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DEVELOPMENT OF LIGHTWEIGHT MINE ROOF SUPPORTS

Prepared For

**UNITED STATES DEPARTMENT OF THE INTERIOR
BUREAU OF MINES**

Prepared By

**FLUID POWER RESEARCH CENTER
OKLAHOMA STATE UNIVERSITY
STILLWATER, OKLAHOMA**

FINAL REPORT

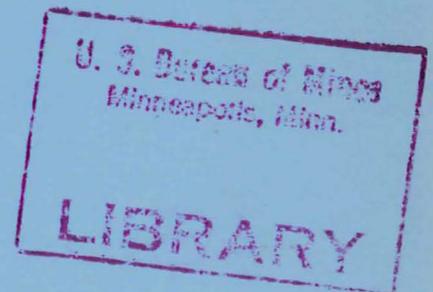
October 1977 – May 1978

Interagency Agreement No. H0272017

U.S. Army Contract No. DAAK70-77-C-0097 (Modification)

Contract Administered By

**U.S. ARMY MERADCOM
Fort Belvoir, Virginia**



Submitted

August 1978



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16. Abstract (Limit: 200 words) This report presents the results of a study to develop optimal designs for light-weight mine roof supports for use in underground coal mines. A load capacity of 196 kN (22 tons) and a height adjustable between 1.829 m (72 inches) and 2.438 m (96 inches) were taken as the design parameters and the design goal was a unit weighing less than 22.3 kg (50 lbs). Three alternative design concepts were explored, viz. mechanical, hydraulic, and hydro-mechanical and a total of 13 designs analyzed in detail. Steel, aluminum alloy and composites were the main potential construction materials investigated for suitability from structural, life, and safety and other considerations. The designs considered most promising for development were: (i) the mechanical screw-pin type using telescopic tubes and a fine adjustment screw thread section (ii) the all-hydraulic type, of steel, aluminum alloy or aramid graphite construction (iii) the hydro-mechanical design using telescopic tubes and a short stroke hydraulic cylinder for fine adjustment It is shown that by using realistic mathematical models for stress and deflection calculations, it is possible to reduce prop weight to the design goal. The analysis presented for all-hydraulic props is applicable to hydraulic cylinders on all mining equipment and similar machinery. This report includes the computer program and user's guide for analyzing hydraulic cylinders as well as a cross-indexed bibliography on structural composites.				
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FOREWORD

This report was prepared by the Fluid Power Research Center, Oklahoma State University, Stillwater, Oklahoma, for the U.S. Bureau of Mines under contract no. DAAK70-77-C-0097 of the U.S. Army Mobility Equipment Research and Development Command. The contract was administered under the technical direction of the Spokane Mining Research Center with Mr. Webster Anderson acting as Technical Project Officer. The contract was administered by Mr. D. L. Vila of the Denver office, U.S.B.M., through the U.S.A. MERADCOM. Mr. Henry Schaefer of the U.S.A. MERADCOM was the project monitor. This report is a summary of the work recently completed for the contract, and was performed during the period of 1 October 1977 to 31 May 1978. This report was submitted by the authors in May, 1978.

No patents or inventions have resulted from this study.

Reference to specific brands, equipment or trade names in this report is made to facilitate understanding and does not imply endorsement by the U.S. Bureau of Mines.

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EXECUTIVE SUMMARY

The objective of this contract was to develop optimal designs for lightweight roof supports for use in underground coal mines. A load capacity of 196 kN (22 tons), height adjustability between 1.829 m (72 inches) and 2.438 m (96 inches) were the prescribed design parameters, and the design goal was a unit weighing less than 22.3 kg (50 lbs.). Five optimal designs were to be developed in sufficient detail to permit prototype construction, and designs were to be evaluated not only from the point of view of structural integrity, but also other considerations, such as ease of handling and installation, safety, operation under mine environmental conditions, MSHA approval, maintenance and ease of manufacture.

Current designs of mine props were scrutinized and an understanding of their use in underground coal mines gained by personal visits. Literature on constructional material and fabrication techniques was also collected and analyzed.

Three alternative design concepts were explored, viz mechanical, hydraulic and hydro-mechanical, and in each category two or more designs developed in detail. Methods of coarse and fine adjustment of height, reservoir and power unit location and construction material were the variables used to arrive at the thirteen alternative designs.

Mechanical props and the short stroke hydraulic cylinders in hydro-mechanical props were analyzed by conventional stress calculations, but the all-hydraulic props were optimized by using a computerized analysis technique developed at the Fluid Power Research Center, Oklahoma State University. Steel, aluminum alloy and composites were the main potential

construction materials investigated for suitability and cost effectiveness.

The designs considered most promising for development are the following:

(i) The mechanical screw-pin type using telescopic tubes for rough height adjustment and a screw thread section for fine adjustment, and a liquid or elastomer spring for overload control. Steel or aluminum alloy would be equally suitable for fabrication using conventional techniques.

(ii) The all-hydraulic prop using an integral power unit and fluid reservoir. Using more realistic stress and deflection analysis than is done currently, the weight of these props can be substantially reduced even when steel or aluminum alloy is the construction material. Even more reduction in dry weight is achieved when hybrid composites such as aramid-graphite are used. In such props the weight of the fluid can be as much as one half that of the dry prop.

(iii) The hydro-mechanical type using telescopic tubes for rough adjustment and a short hydraulic cylinder for fine adjustment and overload control. Such a design would not carry as much "dead" fluid as the all-hydraulic prop, yet have most of the advantages of the all-hydraulic prop. If constructed of a composite, it could be well within the design goal of 22.3 kg (50 lbs.).

The actual capacity of a prop in service depends on the manner of application and handling in the mining environment. The design of all types of props can be improved by studying field usage and especially ascertaining eccentricity of loading and end-moments. If these factors are properly accounted for, the nominal factor of safety used in the load

calculations can be reduced without detracting from their true safety margin.

It is concluded that props of the three types described above should be detail-designed, manufactured and tested. In view of the changes in relative material costs and technological advances, it is difficult to predict which design will be cost-optimal. Based on a survey of current raw material costs and state-of-the-art in fabrication, the steel and aluminum alloy props are estimated to cost approximately the same, and the aramid graphite prop about four times as much. These cost ratios can be expected to change significantly as composites find more and more uses.

T A B L E O F C O N T E N T S

<i>Chapter</i>		<i>Page</i>
I	INTRODUCTION	1
II	REVIEW OF CURRENT DESIGNS	5
III	ALTERNATIVE DESIGNS	15
	Pressurization Capability	24
	Design Requirements	25
	Buckling Resistance	28
IV	OPTIMAL DESIGN	33
	Mechanical	38
	Hydraulic	41
	Fiber Composites in Roof Supports	66
	Other Design Characteristics	73
	Addendum	81
V	CONCLUSIONS & RECOMMENDATIONS	85
	REFERENCES	91
 <i>Appendix</i>		
A	ANNOTATED BIBLIOGRAPHY & CROSS-REFERENCE	95
B	SIZE & WEIGHT CALCULATIONS	117
C	COMPUTER PROGRAM FOR CYLINDER ANALYSIS (SACREG)	131
D	USER'S GUIDE TO SACREG	135
E	FORTRAN LISTING OF SACREG	145
F	PROTECTION OF LIGHT-WEIGHT HYDRAULIC PROPS AGAINST ACCIDENTAL DAMAGE BY LATERAL LOADING	177
G	OVERLOAD PROTECTION IN MECHANICAL PROPS	183
H	WEIGHT OPTIMIZATION OF ALL-HYDRAULIC PROPS	189

L I S T O F F I G U R E S

<i>Figure</i>		<i>Page</i>
2-1	Gullick-Dobson 40 Tonne Quick Yield Prop Gun Set [3]	6
2-2	Dowty Hydraulic Prop [8]	8
2-3	Dowty Main Piston Head [8]	10
2-4	Dowty Design Features [9]	11
2-5	Hy-Jax Design Features [9]	12
3-1	Cylinder Weight vs Operating Pressure	27
4-1	Mechanical-Wedge Design 1	39
4-2	Mechanical-Screw Design 2	40
4-3	Flow Chart of Optimization	43
4-4	Arrangement of a Prop	44
4-5	Print-Out of Computer Analysis for Hydraulic Prop	45
4-6	Locations & Dimensions of Bearings & Seals	50
4-7	Weight vs O.D. for All-Hydraulic Props	53
4-8	Weight vs Deflection for All-Hydraulic Props	55
4-9	Hydraulic Design 3 Internal Reservoir	56
4-10	Hydraulic Design 4 External Power Internal Reservoir	58
4-11	Hydraulic Design 5 External Power External Reservoir	59
4-12	Weight vs Pressure for Short Stroke Cylinders	61
4-13	Hydro-Mechanical Design 6 Manual Power Internal Reservoir	63
4-14	Hydro-Mechanical Design 7 External Power Internal Reservoir	64
4-15	Hydro-Mechanical Design 8 External Power External Reservoir	65
4-16	Mechanical-Screw Pin Design 9	82
C-1	Straight & Flexed Shapes of Hydraulic Cylinder	132
F-1	Lateral Loading of Cylinder	178
F-2	Material Thickness for Lateral Strength	181
H-1	Nomenclature & Dimensions	190

LIST OF TABLES

<i>Table</i>		<i>Page</i>
3-1	Weights & Dimensions of Optimal Cylinders	29
3-2	Buckling Loads for Different Materials	30
4-1	Dimensions & Weights of Parts for Alternative Design	34
4-2	Properties of Optimal All-Hydraulic Props	52
4-3	Fiber and Wire Properties [19]	68
4-4	Typical Room-Temperature Major-Axis Properties, Laminated Epoxy Matrix Composites	69
4-5	Selected Physical & Chemical Properties of Metals & Composites	77
4-6	Cost Estimate for Alternate Designs	80

CHAPTER I

INTRODUCTION

Props are used extensively for temporary support of roofs in underground coal mines. Wooden props have been in use for a long time but suffer from a number of disadvantages, chief of them being installation time and lack of yielding capability. Screw jacks and hydraulic supports have found increasing usage over the years but are not yet used to the fullest extent. The weight of current designs is considered to be the primary reason for this situation. Woodruff [1] presents a graph which indicated that a 2.0 metre long hydraulic prop would weigh 54 kgs. Repeated lifting of such a large weight by one person would not be conducive to their safety and well-being and would almost certainly lower their productivity. The U.S. Bureau of Mines initiated a project aimed at reducing the weight of props by applying the latest developments in hydraulic cylinder structural analysis and materials technology. The objectives were spelled out in detail of the design project as follows:

1. Props to be adjustable between 1.829 M (72") and 2.438 m (96") in length.
2. Minimum load capacity of 196 kN (22 tons).
3. Weigh less than 22.3 kg (50 lbs.).
4. Usable on all mining equipment where hydraulic cylinders are required.
5. Bearing surface area at either end 10.32×10^3 sq. mm. (16 in^2).
6. Power source - external/internal.
7. Factor of safety - nominal value of four.
8. Method of operation, materials and fabrication - consistent

with coal mine health and safety regulations.

In addition to hydraulic props, other conceptual designs were to be explored and five alternate designs finalized in sufficient detail to allow manufacturing drawings to be prepared.

Considerations were to be given to the following aspects of manufacture and use of the props.

1. Cost of manufacture and maintenance
2. Ease of handling and usage
3. Robustness
4. Conformance to MSHA regulations
5. Safety in handling, installation and withdrawal
6. Maintainability
7. Operation in a mine environment
8. Life
9. Fluid compatibility and leakage

The effects of varying the factor of safety and the application of unconventional materials were to be included in the investigations.

In addition to reviewing technical literature and manufacturers' brochures on mine roof supports, visits to a working mine and discussions with prop manufacturers were to be conducted, so as to attain familiarity with the state-of-the-art and practical aspects of prop application.

This report is comprised of 5 chapters and 8 appendices. The second chapter reviews current designs while the third presents concepts for alternative designs. Chapter IV presents detailed descriptions of the optimal designs, and is subdivided into two parts, the first dealing

with conventional materials of construction and the second with composites. The final chapter discusses some general aspects of prop usage in relation to the optimal designs and presents the conclusions of the study and recommendations for further work.

Appendix A contains an annotated bibliography and cross-reference of literature pertinent to the project. It should be especially useful for persons attempting to follow up recommendations on the use of composite materials of construction. Appendix B contains details of weight calculations for the various parts in the optimal designs. The appraisal of the structural integrity of the all-hydraulic prop designs was performed using a sophisticated computer program developed at the Fluid Power Research Center for another project [2]. A description of the program and a user's guide are included as Appendices C and D, respectively, while Appendix F contains the FORTRAN listing.

CHAPTER II

REVIEW OF CURRENT DESIGNS

The description given by Woodruff [1] for hydraulic props appears to represent the majority of commercially-marketed props. Included below are brief descriptions of the more commonly available makes, based on manufacturers technical brochures.

Gullick-Dobson [3]

Smallest capacity offered is 25 Tonnes (245 kN) and the largest prop extends from 1.350 m to 2.03 m with extension pieces up to 457 mm long. The hand set props are extended by a pump handle and a setting load of 8 Tonnes (79 kN) is claimed to be achieved without abnormal effort. The prop can be released from a safe distance. Pronged and domed caps and conical bases are available as additional equipment. No mention of weights is given in the Technical Brochures, but it is estimated that they weigh at least 50 kgs. An interesting concept, which will be referred to in a subsequent chapter, is the prop gun set (Figure 2-1) which allows feeding pressurized fluid from a separate hydraulic power pack. This special design is claimed to be capable of handling sudden shock roof convergence. It also has an indicator which gives a visual signal when the setting load exceeds 20 tonnes (198 kN).

Hemscheidt [4]

Single props of light metal and steel are offered in a height range of .635 m to 3.048 m, and loads up to 390 kN. A separate

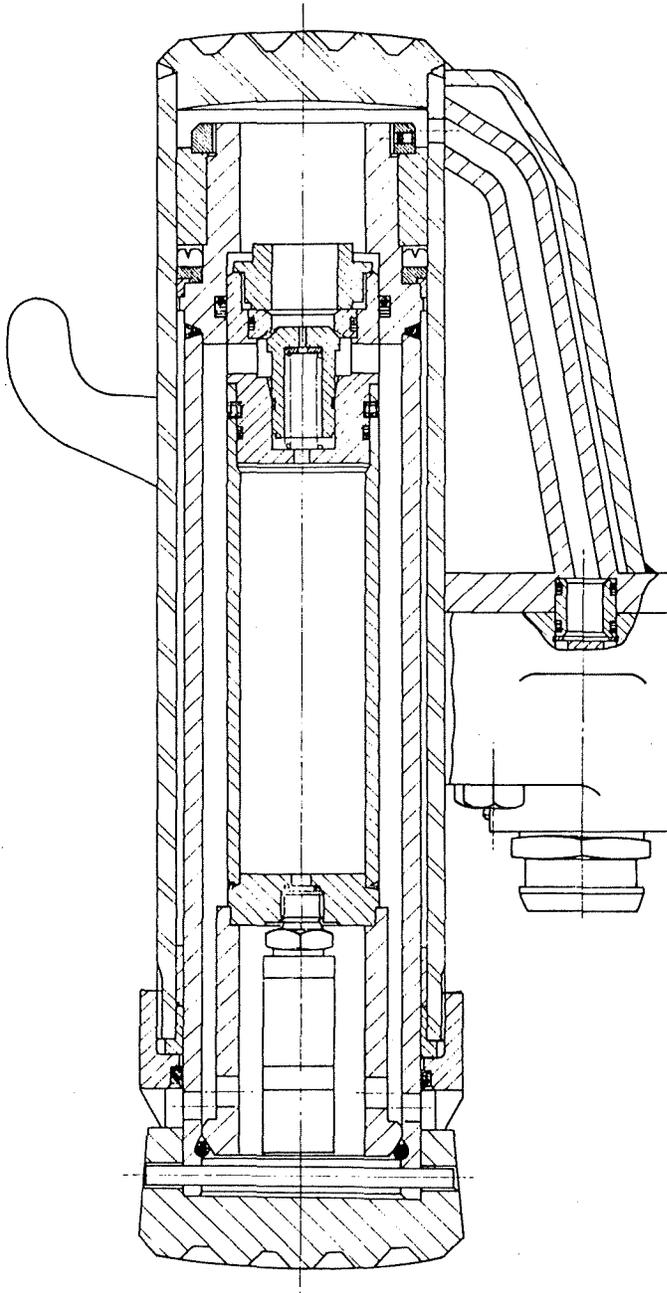


Figure 2-1.

GULLICK-DOBSON 40 Tonne Quick Yield Prop Gun Set [3]

hydraulic power pack is needed to extend the props. From the pictures, it appears that shoes with prongs are provided at either end. Weights are not listed in the brochures furnished, but it is estimated that a 190 kN prop would weigh between 40-50 kg for a 2.5 m extended length. The company also offers a "hydraulic leg" which is essentially a telescopic cylinder. The load capacity of these devices is not mentioned.

Huwood-Irwin [5]

No manual props are offered but the hydraulic cylinders forming part of the longwall and shortwall roof supports are not very different. Mention is made, in an enclosed article, on the use of individual Dowty props, in conjunction with their tandem chocks, for supplementary roof support in the head and tail entries.

Minex [6]

This firm manufactures only self-advancing hydraulic roof supports and crawler sliding hydraulic roof supports.

Marrel-Hydro Mines [7]

This firm manufactures only two and four leg chocks and face supports.

Dowty

The Dowty design is shown in Woodruff [1] and appears to be a very popular one. As discussed by Rao [8], it consists essentially of two steel cylinders, one telescoping inside the other (Figure 2-2).

