	1.35	0.04	0.05	0.30	Opt	0.005-0.05(c)		45	60	19(a)	
C-50	ARM	0.23	1.35	0.04	0.05	0.30	Opt	0.005-0.05(c)		50	65	18(a)	A572, J410c
C-55	ARM	0.25	1.35	0.04	0.05	0.30	Opt	0.005-0.05(c)		55	70	17(a)	
C-60	ARM	0.26	1.35	0.04	0.05	0.30	Opt	0.005-0.05(c)		60	75	16(a)	
C-65	ARM	0.26	1.35	0.04	0.05	0.30	Opt	0.005-0.05(c)		65	80	15(a)	
Armco CT-50(d)	ARM	0.18	1.15-1.60	0.04	0.05	0.15-0.30	Opt	0.01-0.05	—	50	70	17(a)	—
CT-55(d)	ARM	0.18	1.15-1.60	0.04	0.05	0.15-0.30	Opt	0.01-0.05	—	55	70	17(a)	—
CT-60(d)	ARM	0.18	1.15-1.60	0.04	0.05	0.15-0.30	Opt	0.01-0.05	—	60	75	17(a)	—
CT-65(d)	ARM	0.18	1.15-1.60	0.04	0.05	0.15-0.30	Opt	0.01-0.05	—	65	85	17(a)	—
(Sheet)												HR (e)	
Cb/V45	B	0.22	1.35	0.04	0.05	0.30	Opt	0.005	0.01	45	60	25	
Cb/V50	B	0.23	1.35	0.04	0.05	0.30	Opt	0.005	0.01	50	65	22	
Cb/V55	B	0.25	1.35	0.04	0.05	0.30	Opt	0.005	0.01	55	70	20	A607, J410c
Cb/V60	B	0.26	1.50	0.04	0.05	0.30	Opt	0.005	0.01	60	75	18	
Cb/V65	B	0.26	1.50	0.04	0.05	0.30	Opt	0.005	0.01	65	80	16	
(Plates, shapes, bars)													
V42	B(f)	0.22	1.25	0.04	0.05	0.30(g)	Opt	—	0.02	42	63	20(a)	
V45	B	0.22	1.25	0.04	0.05	0.30	Opt	—	0.02	45	65	19(a)	
V50(h)	B	0.22	1.25	0.04	0.05	0.30	Opt	—	0.02	50	70	18(a)	A572
V55(h)	B	0.25	1.35	0.04	0.05	0.30	Opt	—	0.02	55	70	17(a)	
V60(h)	B	0.25	1.35	0.04	0.05	0.30	Opt	—	0.02	60	75	16(a)	
V65(h)	B	0.22	1.25	0.04	0.05	0.30	Opt	—	0.02	65	80	15(a)	
(Sheet)												HR (e) CR	
INX-45	IN	0.22	1.35	0.04	0.05	0.30	Opt	0.005	0.01	45	60	25 22	
50	IN	0.23	1.35	0.04	0.05	0.30	Opt	0.005	0.01	50	65	22 20	
55	IN	0.25	1.35	0.04	0.05	0.30	Opt	0.005	0.01	55	70	20 18	A607, J410c
60	IN	0.26	1.50	0.04	0.05	0.30	Opt	0.005	0.01	60	75	18 16	
65	IN	0.26	1.50	0.04	0.05	0.30	Opt	0.005	0.01	65	80	16 15	
70	IN	0.26	1.65	0.04	0.05	0.30	Opt	0.005	0.01	70	85	14 14	—
(Plates, shapes, bars)													
INX-42	IN	0.21	1.35	0.04	0.05	0.30	Opt	0.01	0.01	42	60	20(a)	
45	IN	0.22	1.35	0.04	0.05	0.30	Opt	0.01	0.01	45	60	19(a)	
50	IN	0.23	1.35	0.04	0.05	0.30	Opt	0.01	0.01	50	65	18(a)	A572, J410c
55	IN	0.25	1.35	0.04	0.05	0.30	Opt	0.01	0.01	55	70	17(a)	
60	IN	0.26	1.35	0.04	0.05	0.30	Opt	0.01	0.01	60	75	16(a)	
65	IN	0.26	1.35	0.04	0.05	0.30	Opt	0.01	0.01	65	80	15(a)	
70	IN	0.26	1.35	0.04	0.05	0.30	Opt	0.01	0.01	70	85	14(a)	J410c
IC-42	I	0.22	1.35	0.04	0.05	0.30	Opt	0.005	—	42	60	25	
45	I	0.22	1.35	0.04	0.05	0.30	Opt	0.005	—	45	60	25	
50	I	0.23	1.35	0.04	0.05	0.30	Opt	0.005	—	50	65	22	
55	I	0.25	1.35	0.04	0.05	0.30	Opt	0.005	—	55	70	20	A572, A607,
60	I	0.26	1.35	0.04	0.05	0.30	Opt	0.005	—	60	75	18	J410c
IV-42	I	0.22	1.35	0.04	0.05	0.30	Opt	—	0.01	42	60	25	
45	I	0.22	1.35	0.04	0.05	0.30	Opt	—	0.01	45	60	25	
50	I	0.23	1.35	0.04	0.05	0.30	Opt	—	0.01	50	65	22	
ICF-42(i)	I	0.22	1.35	0.04	0.05	0.30	—	0.005	—	42	60	25	—
ICF-45(i)	I	0.22	1.35	0.04	0.05	0.30	—	0.005	—	45	60	25	—
ICF-50(i)	I	0.23	1.35	0.04	0.05	0.30	—	0.005	—	50	65	22	—

Maximum values for composition are listed except where ranges, minimum, or typical values are indicated. Typical values are enclosed in parentheses. Mechanical properties are those of sheet or hot rolled plate up to ½ in. thick and are minimums unless typical is indicated by parentheses.

Atmospheric corrosion resistance for these high-strength steels is compared to that of carbon steel. Example: 2 indicates twice the corrosion resistance of carbon steel.

Producer Code: AW = Alan Wood Steel Co.; ALG = Algoma Steel Corp.; ARM = Armco Steel Corp.; B = Bethlehem Steel Corp.; BW = Babcock & Wilcox Co.; DF = Dominion Foundries & Steel Ltd.; IN = Inland Steel Co.; I = Interlake Inc.; JL = Jones & Laughlin Steel Corp.; J = Earle M. Jorgensen Co.; K = Kaiser Steel Corp.; L = Lukens Steel Co.; McL = McLouth Steel Corp.; N = National Steel Corp.; O = Oregon Steel Mills; P = Phoenix Steel Corp.; R = Republic Steel Corp.; SH = Sharon Steel Corp.; SC = Steel Co. of Canada Ltd.; US = United States Steel Corp.; W = Wisconsin Steel Div., International Harvester Co.; WP = Wheeling-Pittsburgh Steel Corp.; Y = Youngstown Sheet & Tube Co.

Table B-2. (continued)

I. Columbium or Vanadium Group—Continued

Name	Producer Code	C	Mn	P	S	Si	Cu Min	Cb Min (Composition, %)	or V Min	Yield, 1,000 Psi (Mechanical Properties)	Tensile, 1,000 Psi	Elongation in 2 In., %	ASTM or SAE Specification No.
IV-55	I	0.25	1.35	0.04	0.05	0.30	Opt	—	0.01	55	70	20	A572, A607,
60	I	0.26	1.35	0.04	0.05	0.30	Opt	—	0.01	60	75	18	J410c
ILX-42(j)	JL	0.20	1.00	0.04	0.05	0.30	Opt	0.01	0.01	42	57	25	A572, J410c
45(k)	JL	0.20	1.10	0.04	0.05	0.30	Opt	0.01	0.01	45	60	24	
50(k)	JL	0.22	1.20	0.04	0.05	0.30	Opt	0.01	0.01	50	65	22	A572, A607,
55(k)	JL	0.24	1.20	0.04	0.05	0.30	Opt	0.01	0.01	55	70	20	J410c
60(k)	JL	0.25	1.35	0.04	0.05	0.30	Opt	0.01	0.01	60	75	18	
65(k)	JL	0.26	1.50	0.04	0.05	0.30	Opt	0.01	0.01	65	80	16	
70(k)	JL	0.26	1.65	0.04	0.05	0.30	Opt	0.01	0.01	70	85	14	A607, J410c
ILY-50 CC(l)	JL	0.12	0.90	0.04	0.05	0.10	Opt	0.01	—	50	65	26	A572, A607, J410c
VAN-50(m)	JL	0.14	1.25	0.03	0.03	0.30	—	0.01	0.02	50	65	25	A572, A607,
VAN-60(m)	JL	0.16	1.40	0.03	0.03	0.30	—	0.01	0.02	60	75	22	J410c
VAN-70(m)	JL	0.18	1.50	0.03	0.03	0.30	—	0.01	0.02	70	85	20	A607, J410c
VAN-80(m)	JL	0.18	1.60	0.03	0.03	0.60	—	—	0.05	80	95	18	A656, J410c
Kaisaloy													
42-CV	K	0.21	1.35	0.04	0.05	0.30	Opt	0.005-0.050	0.01-0.10	42	60	24	
45-CV	K	0.22	1.35	0.04	0.05	0.30	Opt	0.005-0.050	0.01-0.10	45	60	22	
50-CV	K	0.23	1.35	0.04	0.05	0.30	Opt	0.005-0.050	0.01-0.10	50	65	21	A572, A607
55-CV	K	0.25	1.35	0.04	0.05	0.30	Opt	0.005-0.050	0.01-0.10	55	70	20	J410c
60-CV	K	0.26	1.35	0.04	0.05	0.30	Opt	0.005-0.050	0.01-0.10	60	75	18	
65-CV	K	0.26	1.35	0.04	0.05	0.30	Opt	0.005-0.050	0.01-0.10	65	80	15(a)	
UCV-65	L	0.23	1.65	0.04	0.05	0.30	—	—	0.01-0.15	65	80	15(a)	—
MLX-45	McL	0.15	1.00	0.04	0.05	0.10	—	0.005	0.02	45	60	25	
50	McL	0.20	1.00	0.04	0.05	0.10	—	0.005	0.02	50	65	22	A572
55	McL	0.24	1.20	0.04	0.05	0.30	—	0.005	0.02	55	70	22	
GLX-42W (n)	N	0.21	1.35	0.04	0.05	—	Opt	0.01	—	42	60	24	—
45W (n)	N	0.22	1.35	0.04	0.05	—	Opt	0.01	—	45	60	22	
50W (n)	N	0.22	1.35	0.04	0.05	—	Opt	0.01	—	50	65	22	A572, A607,
55W (n)	N	0.25	1.35	0.04	0.05	—	Opt	0.01	—	55	70	20	J410c
60W (n)	N	0.26	1.35	0.04	0.05	—	Opt	0.01	—	60	75	18	
65W (n)	N	0.26	1.35	0.04	0.05	—	Opt	0.01	—	65	80	16	
70W (n)	N	0.26	1.35	0.04	0.05	—	Opt	0.01	—	70	85	14	—
Hi-Yield C-42	N	0.21	0.90	0.04	0.05	—	—	0.005	—	42	60	24	
C-45	N	0.22	1.25	0.04	0.05	—	—	0.005	—	45	60	22	A572, A607,
C-50	N	0.22	1.25	0.04	0.05	—	—	0.005	—	50	65	20	J410c
C-55	N	0.25	1.35	0.04	0.05	—	—	0.005	—	55	70	18	
NAPAC-35	N	0.10	0.55	0.025	0.03	—	—	0.005(o)	—	35	—	—	—
40	N	0.10	0.55	0.025	0.03	—	—	0.005(o)	—	40	—	—	—
45	N	0.10	0.55	0.025	0.03	—	—	0.01(o)	—	45	—	—	—
50	N	0.10	0.55	0.025	0.03	—	—	0.01(o)	—	50	—	—	—
NAPAC-F-40	N	0.10	0.65	0.025	0.03	—	—	0.01(o, p)	—	40	50(q)	30	
F-45	N	0.12	0.75	0.025	0.03	—	—	0.01(o, p)	—	45	55(q)	28	
F-50	N	0.12	0.75	0.025	0.03	—	—	0.015(o, p)	—	50	60(q)	26	
F-55	N	0.12	0.75	0.025	0.03	—	—	0.02(o, p)	—	55	65(q)	24	J410c
F-60	N	0.12	0.75	0.025	0.03	—	—	0.03(o, p)	—	60	70(q)	22	
NAPAC S-45	N	0.15	0.75	0.025	0.03	—	—	0.01(o)	—	45	60	25	
S-50	N	0.15	0.75	0.025	0.03	—	—	0.01(o)	—	50	65	22	
Orelloy-42	O	0.15	0.75	0.011	0.013	0.18	0.09	—	0.02	42	60	24	
45	O	0.18	0.92	0.006	0.012	0.26	0.16	—	0.02	45	60	22	
50	O	0.18	1.23	0.007	0.006	0.27	0.27	—	0.02	50	65	21	A572
55	O	0.21	1.20	0.009	0.014	0.24	0.10	—	0.02	55	70	20	
60	O	0.23	1.32	0.015	0.017	0.24	0.14	—	0.02	60	75	18	
PX SK42	P	0.27	1.20	0.04	0.05	—	Opt	—	—	42	60	19(a)	
A42	P	0.21	1.35	0.04	0.05	0.30	Opt	—	0.02	42	60	20(a)	
A45	P	0.22	1.25	0.05	0.04	0.30	Opt	—	0.02	45	65	19(a)	A572
A50	P	0.26	1.30	0.04	0.05	0.30	Opt	—	0.02	50	70	18(a)	
A55	P	0.26	1.30	0.04	0.05	0.30	Opt	—	0.02	55	70	17(a)	
A60	P	0.26	1.35	0.04	0.05	0.30	Opt	—	0.02	60	75	18	
A65	P	0.26	1.35	0.04	0.05	0.30	Opt	—	0.02	65	80	15	
X42W	R	0.21	1.25	0.04	0.05	0.30	Opt	0.01	0.01	42	60	24	
X45W	R	0.22	1.25	0.04	0.05	0.30	Opt	0.01	0.01	45	60	22	
X50W	R	0.22	1.35	0.04	0.05	0.30	Opt	0.01	0.01	50	65	22	A572, A607,
X55W	R	0.25	1.35	0.04	0.05	0.30	Opt	0.01	0.01	55	70	20	J410c
X60W	R	0.26	1.35	0.04	0.05	0.30	Opt	0.01	0.01	60	75	18	
X65W	R	0.26	1.35	0.04	0.05	0.30	Opt	0.01	0.01	65	80	16	
X70W	R	0.26	1.65	0.04	0.05	0.30	Opt	0.01	0.01	70	85	14	J410c
Maxi-Form 50	R	0.12	0.90	0.015	0.03	—	Opt	0.01(r)	—	50	60	25	—
80	R	0.09	1.60	0.015	0.03	0.60	Opt	0.06-0.15(r)	0.08	80	90	18	—
Sharalloy 45	SH	0.16	0.90	0.04	0.05	0.30	—	0.01	—	45	60	25	
50	SH	0.18	0.90	0.04	0.05	0.30	—	0.01	—	50	65	22	J410c
55	SH	0.18	0.90	0.04	0.05	0.30	—	0.01	—	55	70	22	
60	SH	0.23	0.90	0.04	0.05	0.30	—	0.01	—	60	75	20	

Table B-2. (continued)

I. Columbium or Vanadium Group—Continued

Name	Producer Code	C	Mn	P	S	Si	Cu Min	Cb Min	V Min	Yield, 1,000 Psi	Tensile, 1,000 Psi	Elongation in 2 In., %	ASTM or SAE Specification No.
		Composition, %								Mechanical Properties			
H45	US	0.20	1.00	0.04	0.05	0.30	Opt	0.01	—	45	60	25	
L45	US	0.13	0.90	0.04	0.05	0.10	Opt	0.01	—	45	55	25	
K45	US	0.13	0.90	0.04	0.05	0.10	Opt	0.01	—	45	55	26	
F45	US	0.13	0.90	0.04	0.05	0.10	Opt	0.01	—	45	55	26	
H50	US	0.20	1.00	0.04	0.05	0.30	Opt	0.01	—	50	65	22	
L50	US	0.13	0.90	0.04	0.05	0.10	Opt	0.01	—	50	60	24	
K50 (sheet)	US	0.13	0.90	0.04	0.05	0.10	Opt	0.01	—	50	60	25	J410c
F50	US	0.13	0.90	0.04	0.05	0.10	Opt	0.01	—	50	60	25	
H55	US	0.20	1.25	0.04	0.05	0.30	Opt	0.01	—	55	70	20	
H60	US	0.22	1.25	0.04	0.05	0.30	Opt	0.01	—	60	75	18	
K60	US	0.13	0.90	0.04	0.05	0.10	Opt	0.01	—	60	70	21	
F60	US	0.13	0.90	0.04	0.05	0.10	Opt	0.01	—	60	70	21	
65	US	0.22	1.35	0.04	0.05	0.30	Opt	0.01	—	65	80	16	
70	US	0.26	1.35	0.04	0.05	0.30	Opt	0.01	—	70	85	14	
IH 50(s)	W	0.22	1.50	0.04	0.05	0.70	Opt	0.005-0.05	0.01-0.15	50	75	20	—
IH-60(s)	W	0.22	1.65	0.04	0.05	0.70	Opt	0.005-0.05	0.01-0.15	60	80	18	—
IHX-42	W	0.21	1.35	0.04	0.05	0.30	Opt	0.005-0.05	0.01-0.15	42	60	24	
IHX-45	W	0.22	1.35	0.04	0.05	0.30	Opt	0.005-0.05	0.01-0.15	45	60	22	
IHX-50	W	0.23	1.35	0.04	0.05	0.30	Opt	0.005-0.05	0.01-0.15	50	65	21	
IHX-55	W	0.25	1.35	0.04	0.05	0.30	Opt	0.005-0.05	0.01-0.15	55	70	20	A572
IHX-60	W	0.26	1.35	0.04	0.05	0.30	Opt	0.005-0.05	0.01-0.15	60	75	18	
IHX-65	W	0.26	1.65	0.04	0.05	0.30	Opt	0.005-0.05	0.01-0.15	65	80	17	
IHX-70	W	0.26	1.65	0.04	0.05	0.30	Opt	0.005-0.05	0.01-0.15	70	85	16	—
Pitt-Ten													
X45W	WP	0.20	1.00	0.04	0.05	0.10	Opt	0.01	0.01	45	60	24	
X50W	WP	0.20	1.00	0.04	0.05	0.10	Opt	0.01	0.01	50	65	22	A572, A607
X55W	WP	0.20	1.00	0.04	0.05	0.10	Opt	0.01	0.01	55	70	20	J410c
X60W	WP	0.20	1.00	0.04	0.05	0.10	Opt	0.01	0.01	60	75	18	
YSW-42	Y	0.20	1.10	0.04	0.05	—	Opt	0.01	0.01	42	62	25	A572, J410c
45	Y	0.20	1.25	0.04	0.05	—	Opt	0.01	0.01	45	60	25	
50	Y	0.22	1.25	0.04	0.05	—	Opt	0.01	0.01	50	65	22	
55	Y	0.25	1.35	0.04	0.05	—	Opt	0.01	0.01	55	70	20	A572, A607,
60	Y	0.26	1.35	0.04	0.05	—	Opt	0.01	0.01	60	75	18	J410c
65	Y	0.26	1.35	—	—	—	Opt	0.01	0.01	65	80	18	
70	Y	0.26	1.50	—	—	—	Opt	0.01	0.01	70	85	16	A607, J410c
Algoform													
45	ALG	0.06	0.35	0.01	0.02	0.10	—	0.01	—	45	55	—	
50	ALG	0.06	0.45	0.01	0.02	0.10	—	0.02	—	50	60	—	J410c
60	ALG	0.06	0.75	0.01	0.02	0.10	—	0.04	—	60	70	—	
80	ALG	0.06	0.95	0.01	0.02	0.10	—	0.10	—	80	90	—	
CB/V45	ALG	0.22	1.35	0.04	0.05	—	—	0.01	0.01	45	60	22	
V50	ALG	0.23	1.35	0.04	0.05	—	—	0.01	0.01	50	65	21	A572, A607,
V55	ALG	0.25	1.35	0.04	0.05	—	—	0.01	0.01	55	70	20	J410c
V60	ALG	0.26	1.35	0.04	0.05	—	—	0.01	0.01	60	75	18	
Dofascoloy													
50F	DF	0.15	1.65	0.025	0.035	0.90	—	0.005-0.10	—	50	60	24	
60F	DF	0.15	1.65	0.025	0.035	0.90	—	0.005-0.10	—	60	70	22	A715
70F	DF	0.15	1.65	0.025	0.035	0.90	—	0.005-0.10	—	70	80	20	
80F	DF	0.15	1.65	0.025	0.035	0.90	—	0.005-0.10	—	80	90	18	
45W	DF	0.22	1.35	0.04	0.05	0.10	—	0.005	—	45	60	19(a)	
50W	DF	0.23	1.35	0.04	0.05	0.10	—	0.005	—	50	65	18(a)	
55W	DF	0.25	1.35	0.04	0.05	0.10	—	0.005	—	55	70	17(a)	A572, A607
60W	DF	0.26	1.35	0.04	0.05	0.10	—	0.005	—	60	75	16(a)	
65W	DF	0.26	1.35	0.04	0.05	0.10	—	0.005	—	65	80	15(a)	
70W	DF	0.26	1.65	0.04	0.05	0.30	Opt	0.005	—	70	85	14(a)	A607
80W	DF	0.26	1.65	0.04	0.05	0.30	Opt	0.005	—	80	90	14(a)	A607
CB/V 42	SC	0.21	1.35	0.04	0.05	0.30	Opt	0.005	0.01	42	60	24	A572
45	SC	0.22	1.35	0.04	0.05	0.30	Opt	0.005	0.01	45	60	22	
50	SC	0.23	1.35	0.04	0.05	0.30	Opt	0.005	0.01	50	65	21	A572, A607
55	SC	0.25	1.35	0.04	0.05	0.30	Opt	0.005	0.01	55	70	20	
60	SC	0.26	1.35	0.04	0.05	0.30	Opt	0.005	0.01	60	75	18	
65	SC	0.26	1.35	0.04	0.05	0.30	Opt	0.005	0.01	65	80	15(a)	A572
Stelmax													
45	SC	0.15	1.50	0.02	0.03	0.30	—	0.005	0.01	45	55	25	
50	SC	0.15	1.50	0.02	0.03	0.30	—	0.005	0.01	50	60	24	
60	SC	0.15	1.50	0.02	0.03	0.30	—	0.005	0.01(t)	60	70	22	A715
70	SC	0.15	1.50	0.02	0.03	0.30	—	0.005	0.01(t, u)	70	80	20	
80	SC	0.15	1.50	0.02	0.03	0.30	—	0.005	0.01(t, u)	80	90	18	

Note: Steels in Group I do not contain more than residual copper; therefore, the atmospheric corrosion resistance is usually equal to that of plain carbon steel. If 0.20% Cu minimum is added, then atmospheric corrosion resistance is up to twice that of plain carbon steel. Usually these are semikilled steels but may be killed, particularly at higher strength levels.

(a) Elongation in 8 in.; (b) C-45, C-50, and C-55 are also available in sheets and strip mill plates; elongations are 25, 22, and 20%, respectively; (c) And/or 0.01-0.10 V, or 0.01-0.10 V + 0.015 N max; (d) Controlled rolled to ¾ in. thickness for certified notch toughness; (e) HR=hot rolled, CR=cold rolled; (f) V42, 45, 50, 55, 60, 65 licensed to Alan Wood Steel Co.; (g) Over ½ to 4 in., 0.15 to 0.30 Si; (h) V50 over ¾ in., V55, 60, 65 contain up to 0.015 N; (i) Al and rare earths added for inclusion shape control; (j) Plate and bar only; (k) Lower C and Mn when made for sheet; (l) CC = controlled cooled; (m) 0.01 Al min and Ce or Zr for inclusion shape control; (n) Lower C and Mn when made for sheet; (o) 0.03 Al; (p) 0.06 Zr; (q) Higher tensile strengths available when specified; (r) 0.02 Al; (s) Also classed as Group III steels; (t) Both Cb and V added; (u) 0.02 N.

Table B-2. (continued)

II. Low Manganese-Vanadium-Titanium Group

Copper is usually omitted for better formability

Name	Producer Code	Composition, % C	Mn	P	S	Si	Cu	Other (a)	Mechanical Properties Yield, 1,000 Psi	Tensile, 1,000 Psi	Elongation in 2 In., %	Atmos. Corrosion Resistance	ASTM or SAE Specification No.
VNT-N	ARM	0.22	1.15-1.50	0.035	0.04	0.15-0.50	—	0.04-0.11 V, 0.01-0.03 N	60	80	23	—	A633
VNT-QT	ARM	0.22	1.15-1.50	0.035	0.04	0.15-0.50	—	0.04-0.11 V, 0.01-0.03 N	75	90	20	—	—
Ultra-Form 50	B	0.12	0.60	0.025	0.035	—	—	0.10 Ti	50	60	24	—	A715
80	B	0.12	0.60	0.025	0.035	—	—	0.10 Ti	80	90	18	—	A715
Superform 40	JL	0.20	0.90	0.025	0.03	0.20	—	0.05-0.12 Zr, (0.02-0.06 Al)	40	—	—	1	—
ML-F	McL	0.12	0.75	(0.02)	(0.02)	0.25	0.22	0.005 Cb or 0.02 V	(45)	(62)	(28)	1 to 2	—
NAX-Fine Grain (b)	N	0.18	1.05	0.025	0.03	0.90	Opt	0.06 Zr	50	70	22	2	A606, J410c
Republic 35	R	0.12	0.75	0.04	0.05	0.10	—	0.005 Cb or V	(35)	(47)	(30)	1	—
Par-Ten	US	0.13	0.90	0.04	0.05	—	Opt	0.04 Cb or 0.07 V max	(45)	(55)	(30)	1	—
Pitt-Ten #2	WP	0.15	0.75	0.04	0.05	0.10	—	0.035 V	Properties to meet specific forming problems			1	—
YS-T45	Y	(0.10)	(0.40)	0.04	0.05	—	—	0.01 Al, 0.05 Ti	45	60	25	—	J410c
T50	Y	(0.10)	(0.40)	0.04	0.05	—	—	0.01 Al, 0.05 Ti	50	65	22	—	J410c
	Y	(0.10)	(0.40)	0.04	0.05	—	—	0.01 Al, 0.05 Ti	60	75	18	—	J410c
T70	Y	(0.10)	(0.40)	0.04	0.05	—	—	0.01 Al, 0.05 Ti	70	85	18	—	J410c
T80	Y	(0.10)	(0.45)	0.04	0.05	—	—	0.01 Al, 0.05 Ti	80	95	18	—	J410c

(a) Minimum unless otherwise specified. (b) NAX-Fine Grain is licensed to Republic and Sharon Steel.