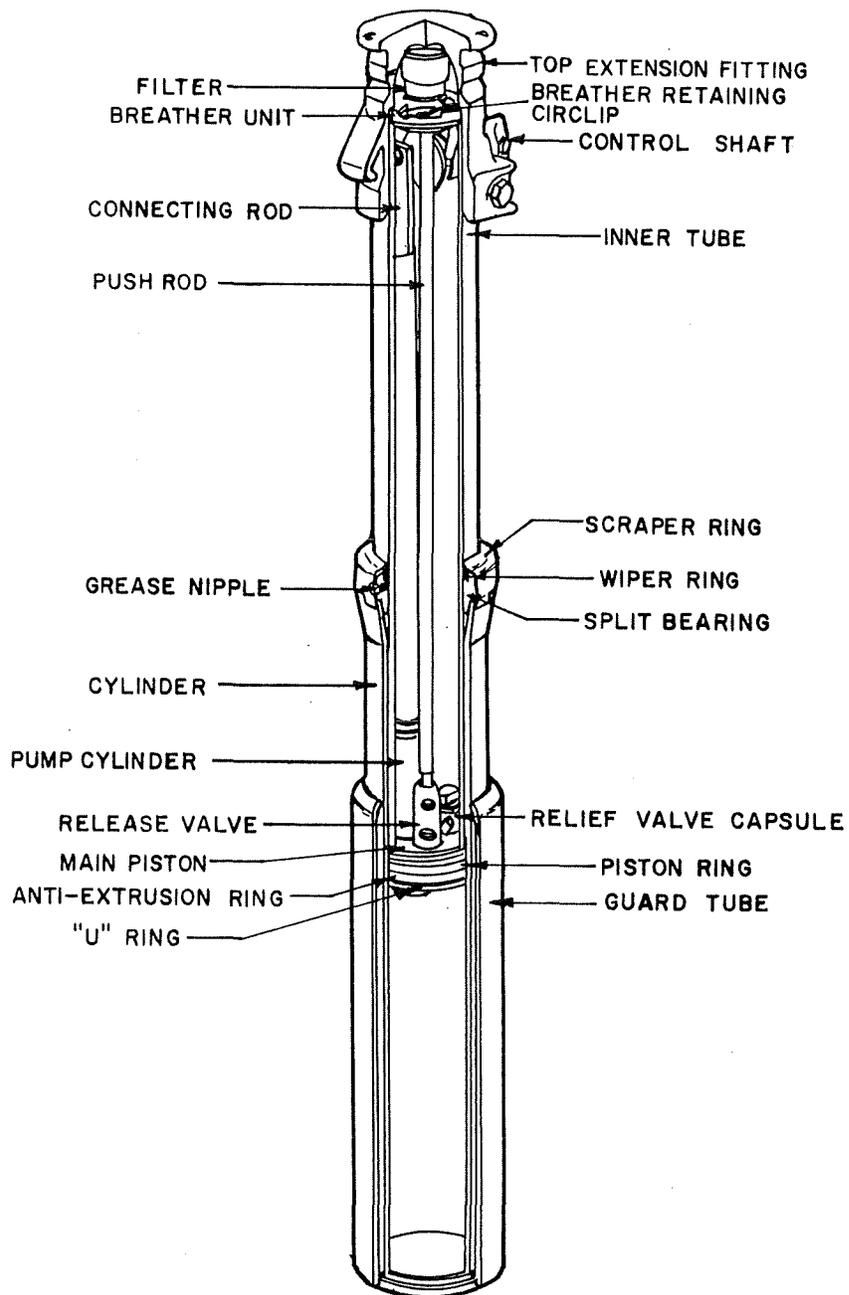


Figure 2-2.

DOWTY HYDRAULIC PROP [8]

The inner one, which has the piston head, also functions as the oil reservoir, and contains all mechanisms for pressurization and release. Figure 2-3 presents some details of the piston head including the non-return valve and sealing arrangement, while Figure 2-4 shows details of the release valve which is located in the piston head. Woodruff [1] mentions a weight range of 22.7 kgs for .6 metres, to 59 kg for 2.13 metre, closed length, for a 20 ton (178 kN) capacity.

Commercial Shearing [9, 10]

This company markets hydraulic props under the trade-name Hy-Jax. They differ from the Dowty design in that the piston, fixed to the lower tube, is stationary, and the moving cylinder incorporates the pump block and reservoir (see Figure 2-5). A bladder in the reservoir isolates the hydraulic fluid from the environment and inhibits entry of dirt and dust into the system. The model 20 Hy-Jax has a capacity of 200 kN and a unit extending from 1.9 metres to 2.51 metres would weigh 46 kgs.

The two reports aimed specifically at designing light-weight hydraulic props are by Anderson [11] and Rao [8]. A comprehensive literature search, including use of a computerized data bank, accessed through the Central Library of Oklahoma State University, failed to reveal technical literature having any direct bearing on the topic.

Anderson [11] developed and tested a light-weight prop made from Al-alloy tubing. It had a load capacity of 65,000 lbs (289 kN), weighed 60 lbs (27.3 kgs) and had an extended length of 1.384 metres. On applying a factor of safety of four, the load capacity of the prop

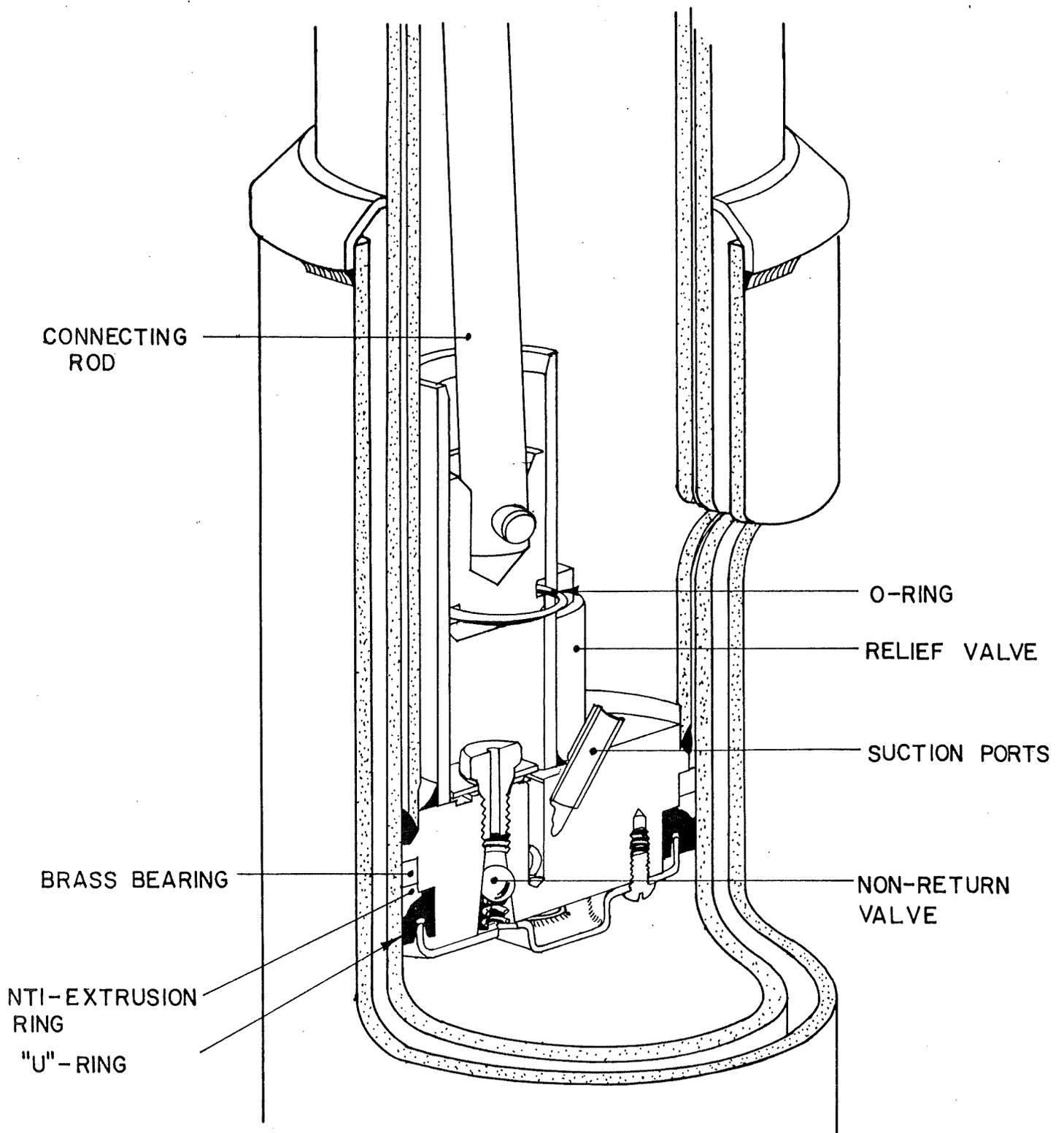


Figure 2-3.

DOWTY MAIN PISTON HEAD [8]

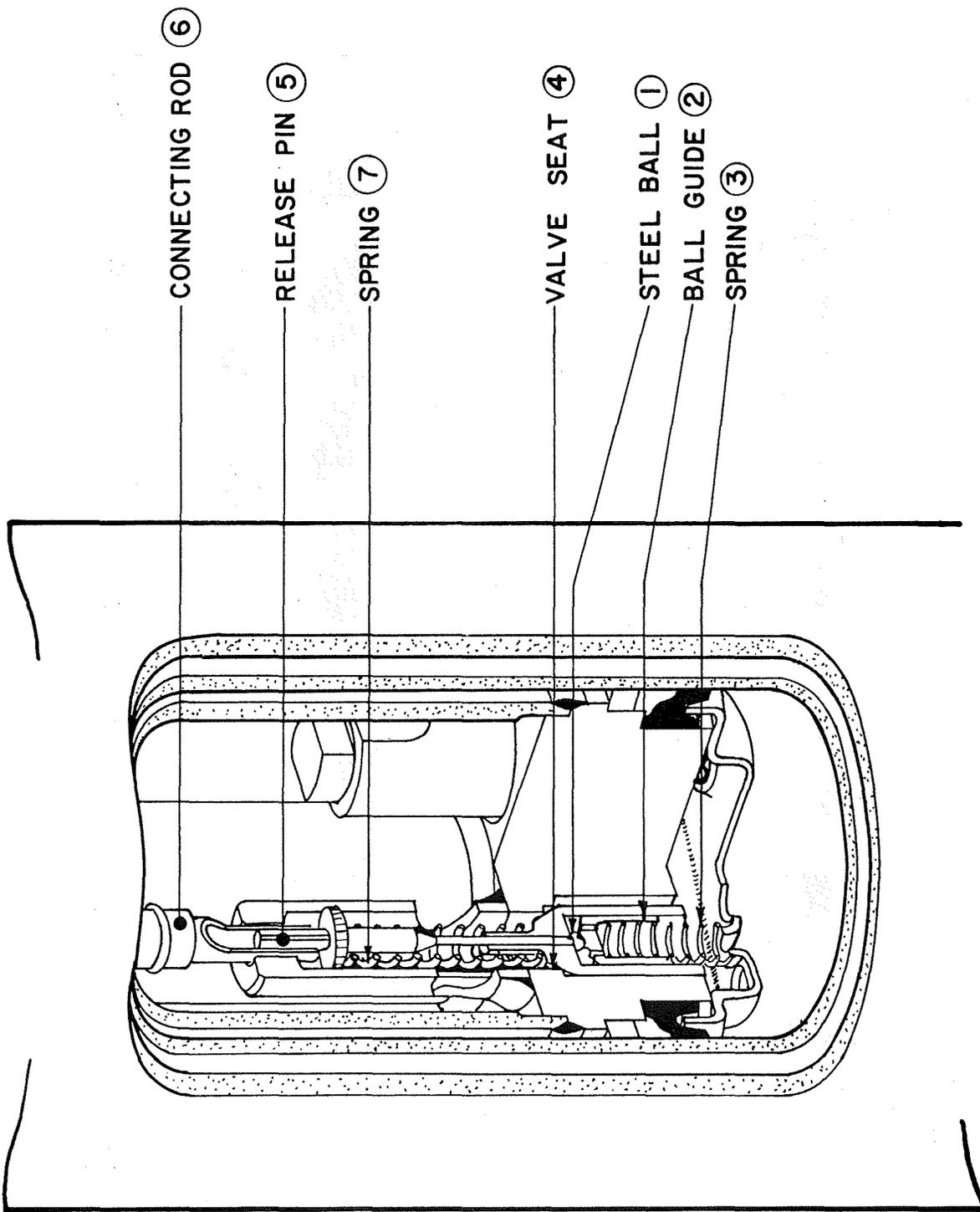


Figure 2-4.

Dowty Design Features [9]

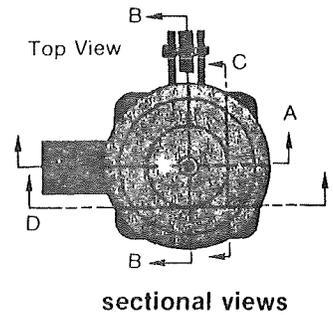
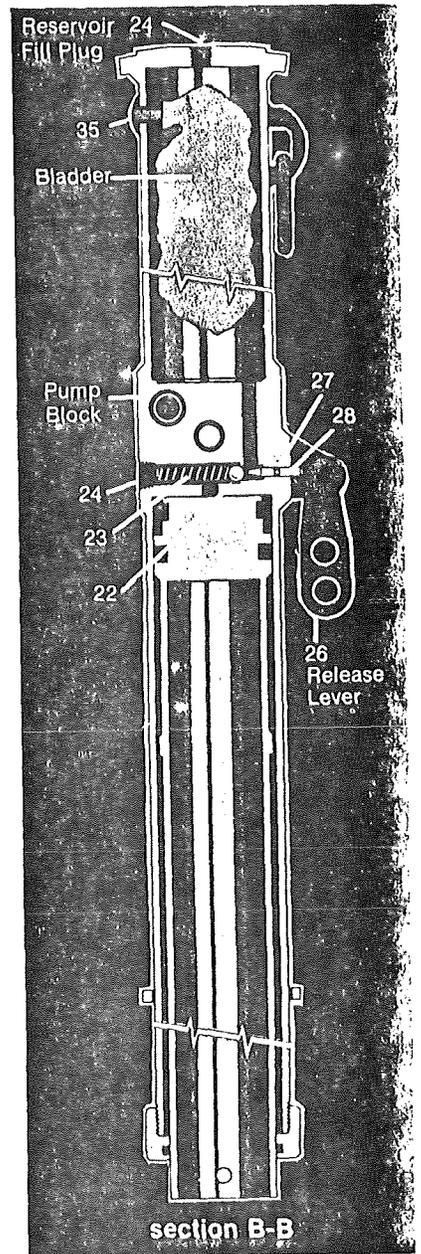
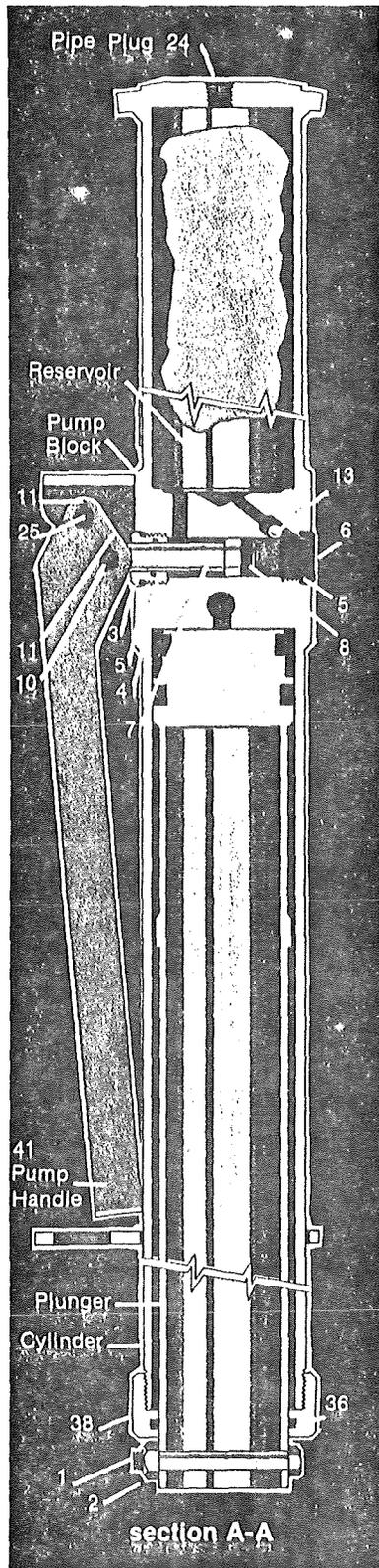


Figure 2-5.

Hy-Jax Design Features [9]

would be 72 kN, i.e., about one-third of the design objective of the current project. According to Anderson, after seven months in a humid atmosphere under loaded conditions, the props were found to be in good condition. Test data furnished by Anderson appear to indicate that failure in the laboratory test specimens was by over-stressing rather than buckling. The test set-up does not appear to have subjected the specimens to any appreciable eccentric loading or end moments.

Rao [8] is probably the first to have systematically evaluated strength vs. weight trade-offs for self-contained props built of conventional material. The design parameters used in the analysis are substantially the same as that specified for the current project. Treatment of the prop as a stepped column, follows historical practice but is technically unsound [12, 13]. No allowances for eccentric loading or end moments are made. Four constructional materials are investigated: SAE 1015 steel, heat-treated Al-alloy 6065-T6, Titanium and Teflon.

After investigating certain common cross-sections for struts, the hollow circular cross-section was found to be the best for flexural rigidity in all directions. Al-alloy was considered the most suitable constructional material after calculating weights and stiffness for a maximum internal fluid pressure of 34.49 MPa. Minimum weights of 52.6 kg for a self-contained prop and 35.7 kg for one fed from a centralized power supply are reported. Since currently available props weigh about 42 kg, Rao's work does not seem to be a significant advance in the design of light-weight props, especially considering the fact that he mistakenly applied a factor of safety of two instead of four.

As part of the effort to become cognizant of practical problems in emplacing roof supports, Dr. G. E. Maroney of the FPRC visited Peabody No. 10 mine, Taylorville, Illinois. He also attended the American Society of Metals Materials Conference and the Materials Engineering Show to identify potentially usable new materials.