III. Manganese and Manganese-Copper Groups

Name	Producer Code	Composition, % C	Mn	P	S	Si	Cu Min	Condition	Mechanical Properties Yield, 1,000 Psi	Tensile, 1,000 Psi	Elongation in 2 In., %	Atmos. Corrosion Resistance	ASTM or SAE Specification No.
A. Manganese Group—Copper not usually indicated													
Armco Lo-Temp(a)	ARM	0.20	0.70-1.35	0.035	0.04	0.15-0.50	—	N	50	70	22	1	A537
Armco Super Lo-Temp(a)	ARM	0.20	0.70-1.35	0.035	0.04	0.15-0.50	—	Q&T	60	80	22	1	A537, A678
Armco LTM-QT	ARM	0.16	0.90-1.50	0.035	0.04	0.15-0.30	—	Q&T	50	70	18(b)	1	A678
Armco LTM-N	ARM	0.14	0.90-1.35	0.035	0.04	0.15-0.30	—	N	42	62	20(b)	1	A633, A662
Armco QTC(c)	ARM	0.20	1.00-1.60	0.035	0.04	0.20-0.50	—	Q&T	75	95	19	1	A678
Armco CT-N(d)	ARM	0.18	1.15-1.60	0.04	0.05	0.15-0.50	Opt	N	50	70	18(b)	—	A633
Armco CT-QT(d)	ARM	0.18	1.15-1.60	0.04	0.05	0.15-0.50	Opt	Q&T	60	80	18(b)	—	—
RQC80	B	0.20	1.35	0.04	0.05	0.15-0.30	—	Q&T	80	95	18	1	—
RQC90(e)	B	0.20	1.35	0.04	0.05	0.15-0.30	—	Q&T	90	100	18	1	—
RQC100(e)	B	0.20	1.50	0.04	0.05	0.15-0.30	—	Q&T	100	110	18	1	—
RQC-60N	B	0.20	0.70-1.60	0.035	0.04	0.15-0.50	—	N	50	70	22	1	A537
RQC 60 Q&T	B	0.20	0.70-1.60	0.035	0.04	0.15-0.50	—	Q&T	60	80	22	1	A537
LT-75HS	L	0.22	1.10-1.60	0.035	0.04	0.20-0.60	Opt	Q&T	75	95-115	17	2	—
LT-75N	L	0.24	0.70-1.35	0.035	0.04	0.15-0.30	—	N	50	70	22	2	A537
LT-75QT	L	0.24	0.70-1.35	0.035	0.04	0.15-0.30	—	Q&T	60	80	22	2	A537
Lukens 45	L	0.20	1.20	0.04	0.05	—	—	—	45	65	24	1	—
50	L	0.20	1.35	0.04	0.05	—	—	—	50	70	24	1	—
55	L	0.22	1.35	0.04	0.05	—	—	—	55	75	23	1	—
60	L	0.22	1.60	0.04	0.05	0.15-0.30	—	—	60	80	23	1	—
PX50	P	0.20	1.00-1.50	0.04	0.05	0.15-0.50	—	N	50	70-90	22	1	—
Char-Pac (Norm.)	US	0.20	0.70-1.35	0.035	0.04	0.15-0.50	0.35 max	N	50	70	22	1	A537
Char-Pac (Q&T)	US	0.20	0.70-1.35	0.035	0.04	0.15-0.50	0.35 max	Q&T	60	80	22	1	A537
Con-Pac 80(f)	US	0.20	1.35	0.04	0.05	0.15-0.40	—	Q&T	80	95	18	1	—
90(f)	US	0.20	1.35	0.04	0.05	0.15-0.40	—	Q&T	90	100	18	1	—
100(f)	US	0.20	1.50	0.04	0.05	0.15-0.40	—	Q&T	100	110	18	1	—
M(f)	US	0.20	1.60	0.035	0.04	0.15-0.50	—	Q&T	75	95	19	1	—
B. Manganese-Copper Group													
AW-440	AW	0.28	1.10-1.60	0.04	0.05	0.30	0.20	—	50	70	18(b)	2	A440
Med. Mn	B	0.28	1.10-1.60	0.04	0.05	0.30	0.20-0.35	—	50	70	20	2	
Hi-Man	IN	0.28	1.10-1.60	0.04	0.05	0.30	0.20	—	50	70	18(b)	2	A440, J410c

Table B-2. (continued)

III. Manganese and Manganese-Copper Groups (Continued)

Name	Producer Code	Composition, % C	Mn	P	S	Si	Cu Min	Condition	Mechanical Properties Yield, 1,000 Psi	Tensile, 1,000 Psi	Elongation in 2 In., %	Atmos. Corrosion Resistance	ASTM or SAE Specification No.
Jalten-3	JL	0.25	1.60	0.04	0.05	0.30	0.20	—	50	70	22	2	A440, J410c
Kaisaloy 50 MM	K	0.27	1.10-1.60	0.035	0.04	0.30	0.20	—	50	70	18(b)	2	A440, J410c
Lukens A440	L	0.28	1.10-1.60	0.04	0.05	0.30	0.20	—	50	70	18(b)	2	A440
NAX-Hi Mang.	N	0.28	1.10-1.60	0.04	0.05	0.30	0.20	—	50	70	22	2	A440
Orelloy 440	O	(0.19)	(1.28)	(0.020)	(0.030)	(0.23)	(0.35)	—	50	70	18(b)	2	A440
Republic M	R	0.25	1.10-1.60	0.04	0.05	0.30	0.20	—	50	75	20	2	J410c
Yo Man	Y	0.25	1.10-1.60	0.04	0.05	0.30	0.20	—	50	70	20	2	A440, J410c
Dofascoloy M	DF	0.28	1.10-1.60	0.04	0.05	0.30	0.20	—	50	70	18(b)	2	A440, J410c

Note. N, normalized; Q&T, quenched and tempered; (a) 0.25 Ni, 0.25 Cr, 0.08 Mo; (b) Elongation in 8 in.; (c) 0.35 Ni, 0.25 Cr, 0.08 Mo; (d) Contains 0.01-0.05 Cb; (e) 0.0005 B min; (f) Boron-treated.

IV. Manganese-Vanadium-Copper Group

Name	Producer Code	Composition, % C	Mn	P	S	Si	Cu Min	V Min	Mechanical Properties Yield, 1,000 Psi	Tensile, 1,000 Psi	Elongation in 2 In., %	Atmos. Corrosion Resistance	ASTM or SAE Specification No.
AW-441	AW	0.22	1.25	0.04	0.05	0.30	0.20	0.02	50	70	22	2	A441
Armco High Strength B	ARM	0.22	0.85-1.25	0.04	0.05	0.30	0.20	0.02	50	70	18 (a)	2	A441, J410c
Mn-V	B	0.22	0.85-1.25	0.04	0.05	0.30	0.20	0.02	50	70	18 (a)	2	A441, A606
Tri-Steel	IN	0.22	0.85-1.25	0.04	0.05	0.30	0.20	0.02	50	70	22	2	A441, J410c
Jalten 1	JL	0.15	1.25	0.04	0.05	0.30	0.20	0.02	50	70	22	2	A441, J410c
Kaisaloy 50 MV	K	0.22	0.85-1.25	0.035	0.04	0.30	0.20	0.02	50	70	18 (a)	2	A441, J410c
Lukens A441	L	0.22	1.25	0.04	0.05	0.30	0.20	0.02	50	70	18 (a)	2	A441
GLS-441	N	0.22	1.25	0.04	0.04	0.30	0.20	0.02	50	70	22	2	A441
ML-F (A-441)	McL	0.22	1.25	0.04	0.05	0.30	0.20	0.02	50	70	22	2	A441
Orelloy 441	O	0.17	1.19	0.018	0.031	0.24	0.36	0.04	50	70	22	2	A441
Clay-Loy	P	0.22	1.25	0.04	0.05	0.35	0.20	0.02	50	70	18	2	A441
Republic A-441	R	0.22	0.85-1.25	0.04	0.05	0.30	0.20	0.02	50	70	22	2	A441
Tri-Ten	US	0.22	1.25	0.04	0.05	0.30	0.20	0.02	50	70	18 (a)	2	A441
YSW A441	Y	0.22	1.25	0.04	0.05	0.30	0.20	0.02	50	70	18	2	A441, J410c
Dofascoloy MV	DF	0.22	0.85-1.25	0.04	0.05	0.30	0.20	0.02	50	70	18 (a)	2	A441
Stelco-Vanadium	SC	0.22	1.25	0.04	0.05	0.30	0.20	0.02	50	70	22	2	A441
Wgh.-Pgh. A441	WP	0.22	0.85-1.25	0.04	0.05	0.30	0.20	0.02	50	70	18 (a)	2	A441

(a) Elongation in 8 in.

V. Multiple Alloy and Copper Group

Name	Producer Code	Composition, % C	Mn	P	S	Si	Cu	Mo	Cr	Ni	Other (a)	Mechanical Properties Yield, 1,000 Psi	Tensile, 1,000 Psi	Elongation in 2 In., %	Atmos. Corrosion Resistance	ASTM or SAE Specification No.
Armco High Strength A	ARM	0.12	0.90	0.04	0.05	0.15-0.70	0.20-0.50	—	0.50-1.00	0.25 min	0.07 Ti max	50	70	18 (b)	4 to 6	A242
Armco High Strength A588	ARM	0.20	0.75-1.25	0.04	0.05	0.15-0.30	0.20-0.40	—	0.40-0.70	0.25-0.50	0.01-0.10 V	50	70	18 (b)	4	A588
Mayari R-50	B	0.20	0.75-1.25	0.04	0.05	0.15-0.30	0.20-0.40	—	0.40-0.70	0.25-0.50	0.01-0.10 V	50	70	21	4	A588, J410c
Mayari R-60	B	0.20	0.75-1.35	0.04	0.05	0.15-0.30	0.20-0.40	—	0.40-0.70	0.25-0.50	0.01-0.10 V	60	80	16 (b)	4	—
Ni-Cu-Ti	JL	0.15	1.00	0.04	0.05	0.50	0.30 min	—	—	0.70	0.05 Ti	50	70	22	4	A588
Kaisaloy 45FG	K	0.12	0.60	0.035	0.04	0.50	0.30	0.10	0.25	0.60	0.02 V, 0.005 Ti	45	60	25	6 max	—
50CR	K	0.20	1.25	0.035	0.04	0.25-0.75	0.20-0.35	0.15	0.10-0.25	0.30-0.60	0.02 V, 0.005 Ti	50	70	22	5 to 8	A242, A588, A606, 410c
60SG	K	0.20	1.25	0.035	0.04	0.35	0.80	0.25	—	0.90	0.005 V	60	80	16(b)	2 to 3	—
70MB	K	0.15	0.75	0.035	0.04	0.35	—	0.40-0.60	—	—	0.001 B	70	85	—	—	—
UCV-60	L	0.25	1.50	0.04	0.05	0.15-0.30	0.25-0.40	—	0.40-0.65	—	0.02-0.10 V	60	70	19	—	—
BCV-42 (c)	L	0.25	1.50	0.04	0.05	0.15-0.30	0.25-0.40	—	0.40-0.65	—	0.02-0.10 V	42	63	19	—	—
46 (c)	L	0.25	1.50	0.04	0.05	0.15-0.30	0.25-0.40	—	0.40-0.65	—	0.02-0.10 V	46	67	19	—	—
50 (c)	L	0.25	1.50	0.04	0.05	0.15-0.30	0.25-0.40	—	0.40-0.65	—	0.02-0.10 V	50	70	19	—	—
55 (c)	L	0.25	1.50	0.04	0.05	0.15-0.30	0.25-0.40	—	0.40-0.65	—	0.02-0.10 V	55	70	19	—	—
60 (c)	L	0.25	1.50	0.04	0.05	0.15-0.30	0.25-0.40	—	0.40-0.65	—	0.02-0.10 V	60	70	19	—	—
70 (c)	L	0.25	1.50	0.04	0.05	0.15-0.30	0.25-0.40	—	0.40-0.65	—	0.02-0.10 V	70	80	16	—	—

Table B-2. (continued)

V. Multiple Alloy and Copper Group—Continued

Name	Producer Code	C	Mn	P	S	Si	Cu	Mo	Cr	Ni	Other (a)	Yield, 1,000 Psi	Tensile, 1,000 Psi	Elongation in 2 In., %	Atmos. Corrosion Resistance	ASTM or SAE Specification No.
		Composition, %										Mechanical Properties				
NAX-High Tensile	N(d)	0.16	0.90	0.025	0.03	0.90	Opt	—	0.80	—	0.06 Zr	50	70	22	4 to 6	A242, A588, A606
NAX-80	N	0.12	1.00	0.02	0.03	0.90	0.80	—	—	—	0.01 Cb, 0.06 Zr(e)	80	90	18	4 to 6	J410c
Orelloy 242 #2	O	0.17	1.05	0.013	0.022	0.20	0.25	—	0.50	0.06	—	50	70	18 (b)	4 to 6	A242
	O	0.18	1.16	0.008	0.005	0.20	0.29	—	0.56	0.13	0.03 V	50	70	18 (b)	4 to 6	A588
Republic 50	R	0.15	0.50-1.00	0.04	0.05	—	0.30-1.00	0.10 min	0.30	0.40-1.10	—	50	70	22	4 to 6	A242, A588, A606, J410c
Republic 60	R	0.15	0.50-1.00	0.04	0.05	—	0.30-1.00	0.10 min	0.30	0.40-1.10	0.01 Cb, a/o 0.02 V	60	80	21	4 to 6	
NAX	SH	(0.15)	(0.70)	0.04	0.04	(0.70)	Opt	—	(0.70)	—	0.05 Zr	50	70	22	4 to 6	J410c
Cor-Ten B	US(f)	0.10-0.19	0.90-1.25	0.04	0.05	0.15-0.30	0.25-0.40	—	0.40-0.65	—	0.02-0.10 V	50	70	19 (b)	4	A242, A588
Cor-Ten C	US(f)	0.12-0.19	0.90-1.35	0.04	0.05	0.15-0.30	0.25-0.40	—	0.40-0.70	—	0.04-0.10 V	60	80	16 (b)	4	—
Yoloy HS	Y	0.15	1.00	0.04	0.05	0.30	0.50-1.00	0.25	—	(1.00)	—	50	70	22	4	A242, A588, A606, J410c
HSX	Y	0.15	1.00	0.04	0.05	0.35	0.50-1.00	0.25	—	(1.00)	—	45	62	25	4	
S (g)	Y	0.20	1.00	0.04	0.05	0.30	0.75-1.25	—	—	(1.90)	—	50	70	22	4 to 6	A242, A606, J410c
Yoloy T-50	Y	(0.10)	(0.45)	0.04	0.05	(0.40)	(0.25)	—	(0.65)	—	0.05 Ti, 0.01 Al	50	70	22	4	A242, A606, J410c
T-60	Y	(0.10)	(0.45)	0.04	0.05	(0.40)	(0.25)	—	(0.65)	—	0.05 Ti, 0.01 Al	60	80	20	4	
T-70	Y	(0.10)	(0.45)	0.04	0.05	(0.40)	(0.25)	—	(0.65)	—	0.05 Ti, 0.01 Al	70	85	18	4	
T-80	Y	(0.10)	(0.45)	0.04	0.05	(0.40)	(0.25)	—	(0.65)	—	0.05 Ti, 0.01 Al	80	95	16	4	
Algotuf 50	ALG	0.18	1.25	0.03	0.05	0.30	0.20-0.40	—	0.60	0.50	0.02 V	50	70	22	4 to 5	A588
70	ALG	0.20	1.60	0.025	0.035	0.35	0.50	—	—	0.50	0.06 Cb, 0.10 V	70	90	15	4 to 6	—
Dofascoloy #1	DF	0.25	1.25	0.04	0.05	0.30	0.60	—	—	0.90	—	50	70	18 (b)	4 to 5	A242, A606
Dofascoloy ZR	DF	0.20	1.00	0.04	0.05	0.50-1.00	0.20	—	0.80	—	0.03 Zr	50	70	22	4	A588
Stelcoloy 50	SC	0.15	0.80-1.35	0.04	0.05	0.15-0.30	0.20-0.50	—	0.30-0.50	0.25-0.50	0.01-0.10 V	50	70	21	4 to 5	A242, A588, A606
60	SC	0.20	1.50	0.04	0.05	0.15-0.30	0.20-0.50	—	0.30-0.50	0.25-0.50	0.01-0.12 V	60	80	18	4 to 5	—
70	SC	0.22	1.50	0.03	0.04	0.15-	0.20-0.50	—	—	0.25-0.50	0.02-0.10 V, 0.05 Cb	70	90	14 (b)	2 to 4	—

(a) Minimum, unless otherwise specified; (b) Elongation in 8 in.; (c) Properties are achieved by normalizing BCV-42, 46, 50, and 55; and by quenching and tempering BCV-60 and 70; (d) NAX-High Tensile is licensed to Republic and Sharon Steel; (e) May contain Ti and/or B; (f) Licensed to Alan Wood, Algoma, Colorado Fuel & Iron, Crucible, Granite City (Subs. of National Steel), Inland, Interlake, Jones & Laughlin, Kaiser, Lukens, Republic, and Sharon Steel; (g) Can be precipitation hardened (stress relief annealed) to increase tensile properties.

VI. Multiple Alloy and Copper and Phosphorus Group

Name	Producer Code	C	Mn	P	S	Si	Cu	Ni	Other	Yield, 1,000 Psi	Tensile, 1,000 Psi	Elongation in 2 In., %	Atmos. Corrosion Resistance	ASTM or SAE Specification No.
		Composition, %								Mechanical Properties				
AW Dynalloy 50	AW	(0.13)	(0.90)	(0.08)	(0.03)	(0.20)	(0.35)	(0.45)	(0.08 Mo)	50	70	22	4 to 5	A242
Mayari R (a)	B	0.12	(0.75)	0.12	0.05	(0.55)	0.50	1.00	0.40-1.00 Cr	50	70	18 (c)	5 to 8	A242, A606, J410c
Orelloy 242	O	(0.10)	(0.50)	(0.09)	(0.03)	(0.48)	(0.43)	(0.43)	(0.86 Cr)	50	70	22	4 to 5	—
Orelloy 242 #1	O	(0.10)	(0.50)	(0.09)	(0.03)	(0.48)	(0.43)	(0.43)	(0.86 Cr)	50	70	22	4 to 6	A242
Cor-Ten A	US(b)	0.12	0.20-0.50	0.07-0.15	0.05	0.25-0.75	0.25-0.55	0.65	0.30-1.25 Cr	50	70	19 (c)	5 to 8	A242, A606, J410c
Pitt Ten #1 (d)	WP	0.20	0.55-0.95	0.07	0.05	0.20	0.30-0.50	0.45-0.95	0.20-0.50 Cr	50	70	22	4 to 6	J410c
Dofascoloy P	DF	0.16	0.90	0.12	0.05	0.15-0.35	0.60	0.90	0.60 Cr	50	70	18	4 to 6	A242, A606
Dofascoloy #2	DF	0.22	1.25	0.12	0.05	0.30	0.60	0.90	0.60 Cr	45	65	22	—	A606
Stelcoloy G	SC	0.12	0.75	0.12	0.04	0.15-0.30	0.30-0.60	0.30-0.60	0.30-0.60 Cr	50	70	22	4 to 6	A242, A606

(a) 0.10 Zr; (b) Cor-Ten A is licensed to Alan Wood, Algoma, Colorado Fuel & Iron, Crucible, Granite City (Subs. of National Steel), Inland, Interlake, Jones & Laughlin, Kaiser, Lukens, Republic, and Sharon Steel; (c) Elongation in 8 in.; (d) Can be precipitation hardened and made with lower phosphorus.

Table B-2. (continued)

VII. Precipitation-Hardening Alloys

Name	Producer Code	Composition, % C	Mn	P	S	Si	Cu	Mo	Other	Mechanical Properties: Yield, 1,000 Psi	Tensile, 1,000 Psi	Elongation in 2 In., %	Atmos. Corrosion Resistance	ASTM or SAE Specification No.
AWP-80(a)	AW	0.08	0.90-1.30	0.04	0.04	0.15-0.35	—	0.20-0.30	0.01-0.07 Cb, 0.0005 B min, 0.05 Al min	80	90	18	—	—
70	R	0.20	1.00	0.04	0.04	0.15	1.00-1.50	0.20-0.30	1.20-1.75 Ni	70	90	18	4 to 6	—
80	R	0.20	1.00	0.04	0.04	0.15	1.00-1.50	0.20-0.30	1.20-1.75 Ni, 0.01 Cb, a/o 0.02 V	80	100	18	4 to 6	—
Ni-Cu-Cb	US	0.06	0.40-0.65	0.035	0.04	0.20-0.35	1.00-1.30	—	1.20-1.50 Ni, 0.02 Cb	75-85	88-90	18	—	A710B
Yoloy S	Y	0.20	1.00	0.04	0.05	0.30	0.75-1.25	—	1.60-2.20 Ni	65	(90)	20	4 to 6	—
Nicuten 85	ALG	(0.06)	(0.35)	0.04	0.04	—	(1.50)	—	(1.00 Ni), (0.02 V)	85	90	20	—	—

(a) Also classed as a Group V and VIII steel.

VIII. Constructional Alloys (Extra High-Strength Steels)

Name	Producer Code	Composition, % C	Mn	P	S	Si	Ni	Cr	Mo	Other (a)	Mechanical Properties (b): Yield, 1,000 Psi	Tensile, 1,000 Psi	Elongation in 2 In., %	ASTM or SAE Specification No.
SSS-100	ARM	0.12-0.20	0.40-0.70	0.035	0.04	0.20-0.35	—	1.40-2.00	0.40-0.60	0.20-0.40 Cu,	100	115	16	
SSS-100A	ARM	0.13-0.20	0.40-0.70	0.035	0.04	0.20-0.35	—	0.85-1.20	0.15-0.25	0.04-0.10 V	100	115	16	A514,
SSS-100B	ARM	0.13-0.20	0.40-0.70	0.035	0.04	0.20-0.35	—	1.15-1.65	0.25-0.40	or Ti,0.0015-0.005 B	100	115	16	A517
HY80	ARM	0.18	0.10-0.40	0.025	0.025	0.15-0.35	2.00-3.25	1.00-1.80	0.20-0.60	—	80	—	20	—
HY100	ARM	0.20	0.10-0.40	0.025	0.025	0.15-0.35	2.25-3.50	1.00-1.80	0.20-0.60	—	100	—	18	—
Stroloy 2A	BW	0.15-0.21	0.70-1.00	0.025	0.025	0.20-0.35	0.40-0.70	0.80-1.10	0.20-0.30	0.001 B	100	115-145	15	—
Stroloy 5C	BW	0.15-0.21	0.70-1.00	0.025	0.025	0.20-0.35	—	0.75-1.10	0.15-0.25	0.001 B	100	115-145	15	—
RQ100A	B	0.12-0.21	0.45-0.70	0.035	0.04	0.20-0.35	—	—	0.50-0.65	0.001-0.005 B	100	115-135	18	A514
RQ100B	B	0.12-0.21	0.45-0.70	0.035	0.04	0.20-0.35	1.20-1.50	—	0.45-0.60	0.001-0.005 B	100	115-135	18	A514
RQ100	B	0.12-0.21	0.45-0.70	0.035	0.04	0.20-0.35	1.20-1.50	0.85-1.20	0.45-0.60	0.001-0.005 B	100	115-135	18	A514
Jalloy S-90	JL	0.10-0.20	1.50	0.04	0.04	0.50	—	1.50	0.30	0.0005 B	90	(100)	(18)	A514
S-100	JL	0.10-0.20	1.50	0.04	0.04	0.50	—	1.50	0.30	0.0005 B	100	(110)	(18)	A514, A517
S-110	JL	0.10-0.20	1.50	0.04	0.04	0.50	—	1.50	0.30	0.0005 B	110	(120)	(17)	A514, A517
J-100	J	0.16-0.20	0.60-0.90	0.025	0.025	0.20-0.35	1.10-1.40	0.50-0.75	0.40-0.60	0.06-0.12 V	100	115	16	—
HY80	J	0.18	0.10-0.40	0.025	0.025	0.15-0.35	2.00-3.25	1.00-1.80	0.20-0.60	—	80	—	20	—
HY100	J	0.20	0.10-0.40	0.025	0.025	0.15-0.35	2.20-3.50	1.00-1.80	0.20-0.60	—	100	—	18	—
HY140	J	0.16	0.60-0.90	0.015	0.015	0.15-0.35	4.75-5.25	0.40-0.70	0.30-0.65	0.05-0.10 V, 0.20 Cu max	140	—	15	—
Ni-Mo	J	0.28	0.15-0.45	0.020	0.020	0.15-0.35	2.75-3.50	—	0.25-0.60	0.08 V max	75	110	20	—
Kaisaloy 100	K	0.15-0.21	0.80-1.10	0.035	0.04	0.40-0.80	—	0.50-0.80	0.28	0.05-0.15 Zr, 0.0025 B max	100	115-135	18	A514
N-A-XTRA-80	N	0.21	0.60-1.10	0.04	0.04	0.40-0.90	—	0.40-0.90	0.28	0.05 Zr, 0.0025 B max	80	95	18	
90	N	0.21	0.60-1.10	0.04	0.04	0.40-0.90	—	0.40-0.90	0.28	0.05 Zr, 0.0025 B max	90	105	18	A514, A517
100	N	0.21	0.60-1.10	0.04	0.04	0.40-0.90	—	0.40-0.90	0.28	0.05 Zr, 0.0025 B max	100	115	18	
110	N	0.21	0.60-1.10	0.04	0.04	0.40-0.90	—	0.40-0.90	0.28	0.05 Zr, 0.0025 B max	110	125	18	
PX80 Plus	P	0.15-0.21	0.80-1.10	0.035	0.04	0.40-0.90	—	0.50-0.90	0.28		80	95	18	A514
PX90 Plus	P	0.15-0.21	0.80-1.10	0.035	0.04	0.40-0.90	—	0.50-0.90	0.28	0.05 Zr, 0.0025	90	105	18	A514
PX100 Plus	P	0.15-0.21	0.80-1.10	0.035	0.04	0.40-0.90	—	0.50-0.90	0.28	B max	100	115	18	A514, A517
PX110 Plus	P	0.15-0.21	0.80-1.10	0.035	0.04	0.40-0.90	—	0.50-0.90	0.28		110	125	18	A514, A517
HY80	US/L	0.18	0.10-0.40	0.025	0.025	0.15-0.35	2.00-3.25	1.00-1.80	0.20-0.60	—	80	—	20	—
HY100	US/L	0.20	0.10-0.40	0.025	0.025	0.15-0.35	2.25-3.50	1.00-1.80	0.20-0.60	—	100	—	18	—
HY130	US/L	0.12	0.60-0.90	0.010	0.015	0.20-0.35	4.75-5.25	0.40-0.70	0.30-0.65	0.05-0.10 V	130	—	15	—
T-1	US/L	0.10-0.20	0.60-1.00	0.035	0.04	0.15-0.35	0.70-1.00	0.40-0.65	0.40-0.60	0.15-0.50 Cu, 0.0005-0.006 B, 0.03-0.08 V	100	115-135	18	A514, A517-F
T-1-A (c)	US/L	0.12-0.21	0.70-1.00	0.035	0.04	0.20-0.35	—	0.40-0.65	0.15-0.25	0.0005-0.005B, 0.01-0.03 Ti, 0.03-0.08 V	100	115-135	18	A514, A517-B
T-1-B (c)	US/L	0.12-0.21	0.95-1.30	0.035	0.04	0.20-0.35	0.30-0.70	0.40-0.65	0.20-0.30	0.0005 B, 0.03-0.08 V	100	115-135	18	A514, A517-H
Yoloy T-80	Y	(0.10)	(0.45)	0.04	0.05	(0.40)	—	(0.65)	—	(0.25 Cu), 0.05 Ti, 0.01 Al	80	95	16	J410c
YS-T80	Y	(0.10)	(0.45)	0.04	0.05	—	—	—	—	0.01 Al, 0.05 Ti	80	95	18	A656, J410c