CHAPTER III

ALTERNATIVE DESIGNS

Apart from hydraulic props, the only other commonly used, manually emplaceable, roof supports are wooden props and screw jacks. Both suffer from the disadvantages that they are not yieldable, have limited load capacity and can be easily over-stressed in the process of setting up. Even though these two varieties were not considered capable of improvement to meet the design objectives, the option of using mechanical props was not ruled out. Inspection of current designs and prop application practices seemed to indicate that merging the good features of mechanical and hydraulic props could conceivably lead to a new and viable design.

Thus the alternative concepts could be classified in three main categories, i.e., Mechanical, Hydraulic and Hydro-Mechanical. Methods of extension and locking, and location of power units and oil reservoirs would result in alternative sub-designs in each main category. Details of the optimal designs are presented in the next chapter. In the remainder of this chapter some of the basic structural considerations governing the design of different categories of props will be presented.

Mechanical

The mechanical prop can be modeled as a strut of adjustable length. Designing the strut to withstand the rated load when it is fully extended ensures that it will be safe for lesser lengths. The Euler formula (s), applicable to uniform slender columns can be found in any textbook or handbook [14, 15]. For a pin-ended column, under central loading, the buckling load is given by the following equation:

$$F_e = \frac{\pi^2 EA}{(l/k)^2} \quad (3.1)$$

Where: F_e = Euler critical load

E = Young's modulus of material

A = Cross-section area

l = length between pinned ends

k = radius of gyration

A very useful quantity in column analysis is "r," the slenderness ratio defined as l/k .

The Euler formula is reasonably correct for slenderness ratios in excess of 60. For values of r between 30 and 60, the structural member is referred to as an intermediate-length column and a variety of empirical formulas have been devised to guide designers. Other formulas are available for short columns, i.e., r below 30.

At this stage it is important to note that equation 3.1 does not involve the tensile strength of the material. Since most steels have about the same Young's modulus, use of a high tensile strength alloy would not increase the Euler load. A parallel situation exists for Al-alloys. Note also that equation 3.1 does not assist in calculating the deflection of the strut. In fact, according to classical buckling theory, the deflection is zero when the load is below the critical load and infinite above it. Variations of equation 3.1, which allow for different end-fixity, eccentric loading and non-uniformity of the cross-section can be found in the literature. These equations are almost always complex and rarely permit straight-forward calculation of geometrical parameters, such as radii or wall-thickness.

The hollow circular cross-section was established by Rao [8] as the best for withstanding flexure in any plane. It is therefore the logical choice for a mechanical prop, unless non-structural considerations prevail. It can be easily shown that the same circular cross-sectional area, located at a larger mean radius from the center results in a stiffer column, i.e., a higher "k" for the same "A" in equation 3.1. Practical considerations, however, limit the maximum mean radius and consequently the minimum wall thickness.

To get an idea of how large a circular tube would be needed for a steel strut to meet the design load, the Euler load for a pin-ended column can be calculated using:

$$L = 2.438 \text{ m} \quad \text{and} \quad E = 207 \text{ GPa}$$

If we assume a tube of I.D. 101 mm and a wall thickness of 4.76 mm so that

$$D_i = 101 \text{ mm}$$

$$D_o = 110.5 \text{ mm}$$

$$A = \frac{\pi}{4}(D_o^2 - D_i^2) \\ = 956 \text{ sq. mm.}$$

$$k = \sqrt{\frac{D_o^2 + D_i^2}{16}}$$

$$= 37.6 \text{ mm}$$

$$r = 64.85$$

$$\text{Therefore, } F_e = \frac{\pi^2 EA}{(\ell/k)^2} = \frac{\pi^2 \cdot 207 \cdot 10^9 \cdot 956 \cdot 10^{-6}}{64 \cdot 85} \quad (3.2) \\ = 774.6 \text{ kN}$$

With a factor of safety of four, the safe load would be 194 kN, which is very close to the design objective.

Since the slenderness ratio is 64.85 the strut barely falls in the "slender column" category. Any increase in k would lower the slenderness ratio.

The length adjustment on mine roof supports fulfils two needs:

1. Gross changes in roof heights from seam to seam and location to location.
2. Roof and floor irregularities and deformation during setting up.

The adjustment mechanism could be the same for both types of length changes, but it was felt that a rough adjustment section, which would have movements of, say 120 mm, in combination with a fine adjustment section, with a capability of infinite adjustment between zero and 150 mm, would result in the best overall designs.

One of the two promising candidates for rough adjustment was considered to be a sliding tube arrangement--a method which has proved successful in many jacks, retainers and props. The second was the conventional screw. For fine adjustment, the alternatives considered feasible were (a) wedges and (b) screws.

Many alternative methods of locking telescoping tubes, at different lengths, are possible. The shear-pin concept was considered not only the simplest to implement, but also fairly rugged. To withstand a load of 196 kN in double shear, with a safety factor of four, a pin of SAE 1015 (tensile strength 400 MPa) would have to be of diameter D_p , given by

$$D_p = \left[\frac{F \cdot N}{\pi/2 \cdot f_{\text{shear}}} \right]^{1/2} = \left[\frac{196 \cdot 10^3 \cdot 4}{\pi/2 \cdot 400 \cdot 10^6} \right]^{1/2} \quad (3.3)$$

$$= 0.0353 \text{ m, say } 40 \text{ mm.}$$

The wall thickness of the telescopic tubes, where the pin transfers load, would be sized to take the compressive load. Assuming that this load is uniformly distributed over the projected area, and the compressive strength of the tube material is also 400 MPa, the wall thickness is given by

$$t = \frac{F \cdot N}{f_{\text{comp}} \cdot 2\pi D_p} = \frac{196 \cdot 10^3 \cdot 4}{\pi \cdot 0.04 \cdot 2 \cdot 400 \cdot 10^6} \quad (3.4)$$

$$= 0.0078 \text{ m, say } 8 \text{ mm.}$$

A factor of safety of four has been incorporated in the above calculation. It may be noted that this thickness is almost double the 4.75 mm wall thickness used for the Euler load calculations presented earlier. The tubes could be reinforced in the load transfer zone, rather than above the entire length.

Thus, a pair of steel tubes 101 mm I.D./110.5 mm OD by 0.21 metres and 110.5 mm I.D./120.7 mm OD by 1.63 metres could constitute the basic elements of a mechanical prop. It would weigh approximately 40 kg. Note that this is not an optimized "design," since no effort was made to select the "best" tube diameters.

A similar analysis can be performed for Al-alloy construction, as follows:

Construction material is 6061-T6, of tensile strength 207 MPa, and pump modulus 69 GPa.

Assume a tube of I.D. 127 mm and a wall thickness of 8 mm, so that

$$D_i = 127 \text{ mm}$$

$$D_o = 143 \text{ mm}$$

$$A = \frac{\pi}{4} (D_o^2 - D_i^2) \\ = 3390 \text{ sq. mm.}$$

$$k = \frac{D_o^2 + D_i^2}{16}$$

$$= 47.8 \text{ mm}$$

$$r = 2438/47/8 = 51$$

This gives an Euler load, F_e ,

$$F_e = \frac{\pi^2 EA}{(\ell/k)^2} \tag{3.5} \\ = 887.8 \text{ kN}$$

With a safety factor of four, the load capacity would be 221.9 kN which is more than the design objective of 195 kN. Since the slenderness ratio lies between 30 and 60, the Al-alloy prop would be classified as an intermediate length column, and the Euler formula results should be confirmed by more accurate analysis.

A 40 mm diameter steel pin would be satisfactory to transfer the load from one tube to the other provided the safe compressive stress was not exceeded in the area of contact. The wall thickness would be given by:

$$t = \frac{FN}{f_{\text{comp}} 2\pi D_p} = \frac{196 \cdot 10^3 \cdot 4}{207 \cdot 10^6 \cdot 2\pi \cdot 0.04} = 0.015 \text{ m} \tag{3.6}$$

A safety factor of four has been included in the above

calculation. This thickness is more than the tube wall thickness of 8 mm used in the Euler load calculation. Consequently, an Al-alloy prop will need local reinforcement in the load transfer zone.

The pair of Al-alloy tubes 127 mm ID/143 MM OD by 0.91 metres and 143 mm ID/159 mm OD by 1.63 metres, constituting the basic elements of the prop, would weigh approximately 33 kgs. Using Al-alloy has necessitated thicker tubing but the weight is approximately 20% lighter than the steel prop.

Details of the wedge arrangement for fine adjustment are described in the next chapter and corresponding weight calculations presented in Appendix B. These parts of the mechanical telescopic tube prop are less severely stressed than the tubes. The compressive stress would be well within the ultimate stress for Al-alloy or Steel used in the shear pin calculations.

The screw type prop is essentially a screw jack equipped with suitable end plates. For good strength and wear resistance, the screw would be made of steel in preference to Al-alloy. In view of the length of the prop, the screw should be designed to withstand buckling failure.

A hollow screw with a core diameter of 114.3 mm and an ID of 104.8 mm would have, for a column length of 2.438 m, an Euler load given by:

$$F_e = \frac{\pi^2 EA}{(l/k)^2}$$

Where: E = 207 GPa

$$\begin{aligned} A &= \pi/4(D_o^2 - D_i^2) = \pi/4(114.3^2 - 104.8^2) \\ &= 1635 \text{ sq. mm.} \end{aligned}$$

$$k = \frac{D_o^2 + D_i^2}{16} = \frac{114.3^2 + 104.8^2}{16}$$

$$= 38.77 \text{ mm}$$

$$r = \ell/k = 2.438/0.03877 = 63.7$$

$$\text{Therefore, } F_e = \frac{\pi^2 \cdot 207 \cdot 10^9 \cdot 1635 \cdot 10^{-6}}{(63.7)^2} = 824.10^3 \text{ N} \quad (3.7)$$

With a safety factor of four, this results in a safe load of 206 kN which is more than the design objective.

The screw would be left-handed for one-half the length and right-handed for the other half, and thus engaging in two nuts simultaneously. These nuts are to be fixed in tubes of steel, 127 mm ID/133.5 mm OD by suitable lengths, calculated in appendix B. These tubes have a cross-section area and radius of gyration as follows:

$$A = \frac{\pi}{4} (133.5^2 - 127)^2$$

$$= 1330 \text{ sq. mm.}$$

$$k = \frac{133.5^2 + 127^2}{16}$$

$$= 46.06$$

The moment of inertia of the tubes, and the hollow screw is given by the equation:

$$I = Ak^2$$

Inserting the appropriate numerical values, we obtain

$$I_{\text{screw}} = 1635 \times (38.77)^2$$

$$= 2.458 \times 10^6 \text{ mm}^4$$

$$I_{\text{tube}} = 1330 \times (46.06)^2$$

$$= 2.822 \times 10^6 \text{ mm}^4$$

It is concluded that the tubes will be as stiff as the screw and therefore

satisfactory.

An estimation of the weight of the screw-type prop (detail calculations presented in Appendix B) gave a value of 50 kgs, which is seen to be much higher than the values for both steel and Al-alloy telescopic-tube props. It is concluded that the screw-type does not result in as efficient a use of metal as the telescopic type. This is not surprising, considering firstly the fact that only a small fraction of the threads on the screw are transmitting a load at any time, and the rest is "dead weight," and secondly due to space, kinematic and manufacturing constraints, the screw has to be kept reasonably small in diameter and consequently, from a structural point of view, results in poor distribution of available material.

It is felt that the screw is a more convenient arrangement for fine adjustment than the wedges, if only because one person can adjust it by applying a torque to the screw. Since the telescopic-tubes result in a lightweight design, it is logical to attempt to develop a composite design combining the advantages of the two types. In such a design, the telescopic tubes would furnish the rough adjustment, in say, increments of 120 mm, while the screw would provide fine adjustment. Details of the design are presented in the next chapter.

Hydraulic

The all-hydraulic prop is essentially a hydraulic cylinder. If it is self-contained, like the Dowty or commercial design [8, 9], the oil reservoir and the pump are encased in the tubes comprising the prop. The other alternative arrangements are (a) external reservoir, internal

pump, and (b) external reservoir, external pump.

In option (a) above, the prop would be lightened by the amount of reserve oil, but every time it needed to be installed or removed, it would have to be connected to a centralized or mobile reservoir. The pump could be of the manual operation-type or of the rotary variety, drivable by an electric/pneumatic/hydraulic motor. Option (b) would result in the lightest arrangement since the prop would be little more than a hydraulic cylinder. In order to use the prop, it would be hydraulically connected, through say, a quick disconnect coupling, to a mobile/centralized power source.

If the oil reservoir and the pump are disregarded, the three alternative concepts of the all-hydraulic prop call for the same structural requirements, i.e., (i) ability to withstand hydraulic pressure, and (ii) ability to withstand excessive deflection (buckling).

These considerations will be treated separately at first to illustrate the influence of material properties. In the development of the optimal designs, a computer program, which calculates principal stress and deflections, and thus automatically meet both requirements, has been used.

PRESSURIZATION CAPABILITY

The entire prop will be treated as a cylindrical tube of uniform ID and OD, corresponding to those of the outer tube of the actual prop.

Load to be supported = F

Internal fluid pressure = p

Maximum hoop stress (at ID) = σ_i

Cylinder diameters

$$\text{Inner} = D_i$$

$$\text{Outer} = D_o$$

DESIGN REQUIREMENTS

Maximum load = 784 kN

Maximum internal pressure = 69 MPa

Since the load is supported by the internal pressure,

$$F = \frac{\pi}{4} D_i^2 p \quad (3.8)$$

Considering the tube as a thick wall cylinder, the hoop stress is

given by:

$$\sigma_i = \left(\frac{D_o^2 + D_i^2}{D_o^2 - D_i^2} \right) p \quad (3.9)$$

Combining equations 3.8 and 3.9,

$$\sigma_i = \left(\frac{D_o^2 + \frac{4F}{\pi p}}{D_o^2 - \frac{4F}{\pi p}} \right) p \quad (3.10)$$

which yields:

$$D_o^2 = \left(\frac{\sigma_i + p}{\sigma_i - p} \right) \frac{4F}{\pi p} \quad (3.11)$$

and therefore:

$$D_o^2 - D_i^2 = \left(\frac{2p}{\sigma_i - p} \right) \frac{4F}{\pi p} \quad (3.12)$$

The cylinder weight per unit length will be:

$$W_c \triangleq \frac{\pi}{4} (D_o^2 - D_i^2) \rho$$

where ρ is the material density.

Using equation 3.12, the weight per unit length becomes:

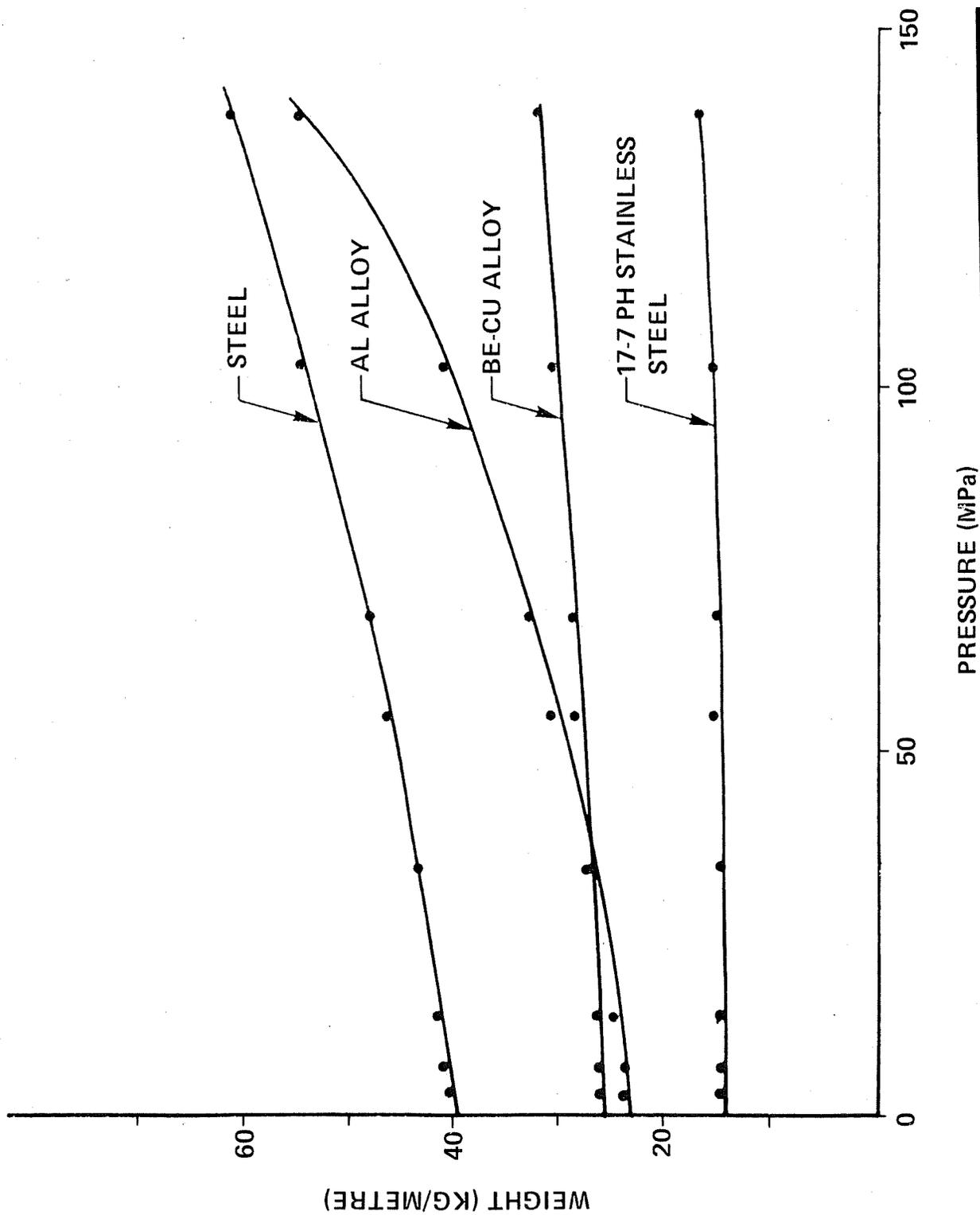
$$W_c = \frac{2F\rho}{\sigma_i - p} \quad (3.14)$$

Plots of W_c versus p are presented in Figure 3-1 for the following materials:

- (a) Steel C1015, with an ultimate tensile strength of 400 MPa and a density of 7709 kg/m³.
- (b) Alloy steel, 17-7 P-H, with an ultimate tensile strength of 1069 MPa and a density of 7518 kg/m³.
- (c) Al-alloy 6061-T6, with an ultimate tensile strength of 240 MPa and a density of 2724 kg/m³.
- (d) Be-Cu alloy, with an ultimate tensile strength of 655 MPa and a density of 8118 kg/m³.