Note: Bars, structural shapes, and tubing can also be supplied in these alloys. (a) Minimum unless otherwise specified; (b) All alloys quenched and tempered except NAX-80, Yoloy T-80, and YS-T80 which are hot rolled; (c) Copper optional.

Table B-2. (Concluded)

IX. Abrasion-Resistant Alloys

Name	Producer Code	Composition, % C	Mn	P	S	Si	Cu	Cr	Mo	Other (a)	Condition	Mechanical Properties Hardness, Bhn	Yield, 1,000 Psi	Tensile, 1,000 Psi
AW-AR	AW	0.35-0.50	1.50-2.00	0.05	0.055	0.15-0.30	—	—	—	—	HR	235	—	—
SSS-AR-321	ARM	0.25	0.40-0.70	0.04	0.05	0.20-0.35	0.20-0.40	0.85-2.0	0.15-0.60	0.04-1.10 Ti or V, 0.0015-0.005 B	Q&T	321	—	—
SSS-AR-360	ARM	0.25	0.40-0.70	0.04	0.05	0.20-0.35	0.20-0.40	0.85-2.0	0.15-0.60	0.04-0.10 Ti or V, 0.0015-0.005 B	Q&T	360	—	—
SSS-AR-400	ARM	0.25	0.40-0.70	0.04	0.05	0.20-0.35	0.20-0.40	0.85-2.0	0.15-0.60	0.04-0.10 Ti or V, 0.0015-0.005 B	Q&T	400	—	—
Armco Abrasion Resisting	ARM	0.33-0.43	1.40-2.00	0.05	0.05	0.15-0.35	—	—	—	—	HR	225	—	—
AR No. 235	B	0.35-0.50	1.40-2.00	0.05	0.05	0.15-0.30	—	—	—	—	HR	(235)	—	—
RQAR-321	B	0.25-0.32	0.40-0.65	0.035	0.040	0.20-0.35	—	0.80-1.15	0.15-0.25	—	Q&T	321	—	—
RQAR-340	B	0.25-0.32	0.40-0.65	0.035	0.040	0.20-0.35	—	0.80-1.15	0.15-0.25	—	Q&T	340	—	—
RQAR-360	B	0.25-0.32	0.40-0.65	0.035	0.040	0.20-0.35	—	0.80-1.15	0.15-0.25	—	Q&T	360	—	—
RQAR-400	B	0.25-0.32	0.40-0.65	0.035	0.040	0.20-0.35	—	0.80-1.15	0.15-0.25	—	Q&T	400	—	—
RQC-321	B	0.28	1.50	0.040	0.050	0.20-0.60	—	—	—	0.005B	Q&T	321	—	—
RQC-340	B	0.28	1.50	0.040	0.050	0.20-0.60	—	—	—	0.005B	Q&T	340	—	—
RQ321A	B	0.12-0.21	0.45-0.70	0.035	0.04	0.20-0.35	—	—	0.50-0.65	0.001-0.005 B	Q&T	321	—	—
RQ340A	B	0.12-0.21	0.45-0.70	0.035	0.04	0.20-0.35	—	—	0.50-0.65	0.001-0.005 B	Q&T	340	—	—
RQ360A	B	0.12-0.21	0.45-0.70	0.035	0.04	0.20-0.35	—	—	0.50-0.65	0.001-0.005 B	Q&T	360	—	—
RQ321B	B	0.12-0.21	0.45-0.70	0.035	0.04	0.20-0.35	—	—	0.45-0.60	1.20-1.50 Ni, 0.001-0.005 B	Q&T	321	—	—
RQ340B	B	0.12-0.21	0.45-0.70	0.035	0.04	0.20-0.35	—	—	0.45-0.60	1.20-1.50 Ni, 0.001-0.005 B	Q&T	340	—	—
RQ360B	B	0.12-0.21	0.45-0.70	0.035	0.04	0.20-0.35	—	—	0.45-0.60	1.20-1.50 Ni, 0.001-0.005 B	Q&T	360	—	—
RQ-321	B	0.12-0.21	0.45-0.70	0.035	0.04	0.20-0.35	—	0.85-1.20	0.45-0.60	1.20-1.50 Ni, 0.001-0.005 B	Q&T	321	—	—
RQ-340	B	0.12-0.21	0.45-0.70	0.035	0.04	0.20-0.35	—	0.85-1.20	0.45-0.60	1.20-1.50 Ni, 0.001-0.005 B	Q&T	340	—	—
RQ-360	B	0.12-0.21	0.45-0.70	0.035	0.04	0.20-0.35	—	0.85-1.20	0.45-0.60	1.20-1.50 Ni, 0.001-0.005 B	Q&T	360	—	—
Abrasion Resisting, Med. Hard	IN	0.35-0.50	1.20-1.65	0.04	0.05	0.10-0.30	—	—	—	—	HR	210	—	—
Abrasion Resisting, Full Hard	IN	0.70-0.85	0.60-1.00	0.04	0.05	0.30	—	—	—	—	HR	250	—	—
Jalloy AR	JL	0.25-0.31	1.65	0.04	0.04	0.15-0.30	0.20	1.20	0.35	0.0005 B	Q&T	—	90	—
Jalloy AR-280	JL	0.25-0.31	1.65	0.04	0.04	0.15-0.30	0.20	1.20	0.35	0.0005 B	Q&T	260	(130)	—
320	JL	0.25-0.31	1.65	0.04	0.04	0.15-0.30	0.20	1.20	0.35	0.0005 B	Q&T	300	(140)	—
360	JL	0.25-0.31	1.65	0.04	0.04	0.15-0.30	0.20	1.20	0.35	0.0005 B	Q&T	340	(157)	—
400	JL	0.25-0.31	1.65	0.04	0.04	0.15-0.30	0.20	1.20	0.35	0.0005 B	Q&T	400	(184)	—
Jalloy AR Q	JL	0.25-0.31	1.65	0.04	0.04	0.15-0.30	0.20	1.20	0.35	0.0005 B	Q&T	500	(217)	—
Jalloy S-340	JL	0.10-0.20	1.50	0.04	0.04	0.50	—	1.50	0.30	0.0005 B	Q&T	320	(150)	—
Kaisaloy AR	K	0.35-0.50	1.50-2.00	0.05	0.055	0.15-0.35	—	—	—	—	HR	—	—	—
Kaisaloy 360 HI-WR	K	0.17-0.25	0.80-1.20	0.035	0.04	0.40-0.90	—	0.50-0.90	0.18-0.60	0.0025 B max	Q&T	360	—	—
Kaisaloy 400 HI-WR	K	0.20-0.30	0.80-1.20	0.035	0.04	0.40-0.90	—	0.50-0.90	0.18-0.60	0.0025 B max	Q&T	400	—	—
AR-300	L	0.28	1.40	0.04	0.05	0.20-0.50	0.20	—	—	—	Q&T	285-321	—	—
XAR-15	N	0.21	0.60-1.10	0.04	0.04	0.90	—	0.45-0.85	0.30	0.05 Zr, 0.0025 B max	Q&T	360	(165)	(180)
XAR-30 (b)	N	0.30	0.60-1.10	0.04	0.04	0.90	—	0.45-0.85	0.30	0.05 Zr, 0.0025 B max	Q&T	360	(165)	(180)
Orelloy AR	O	0.43	1.50	0.024	0.013	0.29	0.08	—	—	—	HR	235	—	—
PX360 Bhn	P	0.15-0.21	0.80-1.10	0.035	0.04	0.40-0.90	—	0.50-0.90	0.28	0.05 Zr, 0.0025 B max	Q&T	321-360	(165)	(180)
Republic 100-AR	R	0.16	1.40	0.015	0.010	0.25	0.70	1.40	0.45	1.40 Ni 0.04 Al	HR	375-415	100	140
AR	US/L	0.35-0.50	1.50-2.00	0.05	0.055	0.15-0.35	—	—	—	—	HR	(200-250)	—	—
300	US	0.32	1.60	0.04	0.05	0.60	—	—	—	0.0005 B	Q&T	285	—	(136)
350	US	0.32	1.60	0.04	0.05	0.60	—	—	—	0.0005 B	Q&T	321	—	(152)
360	US	0.26-0.33	1.15-1.50	0.035	0.040	0.20-0.35	—	0.40-0.65	0.08-0.15	0.0005 B	Q&T	360	—	(175)
T-1-A 321	US	0.12-0.21	0.70-1.00	0.035	0.040	0.20-0.35	Opt	0.40-0.65	0.15-0.25	0.03-0.08 V,	Q&T	321	—	(152)
340										0.01-0.03 Ti,	Q&T	340	—	(163)
360										0.0005-0.005 B	Q&T	360	—	(175)
T-1-B 321	US/L	0.12-0.21	0.95-1.30	0.035	0.040	0.20-0.35	Opt	0.40-0.65	0.20-0.30	0.30-0.70 Ni,	Q&T	321	—	(152)
340										0.03-0.08 V,	Q&T	340	—	(163)
360										0.0005 B	Q&T	360	—	(175)
T-1 321	US/L	0.10-0.20	0.60-1.00	0.035	0.040	0.15-0.35	0.15-0.50	0.40-0.65	0.40-0.60	0.70-1.00 Ni,	Q&T	321	—	(152)
340										0.03-0.08 V,	Q&T	340	—	(163)
360										0.0005-0.006 B	Q&T	360	—	(175)
Austenitic manganese A	US	0.70-0.90	12.50-14.50	0.07	0.04	0.50-0.80	—	0.50	—	3.00-3.50 Ni	HR	—	—	—
B	US	1.00-1.25	11.50-13.50	0.07	0.04	0.15-0.30	—	0.50	—	—	HR	—	—	—
C	US	0.70-0.90	12.50-14.50	0.07	0.04	0.50-0.80	—	0.50	0.40-0.60	1.75-2.25 Ni	HR	—	—	—

Note: HR, hot rolled; Q&T, quenched and tempered; (a) Minimum unless otherwise specified; (b) Available to Bhn 500.

Table B-3 Properties of USS T-1 Steel

MECHANICAL PROPERTIES:

Y.S.	90-100 ksi
U.S.	105-135 ksi
Elong. % (in 2 in.)	15-18%
Red. of Area	35-50%
Hardness	235-293 Brinell
Fatigue Strength	50 ksi
Toughness	Charpy V-notch 15 ft-lbs at -50°F
Transition Temperature	-50°F

PHYSICAL PROPERTIES:

Modulus of Elasticity	29×10^6 psi
Shear Modulus	11×10^6 psi
Poisson's Ratio	0.30
Density	.283 lbs/in^3
Coefficient of Expansion	6.5×10^{-6} in/in/°F (-50to+150°F)

Chemistry

C	Mn	P	S	Si	B	Cr	Cu	Mo	Ni	V
.10	.60	.035	.040	.15	.0005	.40	.15	.40	.70	.03
.20	1.00	max	max	.35	.006	.65	.50	.60	1.00	.08

HEAT TREATMENT: Water quenched from 1650-1750°F and tempered at 1100-1275°F.

WELDABILITY: Must be in quenched and tempered condition.

ATMOSPHERIC CORROSION RESISTANCE: 4-6 times structural carbon steel.

MACHINABILITY: Similar to alloy steels of same hardness.

FORMABILITY: Can be cold formed to inside bend radii of 2-3 times the thickness, depending on thickness.

GAS CUTTING: Must be in quenched and tempered condition.

MATERIAL SPECIFICATIONS: ASTM A 514-F

AVAILABILITY: Plate, bar, tubing, forgings, structural shapes, semi-finished products.

GENERAL: T-1 steel is a high-strength quenched and tempered structural steel with exceptionally good toughness at sub-zero temperatures. The alloying elements impart high strength to sections up to 8" thick while contributing high corrosion resistance. A high hardness grade can be specified (hardnesses of 321 to 360 Brinell) for impact abrasion resistance applications. Fabrication procedures for T-1 steel are available from most suppliers. Knowledge of these procedures is necessary since improper weld heat input, gas cutting, elevated forming temperatures, as well as improper heat treatment prior to flame cutting, welding, and final service application, can result in failure to obtain expected performance.

(Refs. VI-16,19-20,21,22)

Table B -4 Properties of USS T-1 Type A Steel

MECHANICAL PROPERTIES:

Y.S.	100 ksi
U.S.	110-130 ksi
Elong. % (in 2 in.)	16-18%
Red. of Area	35-50%
Hardness	235-293 Brinell
Fatigue Strength	50 ksi
Toughness	Charpy V-notch 15 ft-lbs at -50°F
Transition Temperature	-50°F

PHYSICAL PROPERTIES:

Modulus of Elasticity	29×10^6 psi
Shear Modulus	11×10^6 psi
Poisson's Ratio	0.30
Density	.283 lbs/in^3
Coefficient of Expansion	6.5×10^{-6} in/in/°F (-50 to +150°F)

Chemistry

C	Mn	P	S	Si	Cr	Mo	V	Ti	B
.12 / .21	.70 / 1.00	.035 max	.040 max	.20 / .35	.40 / .65	.15 / .25	.03 / .08	.01 / .03	.0005 / .005

HEAT TREATMENT: Water quenched from 1650-1750 and tempered at 1100-1275°F.

WELDABILITY: Must be in quenched and tempered condition.

ATMOSPHERIC CORROSION RESISTANCE: 2-3 times structural carbon steel.

MACHINABILITY: Similar to alloy steels of same hardness.

FORMABILITY: Can be cold formed to inside bend radii of 2-3 times the thickness, depending on thickness.

GAS CUTTING: Must be in quenched and tempered condition.

MATERIAL SPECIFICATIONS: ASTM A514-B

AVAILABILITY: Plate, bar, tubing, forgings, structural shapes, sheet.

GENERAL: T-1 type A (A514-B) steel is a high-strength quenched and tempered structural steel with a chemistry that makes it economical in thicknesses of 3/16" to 1¼". It is the lowest in cost of the T-1 steels. Fabrication procedures are similar to T-1 regular steel and need to be followed if maximum material performance is to be achieved. A maximum impact abrasion-resistant grade is available with a hardness of 321 to 360 Brinell, depending on thickness.

(Refs. VI-16,19,22,23)

Table B-5 Properties of USS T-1 Type B Steel

MECHANICAL PROPERTIES:

Y.S.	100 ksi
U.S.	110-130 ksi
Elong. % (in 2 in.)	16-18%
Red. of Area	35-50%
Hardness	235-293 Brinell
Fatigue Strength	50 ksi
Toughness	Charpy V-notch, 15 ft-lbs at -50°F
Transition Temperature	-50°F

PHYSICAL PROPERTIES:

Modulus of Elasticity	29×10^6 psi
Shear Modulus	11×10^6 psi
Poisson's Ratio	0.30
Density	.283 lbs/in^3
Coefficient of Expansion	6.5×10^{-6} in/in/°F (-50 to +150°F)

Chemistry

C	Mn	P	S	Si	Cr	Ni	Mo	V	B
.12/.21	.95/1.30	.035 max	.040 max	.20/.35	.40/.65	.30/.70	.20/.30	.03/.08	.0005 min

HEAT TREATMENT: Water quenched from 1650-1750°F and tempered at 1100-1275°F.

WELDABILITY: Must be in quenched and tempered condition.

ATMOSPHERIC CORROSION RESISTANCE: 2-3 times that of structural carbon steel.

MACHINABILITY: Similar to alloy steels of similar hardness.

FORMABILITY: Can be cold formed. Inside bend radii of 2 to 3 times the thickness, depending on thickness

GAS CUTTING: Must be in quenched and tempered condition.

MATERIAL SPECIFICATIONS: ASTM 514-H

AVAILABILITY: Plate, bar, tubing, forgings.

GENERAL: T-1 type B (A514-H) steel is a high-strength quenched and tempered structural steel with a chemistry that effects greater economies in section thicknesses from 1¼" to 2" than its other two T-1 family members. Fabrication procedures are similar to T-1 regular steel and need to be followed if maximum material performance is to be achieved. A maximum impact abrasion-resistant grade is available with a hardness of 321 to 360 Brinell, depending on thickness.

(Refs. VI-16,19,22)

Table B -6 Properties of Cor-Ten A

MECHANICAL PROPERTIES:

Y.S.	50 ksi
U.S.	70 ksi
Elong. % (in 2 in.)	22%
Red. of Area	
Hardness	156 Brinell
Fatigue Strength	38 ksi
Toughness	Charpy V-notch 15 ft-lb at -15°F
Transition Temperature	-15°F

PHYSICAL PROPERTIES:

Modulus of Elasticity	29×10^6 psi
Shear Modulus	11×10^6 psi
Poisson's Ratio	0.30
Density	.284 lbs/in^3
Coefficient of Expansion	6.5×10^{-6} in/in/°F (-50 to 150°F)

Chemistry

C	Mn	P	S	Si	Cu	Cr	Ni			
.12 max	.20 / .50	.07 / .15	.050 max	.25 / .75	.25 / .55	.30 / 1.25	.65 max			

HEAT TREATMENT:

WELDABILITY: Can be welded by all usual methods. Do not weld when metal temperature is below 50°F.

ATMOSPHERIC CORROSION RESISTANCE: 5-8 times that of structural carbon steel.

MACHINABILITY: Similar to carbon steels of equivalent hardness.

FORMABILITY: Cold form bend radii are dependent on thickness up to 1/2" thick. Hot forming is preferred for greater thicknesses.

GAS CUTTING: Preheating not required.

MATERIAL SPECIFICATIONS: ASTM A242 Type 1, A606 Type 4, and A618 Grade I, SAE J410c gd 950D.

AVAILABILITY: Plates, structural shapes, bars, structural tubing, sheets, strip.

GENERAL: Cor-Ten A is a popular HSLA steel because of its superior atmospheric corrosion resistance. Furthermore, it imparts twice the life to paint and protective coatings than that experienced on carbon steels. It can be supplied to the properties listed in sections up to 1/2" thick. Thicker sections are produced with a decrease in strength.

(Refs. VI-16,17,22,24)

Table B -7 Properties of Cor-Ten B

MECHANICAL PROPERTIES:

Y.S.	50 ksi
U.S.	70 ksi
Elong. % (in 2 in.)	21%
Red. of Area	
Hardness	156 Brinell
Fatigue Strength	38 ksi
Toughness	Charpy V-notch, 15 ft-lbs at -10°F
Transition Temperature	-10°F

PHYSICAL PROPERTIES:

Modulus of Elasticity	29×10^6 psi
Shear Modulus	11×10^6 psi
Poisson's Ratio	0.30
Density	.284 lbs/in^3
Coefficient of Expansion	6.5×10^{-6} in/in/°F (-50°F to +150°F)

Chemistry

C	Mn	P	S	Si	Cu	Cr	V			
.10 / .19	.90 / 1.25	.04 max	.05 max	.15 / .30	.25 / .40	.40 / .65	.02 / .10			

HEAT TREATMENT:

WELDABILITY: Can be welded by all usual methods. Preheating 50-200°F required, depending on thickness.

ATMOSPHERIC CORROSION RESISTANCE: Four times that of structural carbon steel.

MACHINABILITY: Similar to alloy steels of similar hardness.

FORMABILITY: Cold form bend radii are dependent on thickness up to 1/2" thick. Hot forming is preferred for greater thicknesses.

GAS CUTTING: Preheat is required prior to gas cutting.

MATERIAL SPECIFICATIONS: ASTM A588 Grade A, A709 Grade 50W.

AVAILABILITY: Plate, structural shapes, bar, structural tubing, sheets, strip.

GENERAL: Cor-Ten B maintains the mechanical properties given up to 4" thicknesses while Cor-Ten A only insures similar strengths up to 1/2" thicknesses. The atmospheric corrosion resistance of Cor-Ten B is not as good as Cor-Ten A but is still four times better than carbon steel. Paint life on Cor-Ten B is good.

(Refs. VI-16,17,22,24)

Table B-8 Properties of Cor-Ten C

MECHANICAL PROPERTIES:

Y.S.	60 ksi
U.S.	80 ksi
Elong. % (in 2 in.)	21%
Red. of Area	
Hardness	
Fatigue Strength	
Toughness	
Transition Temperature	

PHYSICAL PROPERTIES:

Modulus of Elasticity	29×10^6 psi
Shear Modulus	11×10^6 psi
Poisson's Ratio	0.30
Density	.284 lbs/in^3
Coefficient of Expansion	6.5×10^{-6} in/in/°F (-50°F to +150°F)

Chemistry

C	Mn	P	S	Si	Cu	Ni	Cr	V		
.12 / .19	.90 / 1.35	.04 max	.05 max	.15 / .30	.25 / .40	.65 max	.40 / .70	.04 / .10		

HEAT TREATMENT:

WELDABILITY: Can be welded by all usual methods. Preheating 50-100°F required, depending on thickness.

ATMOSPHERIC CORROSION RESISTANCE: Four times that of structural carbon steel.

MACHINABILITY:

FORMABILITY: Cold form bend radii are dependent on thicknesses up to 1/2" thick. Hot forming preferred for greater thicknesses.

GAS CUTTING: Generally not required if steel temperature is 50°F minimum.

MATERIAL SPECIFICATIONS: ASTM A 588

AVAILABILITY: Plate, bars, structural shapes.

GENERAL: Cor-Ten C is similar to Cor-Ten B in chemistry and corrosion resistance but is supplied in sections through only 1" thick. A minimum strength level of 60,000 psi is the result of fine-grain practice.

(Refs. VI-16,17,24)

Table B -9 Properties of USS AR Steel

MECHANICAL PROPERTIES:

Y.S.	
U.S.	97-119 ksi
Elong. % (in 2 in.)	
Red. of Area	
Hardness	200-250 Brinell
Fatigue Strength	
Toughness	
Transition Temperature	

PHYSICAL PROPERTIES:

Modulus of Elasticity	29×10^6 psi
Shear Modulus	11×10^6 psi
Poisson's Ratio	0.30
Density	.283 lbs/in^3
Coefficient of Expansion	6.3×10^{-6} in/in/°F (68 - 212°F)

Chemistry

C	Mn	P	S	Si					
0.35 / 0.50	1.50 / 2.00	.05 max	.055 max	.15 / .35					

HEAT TREATMENT:

WELDABILITY: Preheat to 300-400°F prior to welding, depending on thickness. Post-weld stress relief may be done at 1025°F.

ATMOSPHERIC CORROSION RESISTANCE: Similar to plain carbon steel.

MACHINABILITY:

FORMABILITY: Can be formed cold up to 1/2" thick. Hot forming done up to 1500°F.

GAS CUTTING: Preheat to 300-400°F prior to cutting, depending on thickness.

MATERIAL SPECIFICATIONS: Proprietary - USS AR Abrasion-Resistant Steel

AVAILABILITY: Plate to 2-inch thick.

GENERAL: USS AR steel is used in the as-rolled condition for smooth-sliding abrasion resistance applications.

(Refs. VI-16,17,25)

Table B-10 Properties of USS AR 350 Steel

MECHANICAL PROPERTIES:

Y.S.	
U.S.	152 ksi
Elong. % (in 2 in.)	
Red. of Area	
Hardness	321 Brinell
Fatigue Strength	
Toughness	
Transition Temperature	

PHYSICAL PROPERTIES:

Modulus of Elasticity	29×10^6 psi
Shear Modulus	11×10^6 psi
Poisson's Ratio	0.30
Density	.284 lbs/in^3
Coefficient of Expansion	6.5×10^{-6} in/in/°F (68 - 212°F)

Chemistry

C	Mn	P	S	Si	B					
0.32 max	1.60 max	.04 max	.05 max	.60 max	.0005 min					

HEAT TREATMENT: Quenched and tempered.

WELDABILITY: Preheat to 200-350°F, depending on thickness. Stress relieving not recommended.

ATMOSPHERIC CORROSION RESISTANCE: Similar to plain carbon steel.

MACHINABILITY:

FORMABILITY: Is not easily formed cold, large bend radii required; hot forming not recommended.

GAS CUTTING: Preheat to 200-350°F, depending on thickness.

MATERIAL SPECIFICATIONS: Proprietary - USS AR 350 Abrasion-Resistant Steel

AVAILABILITY: Plate to 1¼-inch thick

GENERAL: AR 350 is a quenched and tempered steel that is used for impact and sliding abrasion resistance.