From an inspection of equation 3.14 and Figure 3-1, the following conclusions can be drawn:

1. Cylinder weight is proportional to the load and material density.
2. Higher operating pressures require heavier cylinders.
3. C1015 steel is structurally sub-optimal for all pressures.
4. Below 35 MPa, Al-alloy 6061-T6 is superior to C1015 steel and Be-Cu alloy but not 17-7 P-H stainless steel.
5. Between 35 MPa and 135 MPa, Be-Cu alloy is better than C1015 steel and Al-alloy.
6. Because of its high tensile strength 17-7 P-H stainless steel is excellent for all pressures. The saving in weight due to raising operating pressure is, however, fairly small.



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Figure 3-1.
Cylinder Weight vs. Operating Pressure

Table 3-1 indicates the weights and dimensions of a cylinder 2.438 metres long, made of the materials plotted in Figure 3-1. It may be noted from the above table that low pressure cylinders, though lighter, are voluminous. The difference between the heaviest and lightest cylinders is least for 17-7 P-H alloy steel. It should also be mentioned that the cylinder weights do not include that of internal fluid. Fluid weight would appreciably increase the "wet" weight of the low pressure cylinders.

BUCKLING RESISTANCE

In the above calculations, the cylinder length has entered only in the weight presented in Table 3-1. For short stroke cylinders, say less than 200 mm in the present case, the mode of failure could be expected to be over-pressurization, and Table 3-1 would be sufficient to make a selection. However, for slender cylinders, as in the case of mechanical props, failure by buckling or bending could occur before that due to over-pressurization.

Even though the application of column-buckling formulas for hydraulic cylinders is not, rigorously speaking, correct [12], a good measure of the relative strength of alternate materials can be conveniently obtained by using the Euler formula.

Assuming pinned ends and concentric loading, the safe Euler load

TABLE 3-1

MATERIAL PRESSURE	C1015			17-7 P-H			6061-T6			Be-Cu Alloy		
	OD (mm)	ID (mm)	Wt (kg)	OD (mm)	ID (mm)	Wt (kg)	OD (mm)	ID (mm)	Wt (kg)	OD (mm)	ID (mm)	Wt (kg)
13.79 MPa	278.4	269.0	75.9	272.5	269.0	27.2	284.7	269.0	45.5	274.8	269.0	48.2
34.49 MPa	185.7	170.2	80.36	175.8	170.2	27.7	196.6	170.2	50.5	179.3	170.2	50.0
68.97 MPa	143.0	120.1	88.8	128.3	120.1	28.6	162.3	120.1	60.3	133.6	120.1	52.7

WEIGHTS AND DIMENSIONS OF OPTIMAL CYLINDERS

(Excluding weight of internal fluid)

for a 2.438 metre column is given by

$$F_e = \frac{\pi^2 EA}{(l/k)^2} \quad (3.15)$$

$$\text{Where } k = \sqrt{\frac{D_o^2 + D_i^2}{16}}$$

Since the 68.97 MPa cylinders are the most slender, they are most susceptible to buckling failure. The buckling loads for these cylinders are shown in Table 3-2.

TABLE 3-2
BUCKLING LOADS FOR DIFFERENT MATERIALS

MATERIAL	E (GPa)	l/k	F _e (kN)	Safe load, with a safety factor of four
C1015	206.9	52.2	3546	886.5 kN
17-7 P-H	200.0	55.5	1025	256.3 kN
Al-alloy	68.97	48.5	2635	658.8 kN
Be-Cu alloy	131.0	54.28	1180	295 kN

It is noted firstly that the cylinders fall within the intermediate-length struts (see discussion on mechanical props, in the first section of this chapter) and secondly, the safe load is in excess of the design requirement of 194 kN.

It should once again be emphasized that tensile strength does not enter into the Euler formula, and the difference between the load capacities of the C1015 steel and 17-7 P-H is for the major part due to

the difference in dimensions, and for a minor part, due to the slight difference in Young's modulus.

Modifications of the Euler formula, so as to match predictions with experimental data, have resulted in a number of empirical formulas for intermediate-length columns made of steel [15]. Few of them can be, however, applied with any degree of certitude to other material.

As a result of basic research performed for the Basic Research Program at the Fluid Power Research Center, Oklahoma State University, a new theory of structural analysis of hydraulic cylinders was advanced in 1974 [12]. This theory specifically takes into account the effect of clearances between the cylinder and the piston, establishes the deflection curve for a cylinder subject to arbitrary end moments and eccentric loading, and thence calculates the safe load, so as to limit principal stress in the cylinder and rod to prescribed values. A brief description of the computer program, SACREG, is included in Appendix C of this report.

CHAPTER IV

OPTIMAL DESIGN

Chapter III has presented the three categories of prop designs considered suitable for development and evaluation, viz, mechanical, hydraulic and hydro-mechanical. Brief descriptions of the final designs will be presented here. Dimensions and weights of parts are included in Appendix B. Since the main objective of the project was to reduce the weight of props, the comparison of alternate designs is best depicted in a table, which shows the break-up of weights of different parts. Table IV.1 presents such a break-up for optimal designs and will be continually referred to in the course of this discussion. An explanation of the column heading is presented below to assist in comprehension.

COLUMN 1

Even though thirteen individual designs are presented in the table, they fall into the three categories described earlier. One factor, which was not introduced in Chapter III, but which results in radical changes in manufacturing technology is the material of the props. Those built of conventional isotropic materials, e.g., metals, are distinguished in this aspect from those built of non-isotropic materials, i.e., composites.

COLUMN 2

Mechanical, hydraulic and hydro-mechanical props designed with a factor of safety of four are followed by designs developed with a safety factor of two.

Table 4-1. Dimensions and Weights of Parts for Alternative Designs.

S #	Design Type	Top End	Bottom End	Rough Adjust. Section	Final Adjust. Section	Overload Control	Outer Diameter	Wall Thickness	Pump	Fluid	Factor of Safety	Total Wt (kg)	Total Wt (lbs)	Remarks
		(kg)	(kg)	(kg)	(kg)	(kg)	(mm)	(mm)	(kg)	(kg)		(kg)	(lbs)	
1	Mechanical Wedge (Al-Alloy)	0.1	0.15	18.76	0.22	21.92 4.16	151	8	—	—	4	41.15* 19.23† 23.49	92.18 43.07 52.6	Maximum Internal Pressure: 40 MPa (5800 psi), Maximum Deflection: 25 mm (1")
2	Mechanical Screw (Steel)	0.34	0.48	—	43.75	21.92	—	—	—	—	4	66.49* 44.57	148.94 99.84	
3	Hydraulic Int. Power Int. Reservoir (Al-Alloy)	0.1	0.17	—	27.76	0.25	145	10.6	0.34	8.27	4	36.55	81.87	
4	Hydraulic Ext. Power Int. Reservoir (Al-Alloy)	0.1	0.17	—	27.76	0.25	145	10.6	0.21	8.27	4	36.42	81.58	
5	Hydraulic Ext. Power Ext. Reservoir (Al-Alloy)	0.1	0.17	—	26.76	0.25	145	10.6	0.21	—	4	27.49	61.58	
6	Hydro-Mech Int. Power Int. Reservoir (Al-Alloy)	0.17	0.19	19.53	2.12	0.25	168.6	13.6	0.34	2.6	4	25.2	56.45	
7	Hydro-Mech Ext. Power Int. Reservoir (Al-Alloy)	0.17	0.19	19.53	2.12	0.25	168.6	13.6	0.21	2.6	4	25.07	56.16	

34

Table 4-1. Dimensions and Weights of Parts for Alternative Designs (Cont.).

S #	Design Type	Top End	Bottom End	Rough Adjust. Section	Final Adjust. Section	Overload Control	Outer Diameter	Wall Thickness	Pump	Fluid	Factor of Safety	Total Wt (kg)	Total Wt (lbs)	Remarks
8	Hydro-Mech Ext. Power Ext. Reservoir (Al-Alloy)	(kg)	(kg)	(kg)	(kg)	(kg)	(mm)	(mm)	(kg)	(kg)		(kg)	(lbs)	Maximum Internal Pressure: 40 MPa (5800 psi), Maximum Deflection: 25 mm (1")
		0.17	0.19	19.53	1.72	0.25	168.6	13.6	0.21	—	4	22.07	49.43	
9	Hydraulic Int. Power Int. Reservoir (Aramid-Graphite)	0.09	0.11	—	14.46	0.25	110	12.5	0.34	5.12	4	20.37	45.63	
10	Hydraulic Int. Power Int. Reservoir (Steel)	0.17	0.17	—	17.41	0.25	162	3.3	0.34	13.59	2	30.3	67.87	
11	Hydraulic Int. Power Int. Reservoir (Al-Alloy)	0.15	0.17	—	15.78	0.25	160	4.25	0.34	12.94	2	29.63	66.37	
12	Hydraulic Int. Power Int. Reservoir (Aramid-Graphite)	0.04	0.08	—	8.81	0.25	130	4	0.34	8.03	2	17.55	39.31	
13	Hydro-Mech Int. Power Int. Reservoir (Al-Alloy)	0.16	0.19	14.24	1.34	0.25	168	6	0.34	2.82	2	19.34	43.32	

* With Overload Control, 33 Belleville Springs, OD = 127 mm, ID = 76.2 mm, thickness is 10.7 mm.

† With Liquid Spring

COLUMN 3 AND 4

All props are designed to have a bearing area of at least 0.01 m². The top and bottom ends are the end caps needed to provide this bearing area.

COLUMN 5

As explained in Chapter III, the purpose of the rough adjustment section is to allow the prop height to be changed by 0.6 m in discrete steps. Only the mechanical wedge (telescopic tube) and hydro-mechanical designs utilize rough adjustment.

COLUMN 6

Final adjustment may be in the form of wedges or short-stroke hydraulic cylinders. In the case of some designs, the same mechanism allows fine, as well as coarse, adjustment.

COLUMN 7

Overload control is provided to make the props yieldable. In the case of the mechanical props, the mechanism may be Belleville springs or a liquid spring. For the hydraulic and hydro-mechanical props, it is the relief valve.

COLUMN 8 AND 9

The minimum outer diameter and wall thickness are indicated here. Detail dimensions are presented in Appendix B. All dimensions are in mm.

COLUMN 10

Only hydraulic and hydro-mechanical props need a pump, whose weight is included in the case of self-contained designs.

COLUMN 11

All hydraulic and hydro-mechanical props contain a liquid column, whose weight is significant. The self-contained props also have an internal fluid reservoir whose gross weight is also significant.

COLUMN 12

The factor of safety shown is that applied to the Euler load in the case of mechanical and hydro-mechanical props, and the "factor of safety on stress" in the computer program SACREG, in the case of hydraulic props.

COLUMN 13 AND 14

Weight of the props, shown in kgs and lbs, is for each design, the total weight of components, including the fluid, if any. This will be referred to sometimes as "wet" weight to signify that it includes the weight of any internal fluid.

COLUMN 15

Constraints on maximum internal pressure refer to hydraulic and hydro-mechanical props. Pressures higher than the value shown can be used in the short-stroke fine adjustment cylinders used in the hydro-mechanical designs. Deflection constraints are dictated more by aesthetic

and psychological reasons than structural reasons.

MECHANICAL

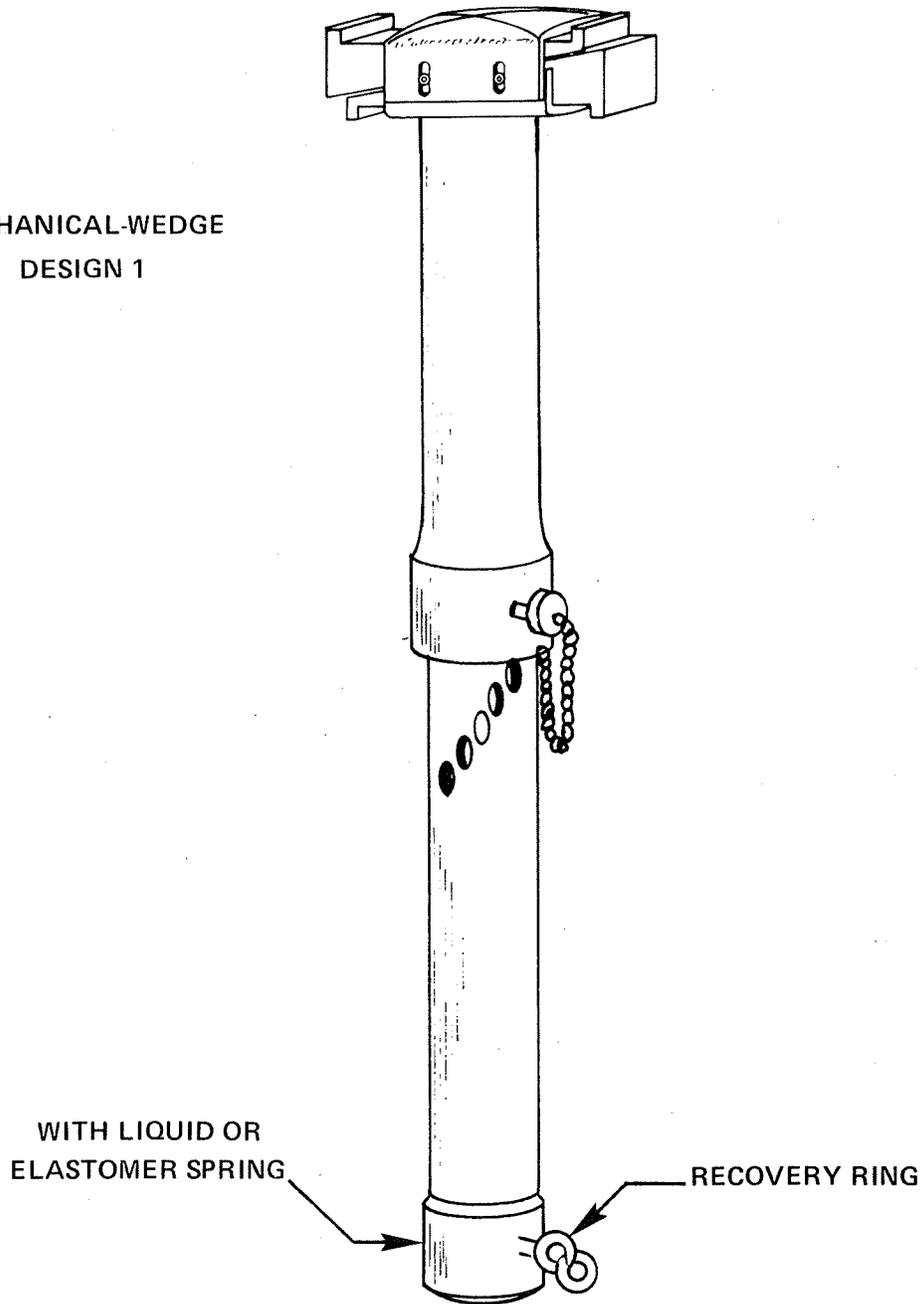
#1 Wedge Type

Aluminum-alloy 6061-T6 was chosen as the material of construction of the telescopic tubes. Formulas for sizing have already been presented in Chapter III. Treating the entire prop as a uniform slender prop results in a conservative design. Figure 4-1 presents a sketch of the conceptual design, and Appendix B contains dimensions and weight calculations. The shear pin has been shown as the load transfer element, but could be replaced by any other device, e.g., snap rings, spring-loaded pins, etc., which would provide the adjustability and be easy to use. As shown in Appendix A, the use of a coil spring for overload protection was discarded due to excessive weight and size, and Belleville springs were found to be compact but heavy. It is seen from Table 4-1 that the springs constitute 53% by weight of the entire prop. If a liquid spring were used, the prop would weigh 21.25 kg. The main disadvantage of the mechanical wedge type is the difficulty of setting up by a single person. Another disadvantage is the inability to release the prop from afar. Dragging the prop by the recovery ring, while under load, could conceivably bend the tubes and permanently reduce their load bearing capacity. The dishing of the top and bottom ends is expected to alleviate this situation.

#2 Screw Type

Leading dimensions of the screw type mechanical prop, shown in Figure 4-2

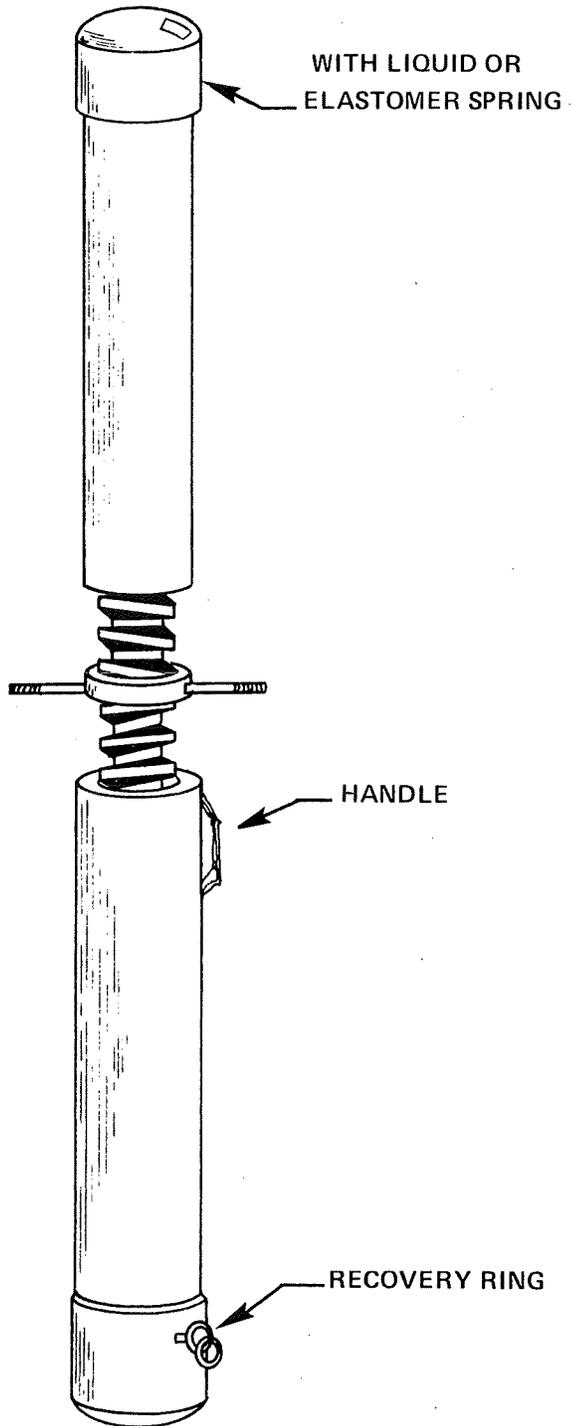
MECHANICAL-WEDGE
DESIGN 1



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FIGURE 4-1
Mechanical-Wedge Design 1

MECHANICAL-SCREW
DESIGN 2



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FIGURE 4-2.
Mechanical-Screw Design 2

have been calculated in Chapter III, and weight calculations presented in Appendix B. A double screw design has been adopted so as to be convenient for the user. The Belleville spring arrangement could be used for overload control but, as in the case of the wedge design, would be heavier than the liquid spring design. This design is heavy, mostly because of its steel screw. It has the advantage of possessing only one mechanism of adjustment. Like the wedge design, this design also does not permit load release from a distance.

HYDRAULIC

As explained in Chapter III, the all-hydraulic prop is essentially a hydraulic cylinder with overload protection in the form of a relief valve. The rough and final adjustments are provided by the same mechanism, i.e., the sliding of the piston in the cylinder bore. Chapter III has already presented the formulas for sizing a cylinder to withstand internal pressure and buckling. One method of optimization is to use the Euler formula to express the cylinder outer diameter in terms of the inner diameter and minimize the weight of the "wet" prop (i.e., after incorporating the weight of the fluid). The dimensions of the cylinder so obtained can then be checked for internal pressurization. The results of such analysis, presented in Appendix H for steel and Al-alloy, indicate that in the former case, internal pressurization is a limiting factor while in the latter, buckling failure limits the maximum load. Optimal props, using the Euler formula, weigh 49 kg for steel and 47.7 kg for Al-alloy. Since the steel prop is limited by internal pressurization to 442 kN, while the Al-alloy can take 783 kN, the latter is clearly superior. The steel prop could, of course, be improved by using stronger material.

Appendix H also shows that, using steel, the prop cannot be made equally strong against buckling and internal pressurization. On the other hand, the Al-alloy prop can be so designed, leading to the conclusion that the minimal weight Al-alloy prop makes better use of its tensile strength and stiffness than steel. It should be stressed that the above results were obtained by assuming that the prop functions as a slender uniform column. Any deviations from this ideal configuration would affect the numbers, but it is felt, that the general conclusions are valid for the size of prop under consideration.

As explained in Chapter III, a computer program to ascertain the load capacity of a hydraulic cylinder subject to arbitrary end moments and eccentric loading was developed at the Fluid Power Research Center, Oklahoma State University [16]. The process of optimizing the design of all hydraulic props, using this computer program, SACREG, consisted of iteratively ascertaining the load capacity, deflection and stresses of a given design and altering the dimensions and material properties so as to achieve the lightest prop. Figure 4-3 presents a flowchart of this process.

Figure 4-4 is a sketch of the hydraulic prop, modeled as a hydraulic cylinder, with salient dimensions marked thereon. (Appendix D lists all the dimensions and material properties needed for computer analysis.) Note that some dimensions, e.g., end pin diameters, pin distance from end plate and stop tube length, are not pertinent to the prop, but need to be given numerical values so as to be acceptable to the computer program. Referring to Figure 4-5, which is a facsimile of the computer print-out, the cylinder and rod lengths and the extended length in

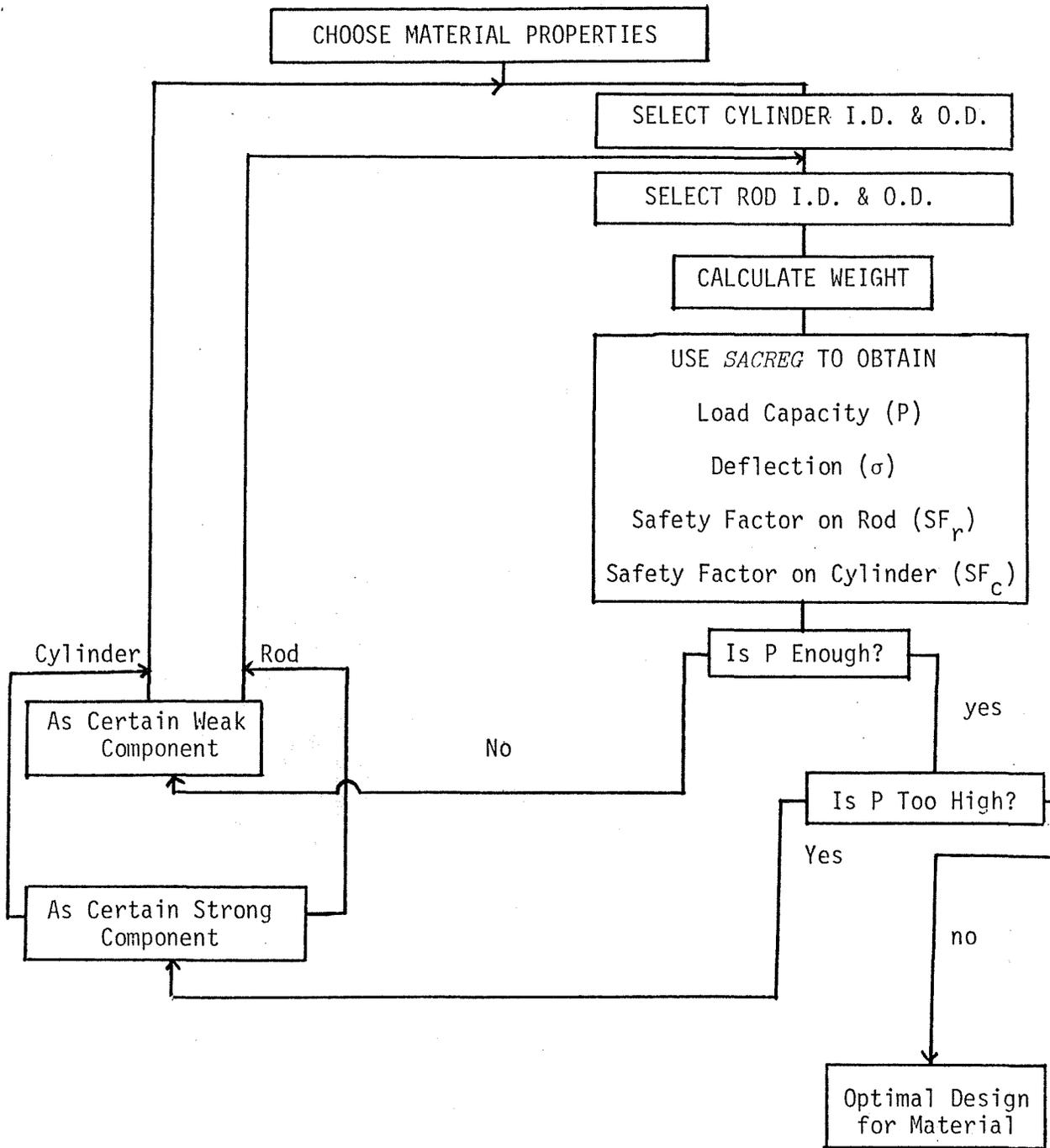
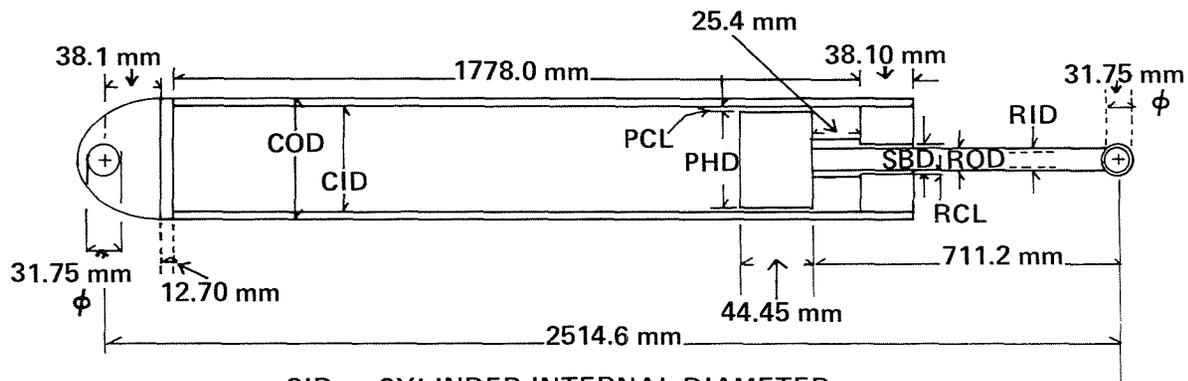


FIGURE 4-3
FLOW CHART OF OPTIMIZATION



- CID CYLINDER INTERNAL DIAMETER
- COD CYLINDER OUTER DIAMETER
- PHD PISTON HEAD DIAMETER
- ROD ROD OUTER DIAMETER
- RID ROD INNER DIAMETER
- SBD STUFFING BOX DIAMETER
- PCL PISTON CLEARANCE
- RCL ROD CLEARANCE

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FIGURE 4-4
Arrangement of a Prop for Computer Analysis

PROGRAM SACREG - STRESS ANALYSIS OF CYLINDERS (REGULAR)

USBM CYLINDER EVALUATION-METRIC 780119.05

PROBLEM 5

AL-ALLOY CYLINDER EVALUATION; EST. WT. - DRY 46.41 KG - WET 52.05

INPUT DATA:

TABLE 1: CONTROL DATA

PROBLEM TYPE = 1 - CRITICAL LOAD ANALYSIS & ANALYSIS FOR A FACTORED LOAD

TABLES RETAINED FROM PREVIOUS PROBLEM

2 3 4 5 6 7

NO KEEP OPTIONS EXERCISED

TABLE 2: UNITS OF MEASUREMENT

LENGTH	LOAD	PRESSURE	ANGULAR
MET	NTN	PA	DEG

TABLE 3: CYLINDER DIMENSIONS

LENGTHS:

STROKE	PISTON HEAD	STUFFING BOX	END PLATE	HINGE DIST.
6.09600D-01	4.44500D-02	3.81000D-02	1.02870D+00	3.81000D-02
CYLINDER	ROD	EXTENDED	STOP TUBE	
7.04850D-01	6.73100D-01	2.43840D+00	5.08000D-02	

DIAMETERS:

CYL. OUTER	CYL. INNER	ROD OUTER	ROD INNER	
1.67640D-01	1.49860D-01	1.38430D-01	1.20650D-01	HOLLOW ROD WITH NO FLUID
CYL. PIN *	ROD PIN *	PISTON HEAD @	STUF. BOX @	
3.17500D-02	3.17500D-02	0.0	0.0	
(* ZERO, THE END IS FIXED) (@ ZERO, OTHER OPTION IS INPUT)				

CLEARANCES BETWEEN:

CYLINDER AND STUFFING BOX	CYLINDER AND PISTON HEAD @	ROD AND STUF. BOX @
0.0	5.08000D-05	5.08000D-05
(@ ZERO, OTHER OPTION IS INPUT)		

TABLE 4: BEARINGS AND SEALS

PISTON BEARINGS:

FIGURE 4-5

PRINT-OUT OF COMPUTER ANALYSIS FOR HYDRAULIC PROP

A	A	A	B	DISTANCE FROM
WIDTH	THICKNESS	YOUNGS MODULUS	STIFFNESS	BACK FACE
2.54000D-03	2.54000D-03	4.82630D+09	0.0	1.90500D-02
2.54000D-03	2.54000D-03	4.82630D+09	0.0	2.41300D-02

ROD BEARINGS:

A	A	A	B	DISTANCE FROM
WIDTH	THICKNESS	YOUNGS MODULUS	STIFFNESS	BACK FACE
1.27000D-02	2.54000D-03	1.03420D+10	0.0	1.90500D-02

(A IS USED TO CALCULATE B - HENCE, EITHER A OR B IS INPUT
ZERO'S ABOVE INDICATE THAT THEY ARE NOT INPUT)

TABLE 5: WEIGHTS AND MATERIAL PROPERTIES

WEIGHTS OF PARTS:

CYLINDER (PER UNIT LENGTH)	ROD	PISTON HEAD	STUFFING BOX
1.20310D+02	9.82500D+01	6.66995D+00	6.66995D+00

MATERIAL PROPERTIES:

YOUNGS MODULUS	YIELD STRESS		
CYLINDER	ROD	CYLINDER	ROD
6.89470D+10	6.89470D+10	4.96420D+08	4.96420D+08

TABLE 6: INCLINATION, FIXITY, FRICTION COEFFICIENTS, LOADING ECCENTRICITY

CYL INCLINATION WITH HORIZONTAL = 9.00000D+01

	CYLINDER END	ROD END
SUPPORT CONDITIONS:	PIN	PIN
FRICTION COEFFICIENTS AT SUPPORTS: (ZERO IF FIXED END)	1.00000D-03	1.00000D-03
LOADING ECCENTRICITIES:	1.27000D-02	1.27000D-02

TABLE 7: FACTOR OF SAFETY OR OPERATING PRESSURE AND/OR ALLOWABLE THETA & F
DEPENDING ON PROBLEM TYPE

FACTOR OF SAFETY = 4.000 ON STRS

PROGRAM SACREG - STRESS ANALYSIS OF CYLINDERS (REGULAR)

USBM CYLINDER EVALUATION-METRIC 780119.05

PROBLEM 5

AL-ALLOY CYLINDER EVALUATION; EST. WT. - DRY 46.41 KG - WET 52.05

RESULTS: CRITICAL LOAD ANALYSIS

CRITICAL LOAD = 7.478D+05 NTN

FIGURE 4-5

PRINT-OUT OF COMPUTER ANALYSIS FOR HYDRAULIC PROP (cont.)

MAXIMUM FLUID PRESSURE = 4.240D+07 PA
CROOKEDNESS ANGLE = 3.83559D-02 DEG

CYLINDER:

MAXIMUM DEFLECTION = 1.644D-02 MET
MAXIMUM LONGITUDINAL STRESS= 1.303D+08 PA
AT A DISTANCE FROM CYL SUP = 1.262D+00 MET
FACTOR OF SAFETY ON CYL = 3.810D+00

MAX SHEAR STRESS IN CYL = 2.481D+08 PA
AT MAX LONG STRESS POINT
AND AT INNER SURFACE
FACTOR OF SAFETY ON CYL = 1.000D+00

MAXIMUM HOOP STRESS IN CYL = 3.797D+08 PA
FACTOR OF SAFETY ON CYL = 1.307D+00

ROD :

MAXIMUM DEFLECTION = 1.472D-02 MET
MAXIMUM LONGITUDINAL STRESS= 3.927D+08 PA
AT A DISTANCE FROM CYL SUP = 1.752D+00 MET
FACTOR OF SAFETY ON ROD = 1.264D+00

FORCES AT SLIDING CONNECTION:

PISTON BEARINGS (SEALS) NO	FORCE
1	2.444D+04 NTN
2	2.690D+04 NTN

ROD BEARINGS (SEALS) NO	FORCE
1	1.841D+05 NTN

F1- FORCE AT PISTON HEAD FRONT FACE = 1.327D+05 NTN
F2- FORCE AT STUFFING BOX FRONT FACE = 0.0 NTN
F3- FORCE AT PISTON HEAD BACK FACE = 0.0 NTN
F4- FORCE AT STUFFING BOX INNER FACE = 0.0 NTN
(ZERO FORCES INDICATE NO CONTACT)

FIGURE 4-5

PRINT-OUT OF COMPUTER ANALYSIS FOR HYDRAULIC PROP (cont.)