(Refs. VI-16,17,25)

Table B-11 Properties of USS AR 360 Steel

MECHANICAL PROPERTIES:

Y.S.	
U.S.	175 ksi
Elong. % (in 2 in.)	
Red. of Area	
Hardness	360 Brinell
Fatigue Strength	
Toughness	
Transition Temperature	

PHYSICAL PROPERTIES:

Modulus of Elasticity	29×10^6 psi
Shear Modulus	11×10^6 psi
Poisson's Ratio	0.30
Density	.284 lbs/in^3
Coefficient of Expansion	6.5×10^{-6} in/in/°F (68 - 212°F)

Chemistry

C	Mn	P	S	Si	Cr	Mo	B		
0.26 / 0.33	1.15 / 1.50	.035 max	.040 max	.20 / .35	.40 / .65	.08 / .15	.0005 min		

HEAT TREATMENT: Quenched and tempered

WELDABILITY: Same as for AR 350

ATMOSPHERIC CORROSION RESISTANCE: Similar to plain carbon steels.

MACHINABILITY:

FORMABILITY: Liberal bend radii required for cold forming. Hot forming not recommended.

GAS CUTTING: Same as for AR 350

MATERIAL SPECIFICATIONS: Proprietary - USS AR 360 Abrasion-Resistant Steel.

AVAILABILITY: Plate to 1½-inch thick

GENERAL: AR 360 is similar to AR 350 except it possesses greater hardness and can maintain these hardnesses to thicker sections due to its alloying modification.

(Refs. VI- 16,17,25)

Table B-12. Expected Minimum Mechanical Properties, Conventional Practice, of Cold-Drawn Carbon Steel Rounds, Squares, and Hexagons.

The tensile and yield strengths of carbon-steel bars are improved by cold drawing. By comparison, the tensile strength of hot-rolled bars is about 10% less, and their yield strength is some 40% less. For example, a low-carbon steel with a yield-to-tensile-strength ratio of about 0.55 in the form of hot-rolled bars will have a ratio of about 0.85 after cold drawing. While there is some sacrifice in elongation, reduction in area, and impact strength, these changes are relatively insignificant in most structural applications or engineering components.

This improvement is of interest to the design engineer seeking a better strength-to-weight ratio or a reduction in costs by the elimination of alloy contents and heat treatment. The enhanced properties may also be useful in applications involving threads, notches, cut-outs, and in other design requirements that might effect strength adversely.

Turned and polished and turned, ground, and polished bars have the mechanical properties of hot-rolled bars.

AISI No. Size, In.	As Cold Drawn					Cold Drawn Followed by Low-Temperature Stress Relief					Cold Drawn Followed by High-Temperature Stress Relief				
	Strength					Strength					Strength				
	Tensile, 1,000 Psi	Yield, 1,000 Psi	Elongation in 2 In., %	Reduction in Area, %	Hardness, Bhn	Tensile, 1,000 Psi	Yield, 1,000 Psi	Elongation in 2 In., %	Reduction in Area, %	Hardness, Bhn	Tensile, 1,000 Psi	Yield, 1,000 Psi	Elongation in 2 In., %	Reduction in Area, %	Hardness, Bhn
1018, 1025															
⅝ to ⅞ incl.	70	60	18	40	143						65	45	20	45	131
Over ⅞ to 1¼ incl.	65	55	16	40	131						60	45	20	45	121
Over 1¼ to 2 incl.	60	50	15	35	121						55	45	16	40	111
Over 2 to 3 incl.	55	45	15	35	111						50	40	15	40	101
1117, 1118															
⅝ to ⅞ incl.	75	65	15	40	149	80	70	15	40	163	70	50	18	45	143
Over ⅞ to 1¼ incl.	70	60	15	40	143	75	65	15	40	149	65	50	16	45	131
Over 1¼ to 2 incl.	65	55	13	35	131	70	60	13	35	143	60	50	15	40	121
Over 2 to 3 incl.	60	50	12	30	121	65	55	12	35	131	55	45	15	40	111
1035															
⅝ to ⅞ incl.	85	75	13	35	170	90	80	13	35	179	80	60	16	45	163
Over ⅞ to 1¼ incl.	80	70	12	35	163	85	75	12	35	170	75	60	15	45	149
Over 1¼ to 2 incl.	75	65	12	35	149	80	70	12	35	163	70	60	15	40	143
Over 2 to 3 incl.	70	60	10	30	143	75	65	10	30	149	65	55	12	35	131
1040, 1140															
⅝ to ⅞ incl.	90	80	12	35	179	95	85	12	35	187	85	65	15	45	170
Over ⅞ to 1¼ incl.	85	75	12	35	170	90	80	12	35	179	80	65	15	45	163
Over 1¼ to 2 incl.	80	70	10	30	163	85	75	10	30	170	75	60	15	40	149
Over 2 to 3 incl.	75	65	10	30	149	80	70	10	30	163	70	55	12	35	143
1045, 1146, 1145															
⅝ to ⅞ incl.	95	85	12	35	187	100	90	12	35	197	90	70	15	45	179
Over ⅞ to 1¼ incl.	90	80	11	30	179	95	85	11	30	187	85	70	15	45	170
Over 1¼ to 2 incl.	85	75	10	30	170	90	80	10	30	179	80	65	15	40	163
Over 2 to 3 incl.	80	70	10	30	163	85	75	10	25	170	75	60	12	35	149
1050, 1137, 1151															
⅝ to ⅞ incl.	100	90	11	35	197	105	95	11	35	212	95	75	15	45	187
Over ⅞ to 1¼ incl.	95	85	11	30	187	100	90	11	30	197	90	75	15	40	179
Over 1¼ to 2 incl.	90	80	10	30	179	95	85	10	30	187	85	70	15	40	170
Over 2 to 3 incl.	85	75	10	30	170	90	80	10	25	179	80	65	12	35	163
1141															
⅝ to ⅞ incl.	105	95	11	30	212	110	100	11	30	223	100	80	15	40	197
Over ⅞ to 1¼ incl.	100	90	10	30	197	105	95	10	30	212	95	80	15	40	187
Over 1¼ to 2 incl.	95	85	10	30	187	100	90	10	25	197	90	75	15	40	179
Over 2 to 3 incl.	90	80	10	20	179	95	85	10	20	187	85	70	12	30	170
1144															
⅝ to ⅞ incl.	110	100	10	30	223	115	105	10	30	229	105	85	15	40	212
Over ⅞ to 1¼ incl.	105	95	10	30	212	110	100	10	30	223	100	85	15	40	197
Over 1¼ to 2 incl.	100	90	10	25	197	105	95	10	25	212	95	80	15	35	187
Over 2 to 3 incl.	95	85	10	20	187	100	90	10	20	197	90	75	12	30	179

Source: AISI Committee of Hot Rolled and Cold Finished Bar Producers.

Table B-13. Estimated Mechanical Properties and Machinability of Hot Rolled and Cold Finished Carbon Steel Bars (SAE)

SAE and AISI No.	Type of processing	Estimated minimum values: Tensile strength, psi	Yield strength, psi	Elongation in 2 in., %	Reduction in area, %	Brinell hardness	Average machinability rating*
Carbon Steel Bars							
1006	Hot rolled	43,000	24,000	30	55	86	
	Cold drawn	48,000	41,000	20	45	95	50
1008	Hot rolled	44,000	24,500	30	55	86	
	Cold drawn	49,000	41,500	20	45	95	55
1009	Hot rolled	43,000	24,000	30	55	86	
	Cold drawn	48,000	41,000	20	45	95	50
1010	Hot rolled	47,000	26,000	28	50	95	
	Cold drawn	53,000	44,000	20	40	105	55
1012	Hot rolled	48,000	26,500	28	50	95	
	Cold drawn	54,000	45,000	19	40	105	55
1015	Hot rolled	50,000	27,500	28	50	101	
	Cold drawn	56,000	47,000	18	40	111	60
1016	Hot rolled	55,000	30,000	25	50	111	
	Cold drawn	61,000	51,000	18	40	121	70
1017	Hot rolled	53,000	29,000	26	50	105	
	Cold drawn	59,000	49,000	18	40	116	65
1018	Hot rolled	58,000	32,000	25	50	116	
	Cold drawn	64,000	54,000	15	40	126	70
1019	Hot rolled	59,000	32,500	25	50	116	
	Cold drawn	66,000	55,000	15	40	131	70
1020	Hot rolled	55,000	30,000	25	50	111	
	Cold drawn	61,000	51,000	15	40	121	65
1021	Hot rolled	61,000	33,000	24	48	116	
	Cold drawn	68,000	57,000	15	40	131	70
1022	Hot rolled	62,000	34,000	23	47	121	
	Cold drawn	69,000	58,000	15	40	137	70
1023	Hot rolled	56,000	31,000	25	50	111	
	Cold drawn	62,000	52,500	15	40	121	65
1024	Hot rolled	74,000	41,000	20	42	149	
	Cold drawn	82,000	69,000	12	35	163	60
1025	Hot rolled	58,000	32,000	25	50	116	
	Cold drawn	64,000	54,000	15	40	126	65
1026	Hot rolled	64,000	35,000	24	49	126	
	Cold drawn	71,000	60,000	15	40	143	75
1027	Hot rolled	75,000	41,000	18	40	149	
	Cold drawn	83,000	70,000	12	35	163	65
1030	Hot rolled	68,000	37,500	20	42	137	
	Cold drawn	76,000	64,000	12	35	149	70
1033	Hot rolled	72,000	39,500	18	40	143	
	Cold drawn	80,000	67,000	12	35	163	70
1035	Hot rolled	72,000	39,500	18	40	143	
	Cold drawn	80,000	67,000	12	35	163	65
1036	Hot rolled	83,000	45,500	16	40	163	
	Cold drawn	92,000	77,500	12	35	187	55
1037	Hot rolled	74,000	40,500	18	40	143	
	Cold drawn	82,000	69,000	12	35	167	65
1038	Hot rolled	75,000	41,000	18	40	149	
	Cold drawn	83,000	70,000	12	35	163	65
1039	Hot rolled	79,000	43,500	16	40	156	
	Cold drawn	88,000	74,000	12	35	179	60
1040	Hot rolled	76,000	42,000	18	40	149	
	Cold drawn	85,000	71,000	12	35	170	60
1041	Hot rolled	92,000	51,000	15	40	187	
	Cold drawn	102,500	87,000	10	30	207	45
	ACD(a)	94,000	80,000	10	45	184	65
1042	Hot rolled	80,000	44,000	16	40	163	
	Cold drawn	89,000	75,000	12	35	179	60
	NCD(b)	85,000	73,000	12	45	179	70
1043	Hot rolled	82,000	45,000	16	40	163	
	Cold drawn	91,000	77,000	12	35	179	60
	NCD(b)	87,000	75,000	12	45	179	70
1045	Hot rolled	82,000	45,000	16	40	163	
	Cold drawn	91,000	77,000	12	35	179	55
	ACD(a)	85,000	73,000	12	45	170	65
1046	Hot rolled	85,000	47,000	15	40	170	
	Cold drawn	94,000	79,000	12	35	187	55
	ACD(a)	90,000	75,000	12	45	174	65
1049	Hot rolled	87,000	48,000	15	35	179	
	Cold drawn	97,000	81,500	10	30	197	45
	ACD(a)	92,000	77,000	10	40	187	55
1050	Hot rolled	90,000	49,500	15	35	179	
	Cold drawn	100,000	84,000	10	30	197	45
	ACD(a)	95,000	80,000	10	40	189	55
1052	Hot rolled	108,000	59,500	12	30	217	
	ACD(a)	98,000	83,000	10	40	193	50
1055	Hot rolled	94,000	51,500	12	30	192	
	ACD(a)	96,000	81,000	10	40	197	55
1060	Hot rolled	98,000	54,000	12	30	201	
	SACD(c)	90,000	70,000	10	45	183	60
1064	Hot rolled	97,000	53,500	12	30	201	
	SACD(c)	89,000	69,000	10	45	183	60
1065	Hot rolled	100,000	55,000	12	30	207	
	SACD(c)	92,000	71,000	10	45	187	60
1070	Hot rolled	102,000	56,000	12	30	212	
	SACD(c)	93,000	72,000	10	45	192	55
1074	Hot rolled	105,000	58,000	12	30	217	
	SACD(c)	94,500	73,000	10	40	192	55
1078	Hot rolled	100,000	55,000	12	30	207	
	SACD(c)	94,000	72,500	10	40	192	55
1080	Hot rolled	112,000	61,500	10	25	229	
	SACD(c)	98,000	75,000	10	40	192	45
1084	Hot rolled	119,000	65,500	10	25	241	
	SACD(c)	100,000	77,000	10	40	192	45
1085	Hot rolled	121,000	66,500	10	25	248	
	SACD(c)	100,500	78,000	10	40	192	45
1086	Hot rolled	112,000	61,500	10	25	229	
	SACD(c)	97,000	74,000	10	40	192	45
1090	Hot rolled	122,000	67,000	10	25	248	
	SACD(c)	101,000	78,000	10	40	197	45
1095	Hot rolled	120,000	66,000	10	25	248	
	SACD(c)	99,000	76,000	10	40	197	45
Resulfurized Carbon Steel Bars(d)							
1111	Hot rolled	55,000	33,000	25	45	121	
	Cold drawn	75,000	58,000	10	35	163	95
1112	Hot rolled	56,000	33,500	25	45	121	
	Cold drawn	78,000	60,000	10	35	167	100
1113	Hot rolled	56,000	33,500	25	45	121	
	Cold drawn	78,000	60,000	10	35	167	135
12L14	Hot rolled	57,000	34,000	22	45	121	
	Cold drawn	78,000	60,000	10	35	163	160
1108	Hot rolled	50,000	27,500	30	50	101	
	Cold drawn	56,000	47,000	20	40	121	80
1109	Hot rolled	50,000	27,500	30	50	101	
	Cold drawn	56,000	47,000	20	40	121	80
1115	Hot rolled	55,000	30,000	25	50	111	
	Cold drawn	61,000	51,000	20	40	121	80
1117	Hot rolled	62,000	34,000	23	47	121	
	Cold drawn	69,000	58,000	15	40	137	90
1118	Hot rolled	65,000	36,000	23	47	131	
	Cold drawn	72,000	61,000	15	40	143	85
1119	Hot rolled	62,000	34,000	23	47	121	
	Cold drawn	69,000	58,000	15	40	137	100
1120	Hot rolled	62,000	34,000	23	47	121	
	Cold drawn	69,000	58,000	15	40	137	80
1126	Hot rolled	64,000	35,000	23	47	126	
	Cold drawn	71,000	59,500	15	40	143	80
1132	Hot rolled	83,000	45,500	16	40	167	
	Cold drawn	92,000	77,000	12	35	183	75
1137	Hot rolled	86,000	48,000	15	35	179	
	Cold drawn	98,000	82,000	10	30	197	70
1138	Hot rolled	73,000	40,000	18	40	149	
	Cold drawn	81,000	68,000	12	35	156	75
1140	Hot rolled	79,000	43,500	16	40	156	
	Cold drawn	88,000	74,000	12	35	170	70
1141	Hot rolled	94,000	51,500	15	35	187	
	Cold drawn	105,100	88,000	10	30	212	70
1144	Hot rolled	97,000	53,000	15	35	197	
	Cold drawn	108,000	90,000	10	30	217	80
1145	Hot rolled	85,000	47,000	15	40	170	
	Cold drawn	94,000	80,000	12	35	187	65
1146	Hot rolled	85,000	47,000	15	40	170	
	Cold drawn	94,000	80,000	12	35	187	70
1151	Hot rolled	92,000	50,500	15	35	187	
	Cold drawn	102,000	86,000	10	30	207	65