ANALYSIS AFTER APPLYING GIVEN FACTOR OF SAFETY OF 4.000 ON STRS:

LOAD = 2.096D+05 NTN
 FLUID PRESSURE = 1.189D+07 PA
 CROOKEDNESS ANGLE = 1.78586D-02 DEG

CYLINDER:

MAXIMUM DEFLECTION = 2.683D-03 MET
 MAXIMUM LONGITUDINAL STRESS= 1.927D+07 PA
 AT A DISTANCE FROM CYL SUP = 1.278D+00 MET
 FACTOR OF SAFETY ON CYL = 2.576D+01

 MAX SHEAR STRESS IN CYL = 6.184D+07 PA
 AT MAX LONG STRESS POINT
 AND AT INNER SURFACE
 FACTOR OF SAFETY ON CYL = 4.014D+00

 MAXIMUM HOOP STRESS IN CYL = 1.065D+08 PA
 FACTOR OF SAFETY ON CYL = 4.663D+00

ROD :

MAXIMUM DEFLECTION = 1.604D-03 MET
 MAXIMUM LONGITUDINAL STRESS= 8.514D+07 PA
 AT A DISTANCE FROM CYL SUP = 1.775D+00 MET
 FACTOR OF SAFETY ON ROD = 5.831D+00

FORCES AT SLIDING CONNECTION:

PISTON BEARINGS (SEALS) NO	FORCE	
1	1.658D+04	NTN
2	1.772D+04	NTN

ROD BEARINGS (SEALS) NO	FORCE	
1	3.430D+04	NTN

F1- FORCE AT PISTON HEAD FRONT FACE = 0.0 NTN
 F2- FORCE AT STUFFING BOX FRONT FACE = 0.0 NTN
 F3- FORCE AT PISTON HEAD BACK FACE = 0.0 NTN
 F4- FORCE AT STUFFING BOX INNER FACE = 0.0 NTN
 (ZERO FORCES INDICATE NO CONTACT)

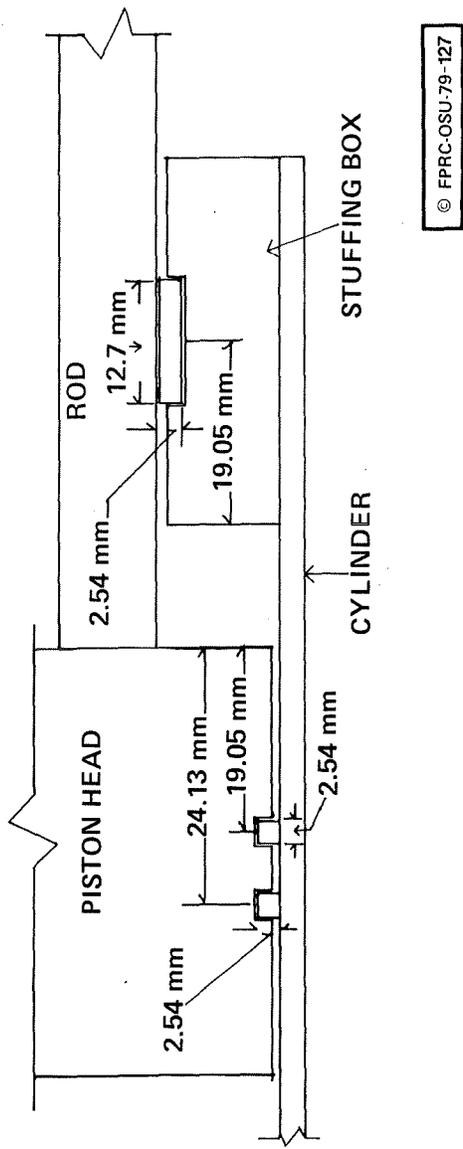
FIGURE 4-5

PRINT-OUT OF COMPUTER ANALYSIS FOR HYDRAULIC PROP (cont.)

Table 3 are the critical longitudinal dimensions. Other dimensions may be altered, if necessary, to obtain the correct overall lengths. Note that the cylinder, whose analysis is presented in Figure 4-5, has an extended length of 2.438 metres and a stroke of 0.6096 metres, which are the design requirements for the prop. The cylinder and rod diameters and the material properties were the only quantities treated as parameters in the optimization process. Numerical values of all other quantities, e.g., pin clearances, piston/cylinder/stuffing box clearances were assumed to reflect good design practice. Figure 4-6 presents the dimensions and locations of the bearings and seals. The clearances at the piston head/cylinder and piston rod/stuffing box interfaces establish the "crookedness angle," which in turn influences the capacity of the prop.

The capacity of a prop is influenced by the manner in which it is set up. If the ground/roof is uneven, the prop could be installed obliquely. In general, the end-forces on the prop will not be concentric with the center-line of the cylinder-rod, but it is extremely difficult to estimate either the maximum eccentricity or the end-fixity. Values of 12.7 mm and 0.001 have been assigned to the eccentricity and pin-friction coefficient (Table 6 in Figure 4-5.)

If the actual eccentricity was larger than this design value, the capacity of the prop would be reduced. The pin-friction coefficient is a very low value, so that the cylinder is essentially pin-ended. Larger values of the friction coefficient would result in a larger load capacity. A factor of safety of four was used for most analysis. Optimal designs, using a factor of safety of two, were also developed, to ascertain the savings in weight.



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FIGURE 4-6

LOCATIONS AND DIMENSIONS OF BEARINGS AND SEALS

The latter part of the computer output consists of two identical sections, each detailing the stresses and deflections in the cylinder and rod; the first being for analysis without a factor of safety, and the second with the specified factor of safety. Since the equations for the deflections are non-linear, the application of the factor of safety on the stress results in a load capacity different from that obtained by application of the factor to the load. In all analysis reported here, the safety factor has been applied on the stress.

Table 4-2 summarizes the results of optimization using SACREG iteratively, as shown in Figure 4-3. Each line in the table corresponds to an optimal design, in the sense that the cylinder and rod are equally strong. An upper limit of 175 mm was imposed on the cylinder OD to ensure firstly that props would not be unwieldy and secondly, that the wall thickness would not be too small (see Appendix F for lateral load resistance calculations). A lower limit of 105 mm was imposed primarily to ensure that the rod was strong enough to carry the desired load without exceeding 40 MPa internal pressure.

Figure 4-7 depicts the weight vs. cylinder OD relationship graphically for the three materials evaluated. It is seen that 145 mm is the optimal OD for both steel and Al-alloy props, and the latter is approximately 20% lighter.

Other important conclusions highlighted in Table 4-2 are

1. Reduction of eccentricity from 12.7 mm to 2.5 mm, increases the load capacity by 12%. Every effort should be made to install the prop so that it is centrally loaded.

Table 4-2. Properties of Optimal All-Hydraulic Props

S#	Material	Cylinder		Rod		Wet Weight kg	Yield Stress MPa	Young's Mod GPa	Defl. mm	Load kN	Pressure MPa	Remarks
		OD mm	ID mm	OD mm	ID mm							
1	Steel	175	169	148	143.5	46.82	1172	206.8	1.75	197	8.8	
2	Steel	160	153	139.7	133.3	47.08	1172	206.8	2.21	200	10.9	
3	Steel	160	153	139.7	133.3	47.08	1172	206.8	4.9	224	12.2	Low Eccentricity(2.5 mm)
3a	Steel	145	137.3	123.0	117.0	42.77	1172	206.8	3.0	196	13.2	
4	Steel	127	114.3	92.7	78.7	54.78	1172	206.8	4.1	255	24.9	
5	Steel	122.5	114.3	92.7	78.7	46.36	1172	206.8	3.9	205	19.9	
6	Steel	162	158.7	140	137	30.96	1172	296.8	4.7	196	9.9	Factor of Safety = 2.0
7	Aluminum Alloy	180	165.3	150	134	44.23	496	68.9	2.2	196	9.1	
8	Aluminum Alloy	167.2	149.9	138.4	120.7	42.92	496	68.9	2.7	210	11.9	
9	Aluminum Alloy	160.0	141.0	123.0	105.0	40.87	496	68.9	3.2	207	13.3	
9b	Aluminum Alloy	145.0	123.8	112.0	94.0	36.03	496	68.9	4.0	196	16.3	
10	Aluminum Alloy	128.0	100.5	88.9	0.0	43.28	496	68.9	5.3	193	24.3	
11	Aluminum Alloy	127.5	100.0	86.0	25.4	39.59	496	68.9	5.7	191	24.3	
12	Aluminum Alloy	160	151.5	125	111.5	28.72	496	68.9	6.3	199	11.0	Factor of Safety = 2.0
13	Aramid-Graphite	160	149	128	118	23.64	772	71.7	4.9	198	11.4	
14	Aramid-Graphite	130	114	90	60	21.0	772	71.7	8.1	201	19.7	
15	Aramid-Graphite	130	122	90	77	16.84	772	71.7	21.2	192	11.6	Factor of Safety = 2.0
16	Aramid-Graphite	116	96	80	37	19.68	772	71.7	11.1	197	27.2	
17	Aramid-Graphite	110	85	75	30	19.58	772	71.7	13.9	204	36.0	
18	Aramid-Graphite	101	85	75	30	16.92	772	71.7	38.9	204	35.9	Factor of Safety = 2.0
18b	Aramid-Graphite	105	78.5	73	0.0	19.1	772	71.7	15.3	193	39.9	

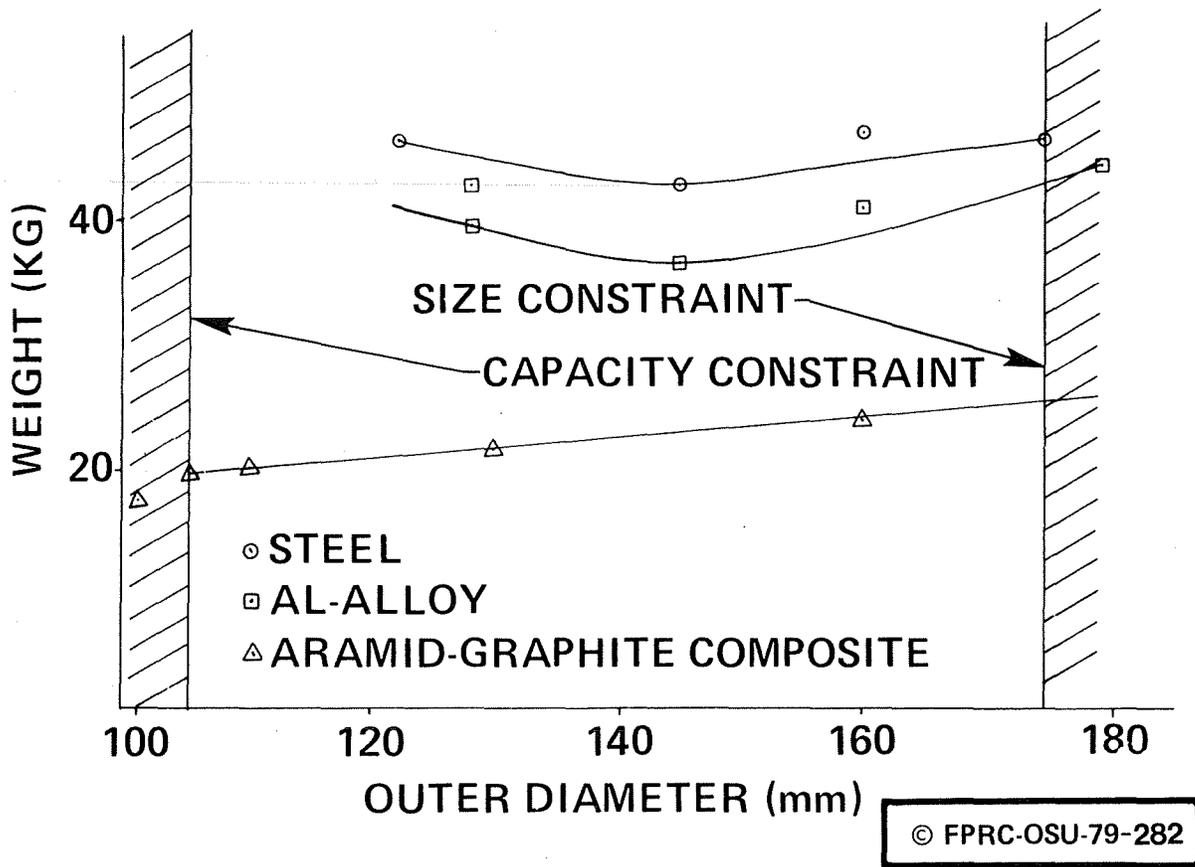


FIGURE 4-7

Weight vs. O.D. for All-Hydraulic Props

2. Steel props exhibit the smallest deflection, Al-alloy being next and composites the largest.
3. Reduction of the factor of safety from four to two decreases the weight approximately 20%--a very significant reduction in the case of steel props.

Figure 4-8 is presented to show the weight vs. deflection trade-offs for optimal all-hydraulic props. As expected, Al-alloy and composite props exhibit higher deflections and lower weights than steel props. One of the reasons for the small difference in weight between steel and Al-alloy is that in both cases, the fluid is a major contributor to the total weight.

Having described the methodology for optimization of hydraulic props, we are now in a position to describe the concept designs which are, structurally speaking, variations of the hydraulic cylinder.

#3 Internal Power, Internal Reservoir

Al-alloy was chosen as the construction material, though as noted above, steel would result in only slightly higher weight. Figure 4-9 shows the conceptual design, and Appendix B contains sketches showing loading dimensions. The prop resembles the Dowty and Commercial props in being self-contained. The pressure release and recovery ring allow the load to be released from a distance, and a bleeding screw (not shown in the sketch) permit air to be released before pressurization. The oil reservoir is located in the lower tube, which also contains the pumping mechanism. Table 4.1 gives the weight break-up and indicates that the fluid constitutes approximately 25% of the total weight. This design

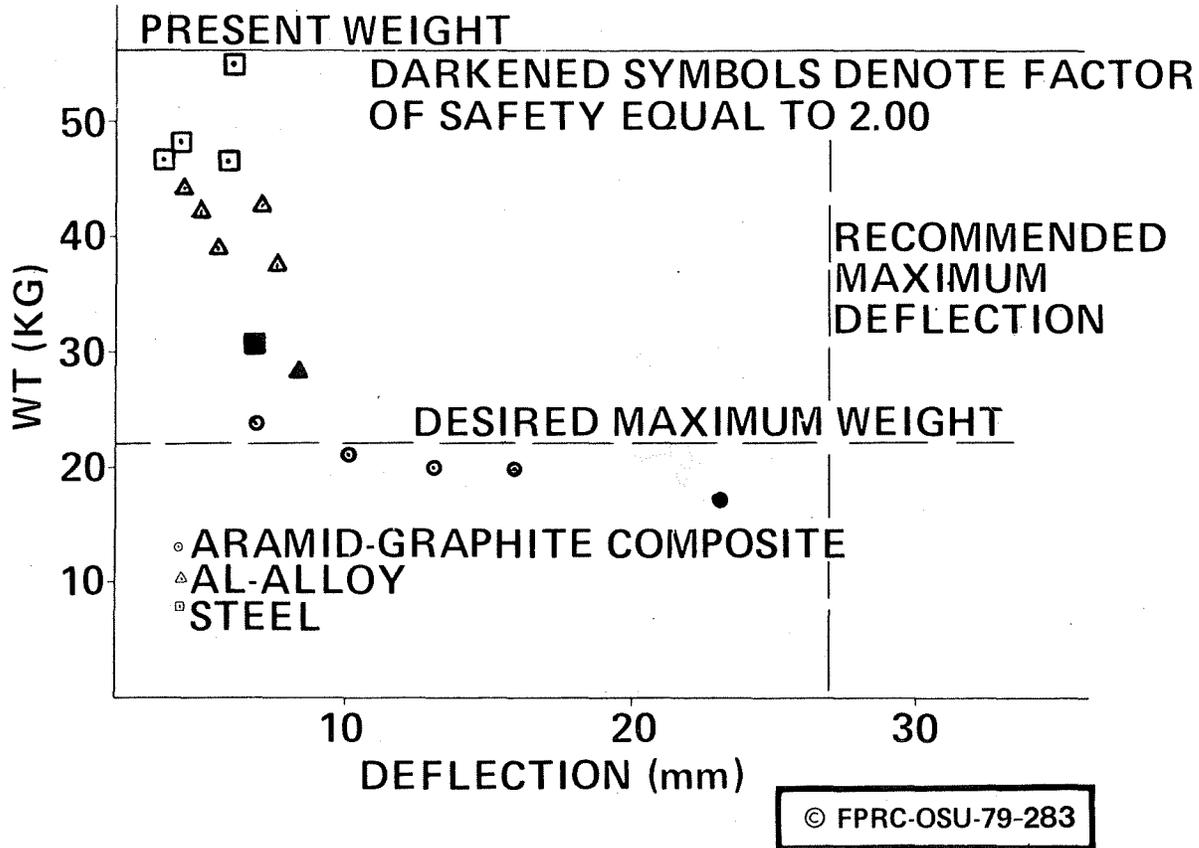
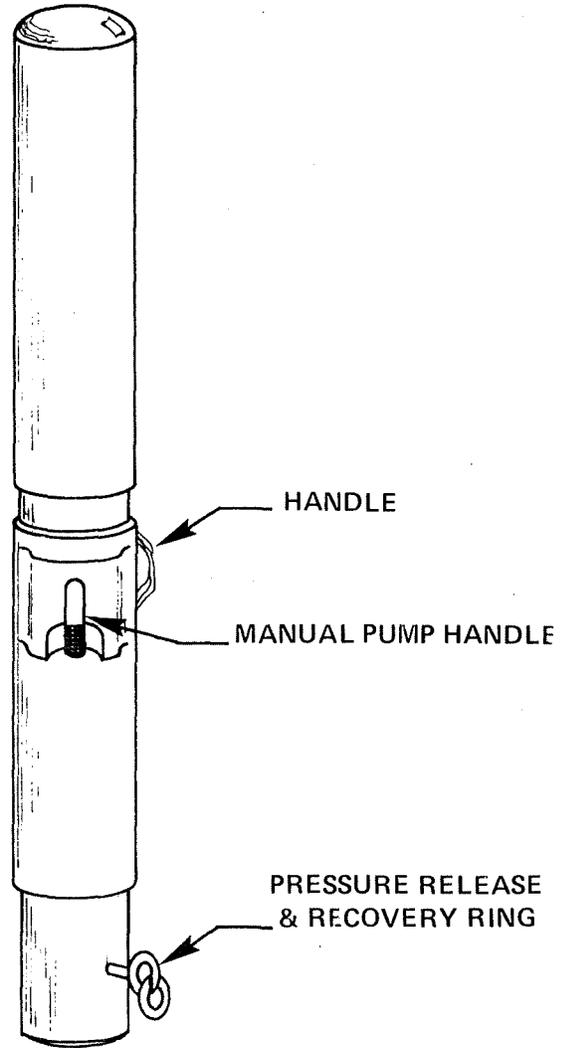


FIGURE 4-8

Weight vs. Deflection for All-Hydraulic Props



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FIGURE 4-9

Hydraulic Design 3 Internal Power Internal Reservoir

compares very favorably with the mechanical designs.

#4 External Power, Internal Reservoir

When this concept was first enunciated, it was felt that providing a power input shaft would reduce weight by substituting the hand pump by a lighter power-operated pump. It was envisioned that rotary power would be available from electrical, hydraulic or pneumatic sources. Discussions with the program monitor indicate that this may not be a promising option due to lack of such power sources. The weight saving does not appear significant. See Figure 4-10 for arrangement.

#5 External Power, External Reservoir

This variant of the hydraulic cylinder (Figure 4-11) is similar to the Gullick-Dobson design of centralized props [3]. It is extended by connecting it to a hydraulic power supply through, say, a quick-connect coupling. Since the general availability of such power packs is still in question, this design has to be considered one of the less promising ones, even though it is the lightest. For application as hydraulic cylinders on mining equipment, it is perhaps the best candidate.

HYDRO-MECHANICAL

The objective in undertaking the furtherance of this design was to combine the best features of hydraulic and mechanical designs. From Table 4-1, it is seen that the mechanical telescopic-tube prop is lighter than the all hydraulic prop. The difference in weight can be ascribed almost entirely to the fluid column. Since the infinite adjustability offered by the hydraulic prop is seldom utilized over the full stroke, a large

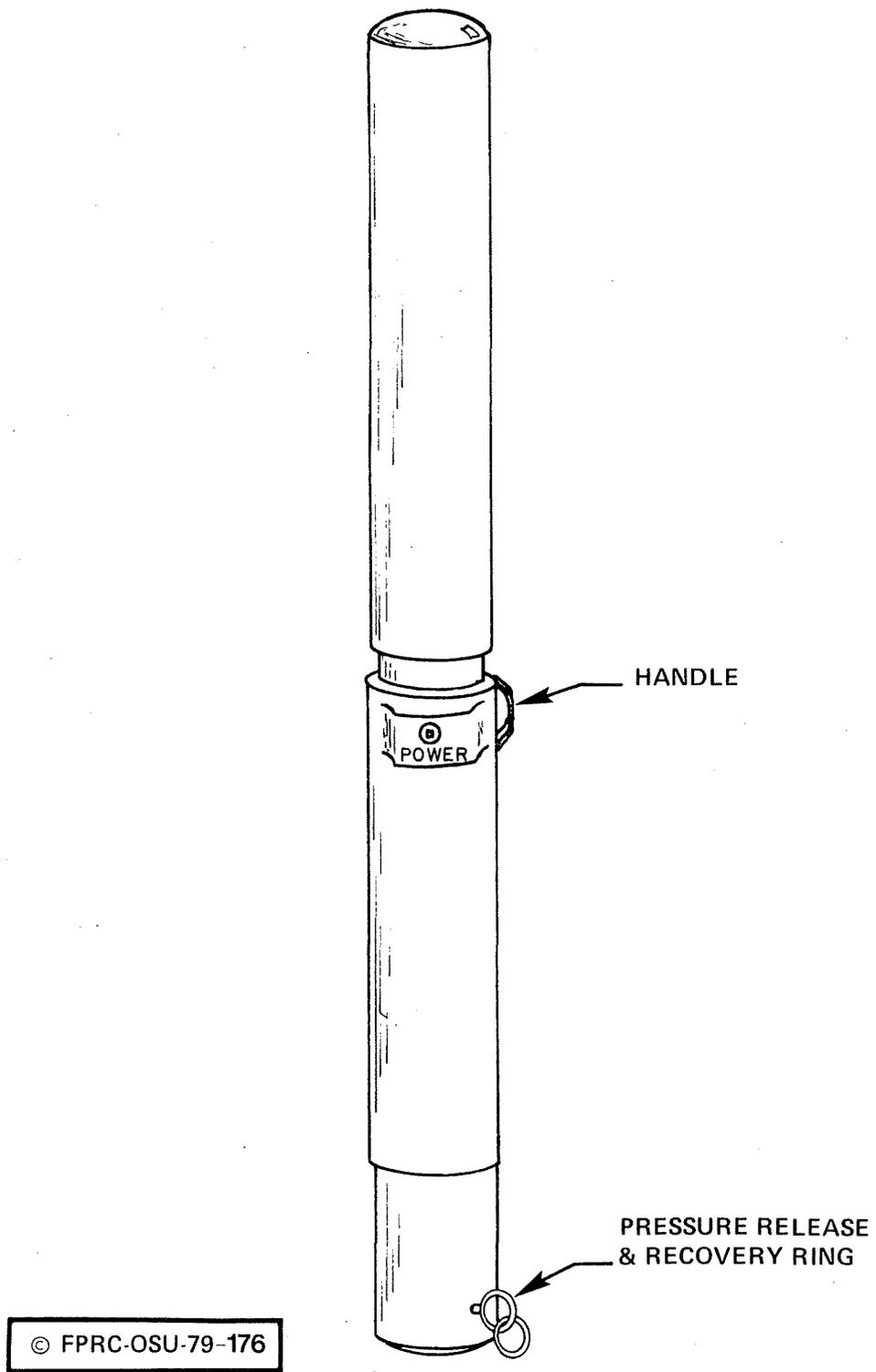


FIGURE 4-10

HYDRAULIC DESIGN 4 EXTERNAL POWER INTERNAL RESERVOIR

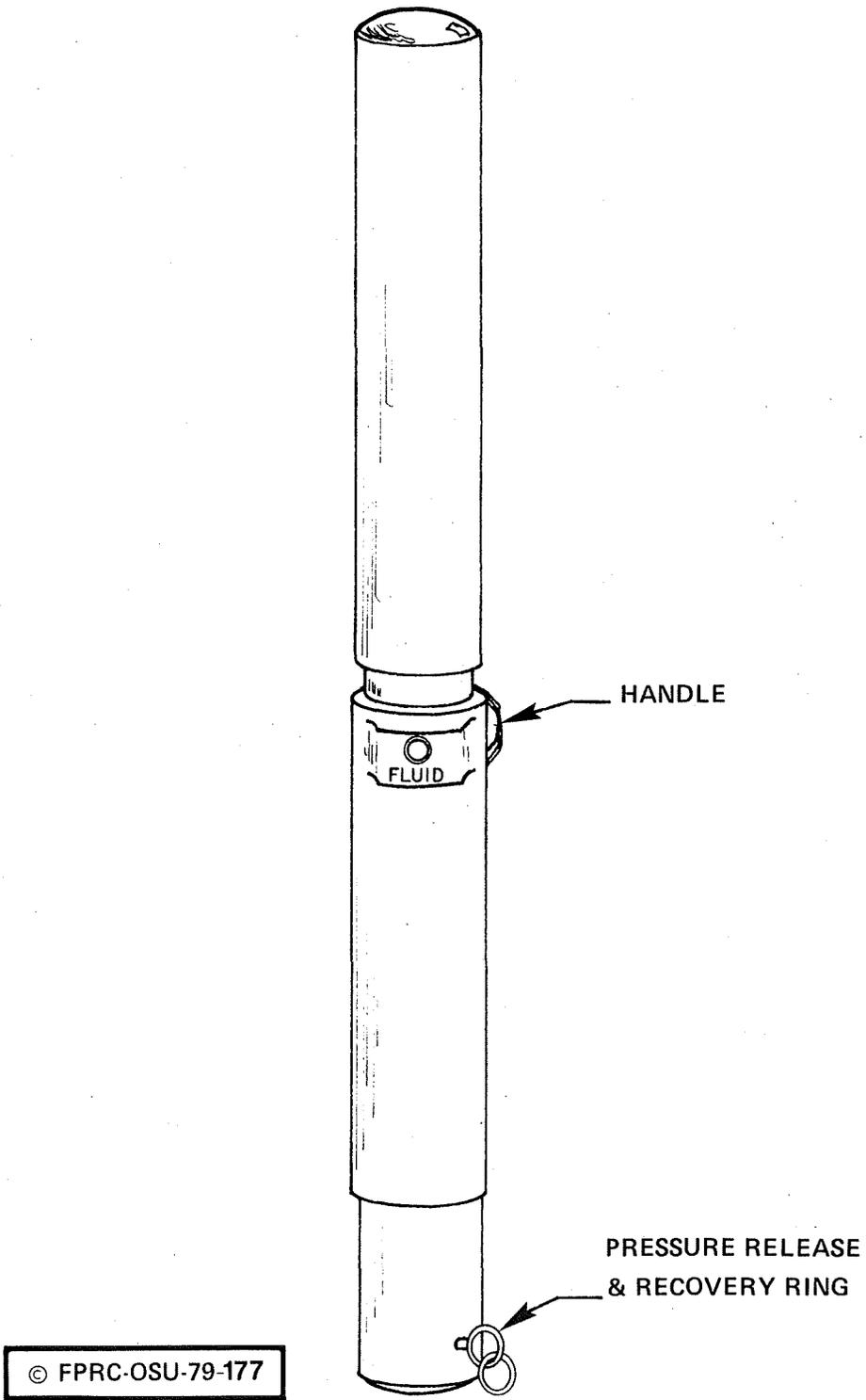
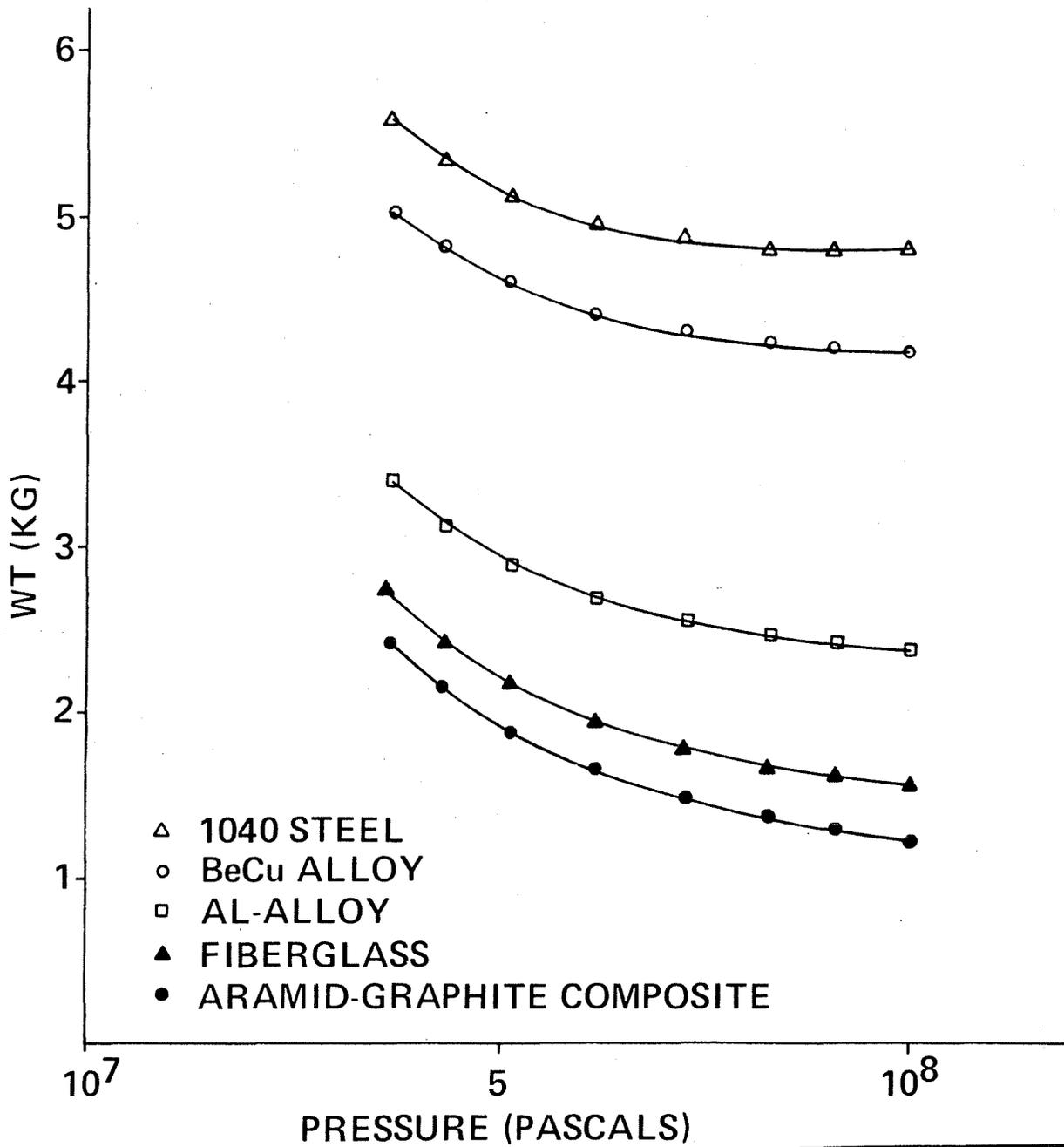


FIGURE 4-11

HYDRAULIC DESIGN 5 EXTERNAL POWER EXTERNAL RESERVOIR

fraction of the fluid column is dead weight. In this connection it may be remarked that the fluid column does not contribute to the strength or rigidity of the prop. The main drawbacks of the mechanical prop has been noted earlier to be its inability to be released from afar and the need for yielding mechanisms, such as springs. By combining the telescopic tubes with a short-stroke cylinder, advantages of both concepts can be obtained in one construction. The mechanical part provides the "coarse" adjustment, in steps of say, 100 mm, so that the prop can be adjusted to the approximate height. The hydraulic cylinder provides the final adjustment as well as the yield mechanism. The mechanical portion, i.e., the tubes, can be designed for bending stresses only, rather than for two-dimensional stresses as in the case of a slender hydraulic cylinder. The short stroke cylinder can be a high pressure jack, say 150 mm stroke, designed primarily as a pressure vessel. Sealing of such a jack would be easier than a long stroke cylinder due to insignificant bending. The stresses in the jack would be primarily hoop stresses, in contrast to the bending stresses in the mechanical portion. The significance of this situation is that non-isotropic materials, i.e., those having different strengths and moduli in different directions, can be advantageously used in the construction. Figure 4-12 is a plot of the wet weight of a jack 150 mm high, with a stroke of 100 mm, when constructed of different materials. It is seen that Al-alloy and the composites reduce the weight by 50% or more. Just as in the case of the all hydraulic prop, variants of the basic design are obtained by locating the pump and reservoir interior or exterior to the prop.



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FIGURE 4-12
 WEIGHT vs. PRESSURE FOR SHORT STROKE CYLINDERS

#6 Manual Power, Internal Reservoir

As depicted in Figure 4-13, this design has a short stroke cylinder on top of a hollow tube. The top portion of the prop consists of a set of telescopic tubes, with the shear-pin adjustment presented earlier. An alternative location for the jack would be the bottom of the prop, but the pump handle would still need to be located about one metre off the ground to facilitate operation. As evidenced by the numbers in Table 4.1, a prop of this configuration would be extremely light.

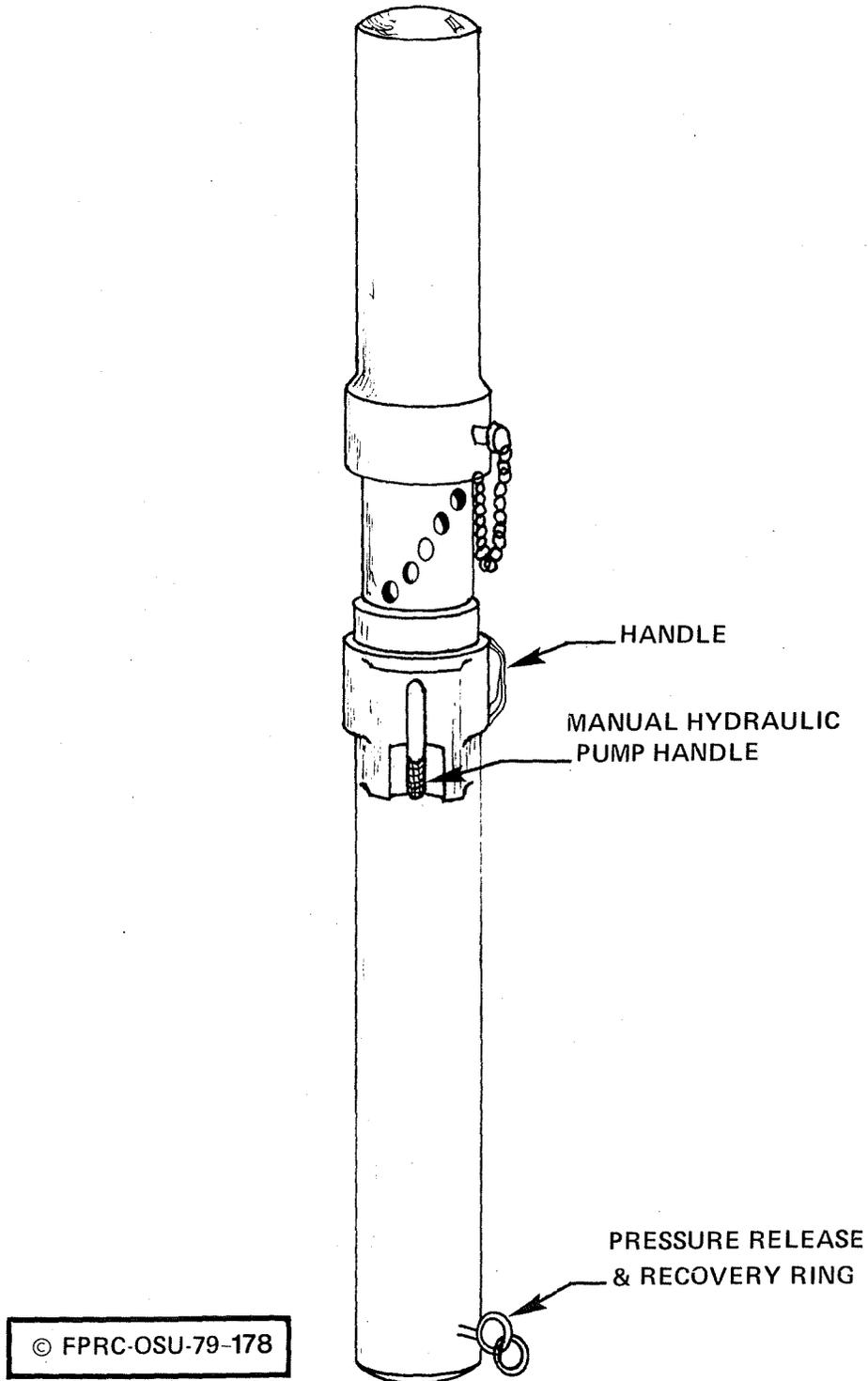
#7 External Power, Internal Reservoir

This design, shown in Figure 4-14, differs from the previous one, only in needing external rotary power. The savings in weight due to elimination of the hand pump is not considered significant.