*Cold drawn 1112 steel = 100%.

(a) ACD, annealed cold drawn. (b) NCD, normalized cold drawn. (c) SACD, spheroidize annealed cold drawn. (d) All SAE 1100 series steels are rated on the basis of 0.10% max Si or coarse grain melting practice.

Mechanical properties listed above are given as a matter of general information. They do not form a part or requirement of any specification unless each instance is approved by the source of supply. The properties shown can generally be expected from bars in sizes ranging from ¾ to 1¼ in., based on the standard round tension-test specimen with 2-in. gage length. Sizes under ¾ in. will have slightly higher strength than those shown.

Properties of turned and polished or turned and ground types of cold finished material will correspond to the hot rolled values.

The properties of cold drawn steels are based on conventional production from hot rolled bars. These properties may be varied by modified cold drawing practices or a combination of cold drawing plus heat treatment for grades 1050 and lower. Grades higher in carbon than 1050 are commonly annealed before cold drawing.

Machinability ratings listed are based on a value of 100% for cold drawn SAE 1112. This value involves turning at a cutting speed of 180 sfm for feeds up to 0.007 ipr and depths of cut up to 0.250 in., using appropriate cutting fluids with tools of high speed steel T1 (18-4-1) hardened to Rockwell C 63 to 65.

Relative machinability data shown were obtained from experimental and actual shop production information, machining cold drawn bars on single and multiple-spindle automatic machines. Various factors influence machinability and, therefore, results shown are average and may be affected to some degree by amount of cold reduction, mechanical properties, grain size and microstructure.

From the 1960 SAE Handbook

Table B-14 Properties of 1018 Cold Drawn Carbon Steel

MECHANICAL PROPERTIES:

Y.S.	45-60 ksi
U.S.	55-70 ksi
Elong. % (in 2 in.)	15-18%
Red. of Area	35-40%
Hardness	111-143 Brinell
Fatigue Strength	35 ksi
Toughness	Charpy V-notch, 75-150 ft-lbs at 100°F
Transition Temperature	0-50°F

PHYSICAL PROPERTIES:

Modulus of Elasticity	29×10^6 psi
Shear Modulus	11×10^6 psi
Poisson's Ratio	0.30
Density	0.284 lbs/in^3
Coefficient of Expansion	6.5×10^{-6} in/in/°F (68 - 212°F)

Chemistry

C	Mn	P	S						
.15/.20	.60/.90	.040 max	.050 max						

HEAT TREATMENT: Post draw stress relief temperature, 700-950°F.

WELDABILITY: Welding destroys cold drawn properties in heat-affected zone.

ATMOSPHERIC CORROSION RESISTANCE: That of plain carbon steel.

MACHINABILITY: Good

FORMABILITY:

GAS CUTTING:

MATERIAL SPECIFICATIONS: AISI C1018

AVAILABILITY: Rounds, squares, hexagons, and flats.

GENERAL: C1018 is cold drawn to achieve a smooth surface, closer dimensional tolerances, and increased strength over the as-rolled condition. Economies can often be achieved with this grade since the cost of cold drawing often offsets the cost of alloying to achieve equal strength.

(Refs. VI-16,17,26)

Table B-15 Properties of 1040 Cold Drawn Carbon Steel

MECHANICAL PROPERTIES:

Y.S.	65-80 ksi
U.S.	75-90 ksi
Elong. % (in 2 in.)	10-12%
Red. of Area	30-35%
Hardness	149-179 Brinell
Fatigue Strength	40-50 ksi
Toughness	Charpy V-notch, 15-25 ft-lbs at 100°F
Transition Temperature	140-170°F

PHYSICAL PROPERTIES:

Modulus of Elasticity	29×10^6 psi
Shear Modulus	11×10^6 psi
Poisson's Ratio	0.30
Density	.283 lbs/in^3
Coefficient of Expansion	6.3×10^{-6} in/in/°F (68 - 212°F)

Chemistry

C	Mn	P	S						
.37 .44	.60 .90	.040 max	.050 max						

HEAT TREATMENT: Post draw stress relief temperature, 700-900°F.

WELDABILITY:

ATMOSPHERIC CORROSION RESISTANCE: That of plain carbon steel.

MACHINABILITY: Good

FORMABILITY: Normally used in the as-drawn condition

GAS CUTTING:

MATERIAL SPECIFICATIONS: AISI C 1040

AVAILABILITY: Rounds, squares, hexagons, and flats.

GENERAL: C 1040 is cold drawn to achieve a smooth surface, closer dimensional tolerances, and increased strength over the as-rolled condition. It is similar to C1018 in that high strength is achieved through drawing rather than alloying, thus effecting cost economies. Welding of 1040 steel requires a preheat and postweld stress relief treatment. The recommended stress relief temperature is 1200°F, and this temperature will reduce the cold drawn properties of cold drawn 1040. In light of this, it appears that welding of cold drawn 1040 steel should be avoided.

(Refs. VI- 16,17,26,27)

APPENDIX C

ACKNOWLEDGEMENT OF PARTICIPATING MINES

Table C-1 Mines Visited and Contacted

	MINE	COMPANY	LOCATION	PERSONNEL CONTACTED
1.	Lucky Friday	Hecla Mining Co.	Mullen, ID	W. E. Crandall M. P. Gross
2.	Sunshine	Sunshine Mining Co.	Kellogg, ID	Ken Castleton George Vasiloff
3.	Crescent	Bunker Hill Co.	Kellogg ID	Robert Miller Gene Wasson Gerald Furnish
4.	Galena	ASARCO, Inc.	Wallace, ID	Fred Owsley Jerry Christian
5.	Coeur	ASARCO, Inc.	Wallace, ID	Fred Owsley Carroll Ward
6.	Bill Smith	Kerr-McGee Nuclear Corporation	Casper, WY	Tom Hart Rod Tregembo
7.	Highlands	Exxon Co., U.S.A.	Casper, WY	Wiley Brooks Bill Taylor Jim Lonergan Sam Bradley
8.	Henderson	Climax	Empire, CO	Jim Crosby Bruce Stanley L. Van Scoyk
9.	Superior	Magma Copper Co.	Superior, AZ	J. W. Murray F. M. Florez Dick Scholl Gayle Botkin
10.	Miami East	Cities Service Co.	Miami, AZ	John Brandon
11.	Buick	Amax Lead & Zinc	Boss, MO	Walter Dean
12.	Ozark Lead	Ozark Lead Company	Elington, MO	Louis Fisher
13.	Elmwood	New Jersey Zinc Co.	Elmwood, TN	Art Bernholdt Richard Dendler
14.	U.S.Pipe No 3	Jim Walter Resources, Inc.	Adger, AL	G. J. Hager Cass Mical
15.	Viburnum	St. Joe Minerals Corp.	Viburnum, MO	Carl L. Smith
16.	Lakeshore	Hecla Mining Co.	Casa Grande,AZ	Ron Peterson Steve Milne

	MINE	COMPANY	LOCATION	PERSONNEL
17.	Ontario	Park City Ventures	Park City, UT	L. J. Maki R. Hardin
18.	Lisbon	Rio Algom Corp.	San Juan, UT	Joe Vansul Guy Bennett
19.	Burgin	Kennecott Copper	Eureka, UT	Paul Hunter Tom Tappa Tim Hannifin Don Powell
20.	Saskatoon	Potash Co. of Amer.	Saskatoon Sask., Canada	A. U. Miacek
21.	Camp Bird	Camp Bird Colo,Inc.	Ouray, CO	
22.	Kleer	Morton Salt Co.	Grand Sabine, TX	
23.	Bulldog	Homestake Mining	Creede, CO	
24.	Leadville Unit	ASARCO, Inc.	Leadville, CO	
25.	Kimballton	Gold Bond Bldg. Products	Kimballton, VA	
26.	Eagle	New Jersey Zinc	Gilman, CO	
27.	Sherman	Day Mines, Inc.	Leadville, CO	
28.	Sunnyside	Standard Metals	Silverton, CO	
29.	Rico Argentine	Rico Argentine Mining Co.	Rico, CO	
30.	Deremo (D-1)	Union Carbide Corp	Uravan, CO	
31.	Silver Bell	Union Carbide Corp	Uravan, CO	
32.	Deremo (D-2)	Union Carbide Corp	Uravan, CO	
33.	Wilson	Union Carbide Corp	Uravan, CO	
34.	Snyder	Union Carbide Corp	Uravan, CO	
35.	Bruce	Cyprus Mines Corp	Bagdad, AZ	
36.	Magmont	Cominco American	Bixby, MO	
37.	Grand Rapids Gypsum	Grand Rapids Gyp-sum Co.	Grand Rapids, MI	
38.	Hobbs Potash	Kerr-McGee Chemical	Hobbs, NM	
39.	Pilot Knobb Pellet	Hanna Mining Co.	Ironton, MO	

	MINE	COMPANY	LOCATION
40.	Cleveland Mine	International Salt Company	Cleveland, OH
41.	Black Hills	Homestake Mining Co	Lead, SD
42.	Operations	Gouverneur Talc Co	Talcville, NY
43.	Buick	AMAX Lead Co. of MO	Boss, MO
44.	Pend Oreille	The Bunker Hill Co	Metaline Fall, WA
45.	Detroit Mine	International Salt Company	Detroit, MI
46.	Mt. Hope Iron	Mt. Hope Mining Co	Mt. Hope, NJ
47.	Nash Draw	Duval Corp.	Carlsbad, NM
48.	Big Island	Stauffer Chemicals	Green River, WY
49.	Operations	Potash of America	Carlsbad, NM
50.	Operations	Globe Refractories	Newell, WV
51.	White Pine	White Pine Copper	White Pine, MI
52.	Operations	Federal-American Partners	Gas Hills, WY
53.	Section 32 Mine	United Nuc. Homesk. Partners	Ambrosia Lake, NM
54.	Section 25 Mine	United Nuc. Homstk. Part.	Ambrosia Lake, NM
55.	Section 15 Mine	United Nuc. Homstk. Part.	Ambrosia Lake, NM
56.	Section 23 Mine	United Nuc. Homstk. Part.	Ambrosia Lake, NM
57.	Ground Hog	ASARCO, Inc.	Vanadium, NM
58.	Morton Salt	Morton Salt Co.	Weeks, LA
59.	Carlsbad	Mississippi Chem.	Carlsbad, NM
60.	Pine Creek Operations	Union Carbide Corp	Bishop, GA
61.	Kelley	Anaconda Company	Butte, MT

	MINE	COMPANY	LOCATION
62.	Steward Mine	Anaconda Company	Butte, MT
63.	Minerva Mines	Allied Chemical	Cave-in-Rock, IL
64.	Grace	Bethlehem Mines	Morgantown, PA
65.	Rural Branch	Ozark Lead Co.	Sweetwater, MO
66.	Balmat No.2	St. Joe Minerals	Balmat, NY
67.	Balmat No.3	St. Joe Minerals	Balmat, NY
68.	Balmat No.4	St. Joe Minerals	Balmat, NY
69.	Crane Shaft	St. Joe Minerals	Balmat, NY
70.	Vertical Shaft	St. Joe Minerals	Balmat, NY
71.	Elmwood	Jersey Miniere Zinc	Elmwood, TN
72.	Green River	FMC Corp.	Green River, WY