#8 External Power, External Reservoir

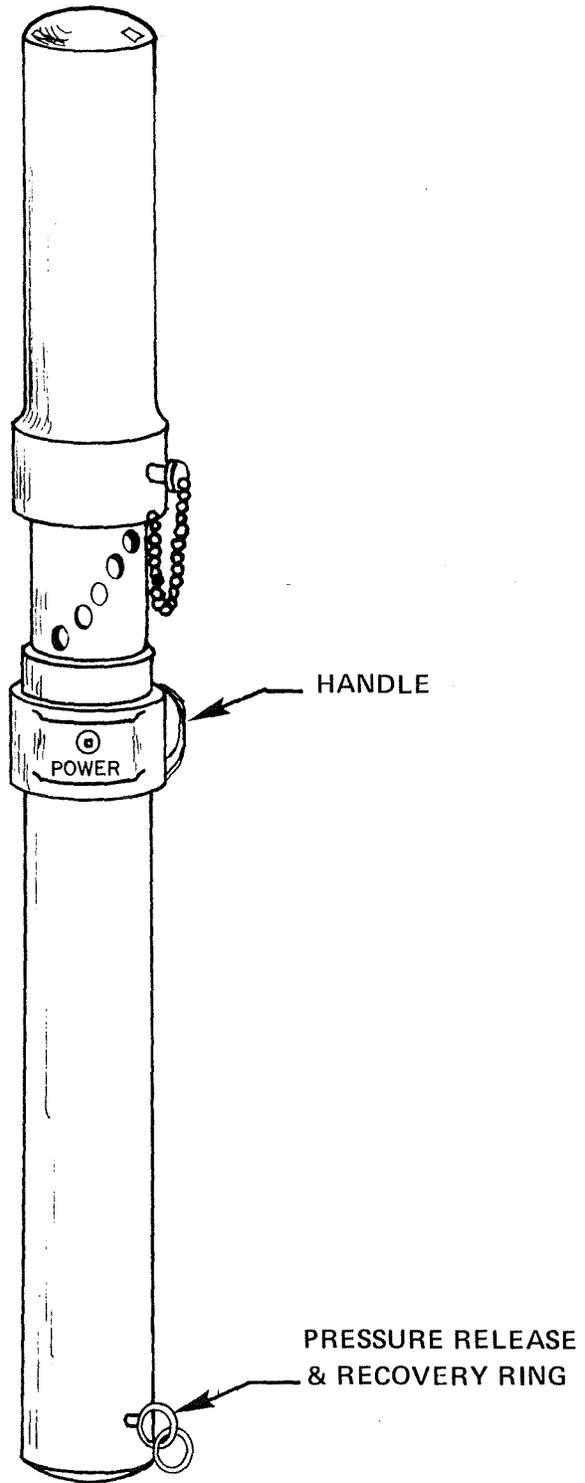
This design, shown in Fig. 4-15, results in the lightest metallic prop, but needs an external source of hydraulic power for actuation. Since the jack stroke is small, the necessary fluid could be carried in a portable hand pump. The design is also suitable for actuation from a centralized hydraulic fluid source.

The remainder of the designs presented in Table 4.1 are variations of the first eight, showing the effects of using aramid-graphite as the constructional material and changing the factor of safety to two. It is seen that in the case of all hydraulic props, the weight-saving obtained by using light construction material is somewhat offset by the virtually unchanged fluid mass. In fact, in the lightest prop (#12), the fluid weighs



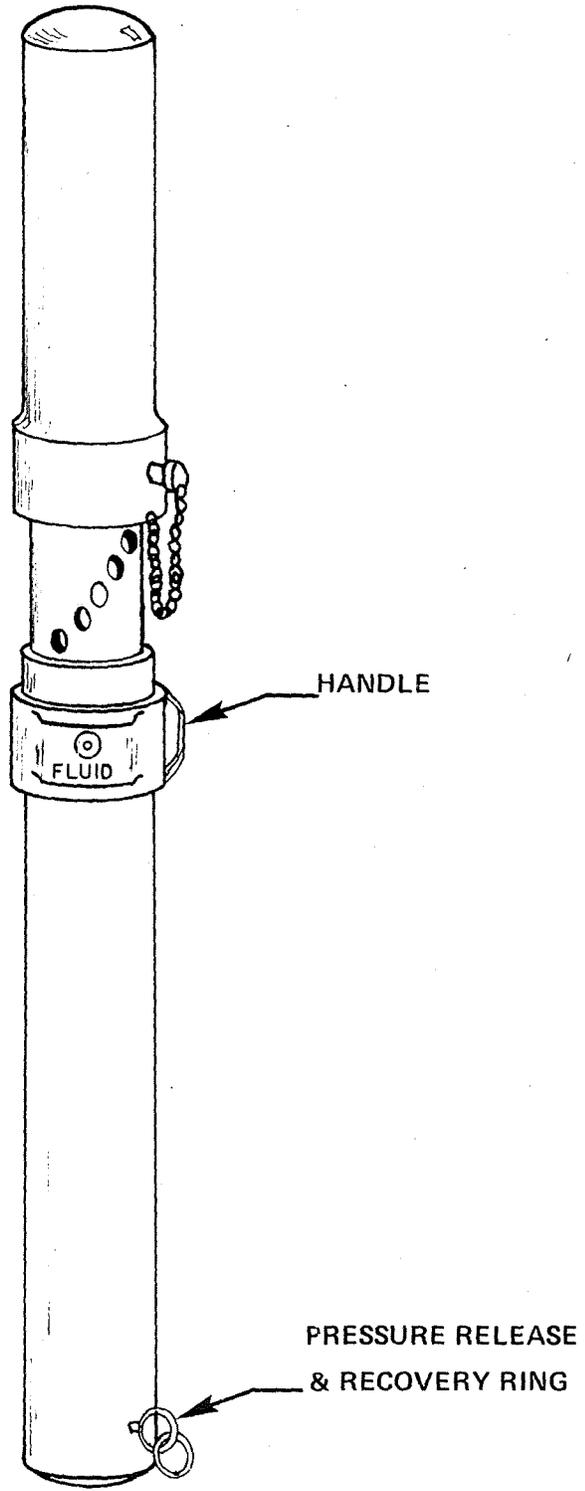
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FIGURE 4-13
HYDRO-MECHANICAL DESIGN 6 MANUAL POWER INTERNAL RESERVOIR



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FIGURE 4-14
HYDRO-MECHANICAL DESIGN 7 EXTERNAL POWER INTERNAL RESERVOIR



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FIGURE 4-15
HYDRO-MECHANICAL DESIGN 8 EXTERNAL POWER EXTERNAL RESERVOIR

about one-third of the entire prop.

Presentation of dimensions and weights of props made of composite material might lead the unwary reader to the conclusions that design and manufacturing techniques for such material are not far different from that for conventional, isotropic materials. Such, however, is not the case. In the next part of this chapter, a brief discussion on the use of fiber composites in prop construction is presented. For a discussion of manufacturing techniques, the interested reader may refer to the citations in Appendix A.

FIBER COMPOSITES IN ROOF SUPPORT PROPS

It is seen that use of steel as the material of construction results in low wall thickness coupled with large diameters. Further reduction in wall thickness, i.e., below 3 mm, would make the props extremely susceptible to accidental damage and misuse. Use of aluminum alloys results in larger wall thicknesses without increase in weight. Here again, wall thicknesses below 5 mm are undesirable in view of damage susceptibility. Thus, it appears that, using conventional materials of construction, a weight of 28 kg is the minimum that can be attained for the all hydraulic prop. Since the aerospace, automobile and chemical industries have achieved impressive weight and cost reductions by substituting composites for homogeneous material, an exploration of their usage for the present application is worth consideration.

Basically, composites consist of two dissimilar materials, one of which is in the form of fibers or filaments, and the other is the matrix or cementing material. Glass fiber is perhaps the most widely known,

though graphite, boron and organic filaments are being increasingly used. The fibers may extend as continuous filaments and be oriented in a specific direction, or they may be chopped and randomly oriented. The latter is very common in low-strength applications, mostly because of ease of manufacture. High strength applications, on the other hand, almost always exploit the properties of the fiber by resorting to filament construction.

The strength of composites depends on:

1. Tensile strength of the mono-filament fiber in three dimensions.
2. Compressive strength of the mono-filament fiber.
3. Percentage by volume (v/o) of fiber in the composite.
4. Shear and adhesive strength of the matrix.
5. Orientation of the mono-filament with respect to external loads.

Since the tensile strength of fibers is usually an order of magnitude larger than that of conventional homogeneous materials, see Table 4-3, initial expectations of load-carrying capacities of composites are very high.

However, in actual practice, the load is not carried completely by the fiber in tension, and it has to be transmitted by the matrix, usually in shear, and hence the effective strength of the composite is less than that of the fiber. To the structural designer engaged in preliminary design, the properties of the composite are more pertinent than that of the pure fiber. Table 4-4 summarizes typical properties for some composites. It is seen that the strength of the composite in tension is approximately half of that of the fiber, given in Table 4-3. Also, that the shear strength is much less than the tensile or compressive strength. This reflects the effect of the matrix on the overall strength. It may also be

TABLE 4-3. Fiber and Wire Properties [19].

Fiber or wire	Density, p lb/in. ³ (kN/m ³)	Tensile strength, S 10 ³ lb/in. ² (GN/m ²)	S/p 10 ⁵ in. (km)	Tensile stiffness, E 10 ⁶ lb/in. ² (GN/m ²)	E/p 10 ⁷ in. (Mm)
Aluminum	.097 (26.3)	90 (.62)	9 (24)	10.6 (73)	11 (2.8)
Titanium	.170 (46.1)	280 (1.9)	16 (41)	16.7 (115)	10 (2.5)
Steel	.282 (76.6)	600 (4.1)	21 (54)	30 (207)	11 (2.7)
<i>E</i> -glass	.092 (25.0)	500 (3.4)	54 (136)	10.5 (72)	11 (2.9)
<i>S</i> -glass	.090 (24.4)	700 (4.8)	78 (197)	12.5 (86)	14 (3.5)
Carbon	.051 (13.8)	250 (1.7)	49 (123)	27 (190)	53 (14)
Beryllium	.067 (18.2)	250 (1.7)	37 (93)	44 (300)	66 (16)
Boron	.093 (25.2)	500 (3.4)	54 (137)	60 (400)	65 (16)
Graphite	.051 (13.8)	250 (1.7)	49 (123)	37 (250)	72 (18)

TABLE 4-4

Typical Room-Temperature Major-Axis Properties, Laminated Epoxy

Matrix Composites

(From Ref. 20)

	Typical Fiber Content (% vol)	Strength ksi (GN/m ²)				Modulus 10 ⁶ psi (GN/m ²)		
		Flexure	Tension	Compression	Shear	Tension	Compression	Flexure
Continuous filament reinforced								
Glass								
E-glass	65		180 (1.24)	140 (.97)	9.2 (.063)	7.8 (53.8)	7.5 (51.7)	
S-glass	65		235 (1.62)	164 (1.14)	11.8 (.081)	10.5 (72.4)	12.4 (85.5)	
Boron	50	260 (1.79)	210 (1.45)	490 (3.38)	14 (.097)	30 (206.9)	33 (228)	280 (1931)
Graphite								
High strength	60	230 (1.59)	200 (1.38)	230 (1.59)	15 (.103)	21.0 (145)	20.5 (141)	220 (1517)
High modulus	60	115 (.79)	110 (.76)	115 (.79)	9 (.062)	29.0 (200)	28.5 (197)	
Intermediate properties	60	200 (1.38)	190 (1.31)	175 (1.21)	13 (.09)	19.2 (132)		
Organic								
PRD-49-III	65		240 (1.66)	40 (.28)	9 (.062)	11.0 (76)		
Balanced construction fabric-reinforced								
Glass								
E-glass	55		63 (.43)	62 (.43)		3.6 (24.8)	3.6 (24.8)	84 (579)
Organic								
PRD-49-III	55		75 (.52)	37 (.26)		4.5 (31)	4.5 (31)	

noted that the continuous filament construction is about three times as strong as the fiber-reinforced construction.

Design of parts using composites is much more difficult than with homogeneous materials, but the following comments by various authorities should help in material and process selection.

"Under proper design and controlled fabrication, hoop tensile strengths of filament wound items can be achieved of over 500,000 psi (3.45 GPa), although a strength of 210,000 psi (1.45 GPa) is more frequently achieved." [21]

"Toughness decreases in metals as strength is increased. Reduced weight, increased strength and improved reliability (increased toughness) are opposing properties. Present alloy steels are notch sensitive above approximately 200,000 psi (1.38 GPa) tensile yield strength." (ibid)

"Limitations of Filament Wound Construction:

1. Lack of ductility.
2. Low modulus of elasticity is a serious disadvantage. Only moderate improvements appear feasible.
3. Interlaminar shear constitutes one of the possible limitations on filament wound parts.
4. Filament wound structures have lower ultimate bearing strengths-(are) more rigid and less ductile. (Have) less ability to withstand stress concentrations around holes and cut-outs.
5. Cost of filament wound parts is low only when volume production is achieved. Manufacturing processes should be mechanized and completely automated to obtain the close tolerances which are required in filament wound structures." (ibid)

"Reinforced Plastics (RP) is a relatively new material of construction and does not have a comparable wealth of experience and demonstrated performance as compared with steel and many metals." [22]

"Metals can be considered as homogeneous and isotropic. The degree of homogeneity in Reinforced Plastics, however, is a function of the manufacturing technique. The low ductility of glass-reinforced plastics does not accommodate stress relief in areas of high stress concentration. This means that stress analysis formulas used with metals may not always be applied similarly to Reinforced Plastics." (ibid)

"Minimum design factors of 4.00 (factors of safety) are suggested for Reinforced Plastics for structural applications. Responsible manufacturers use a minimum design factor of 5.0 for filament wound fiber glass pipe." (ibid)

The importance of the matrix is emphasized in the following remark: "In order for the matrix to transfer the load, it must have a sufficiently high cohesive shear strength, and it must provide high interfacial shear strength. The fiber-matrix interface is in many cases the weak link in the load transfer mechanism within a fiber-reinforced composite." [23]

It has been pointed out in Chapter III that the all-hydraulic prop is subjected to biaxial stresses--internal pressurization results in hoop stresses and end-loading results in bending stresses. Since the latter depend on the deflection, which in turn depends on Young's Modulus, the ideal composite would be stiff in addition to being strong and light. The most common reinforcement material, E glass, exhibits low stiffness (see

Table 4-4). Attempts to design an all-hydraulic prop using an E-glass composite (689.7 MPa tensile strength, 39 GPa Young's Modulus) failed to give a prop smaller than 180 mm diameter. Of the other material produced in filament form, Boron, Aluminum and graphite have reasonably high stiffness [20]. Kevlar, the trade name of an aramid fiber made by DuPont [24] has not only a higher specific tensile strength than graphite and E-glass, but also a high modulus (131 GPa). By combining aramid with graphite, a hybrid composite with the strength and stiffness of graphite and the impact strength of the aramid is obtained [24, 25].

In view of the "good" properties of aramid-graphite, it was chosen as the composite material for use in the computer analysis. A fiber ratio of 40% Kevlar 49 and 60% A-S graphite was selected. Properties of this material are described by Norman [24]. Since the material is anisotropic, both the strength and modulus depend on the direction of loading. The values used in the computer analysis, and included in Table 4-2, correspond to 0° compression. A detailed design of any part made from an anisotropic material should take into account the material properties in other directions.

The construction of hydraulic cylinders and piston rods from aramid-graphite and other similar composites needs manufacturing processes quite different from that for steel or aluminum-alloy. Continuous filament usage requires a winding technique in which inserts, ports, etc. have to be included at the very beginning. Breaking the continuity of a filament, e.g., by machining or cutting, can seriously reduce the local strength and stiffness. [26, 27]

OTHER DESIGN CHARACTERISTICS

SAFETY:

One of the important considerations in underground mining applications is that of safety. The atmosphere inside a mine can be gassy, and mine accidents due to explosions set off by sparks are not unknown, specifically, the impact of aluminum on rusty steel is considered dangerous by some authorities [28]. Since Al-alloy results in light structures and has a reasonably good stiffness, its use for props should not be ruled out because of the possibility of sparking under certain circumstances. Bailey [28] indicates that paint coatings and covering with neoprene or a soft metal can substantially reduce the risk of sparking.

Safety, in other respects, is improved by making the height adjustment simple and rugged. Provision of a load indicator would help in ascertaining if the prop is yielding or if it is being over-pressurized during set-up. From this point of view, the hydraulic and hydro-mechanical props are superior to the mechanical designs, since their yield level can be easily adjusted.

FACTOR OF SAFETY:

Nominal factors of safety of four and two have been used in designing the optimal props presented in Table 4-1. Apart from variations in material properties, and effects of stress concentration, eccentricity of loading, end-fixity conditions and impact loads are probably the major unknown factors in the design of the props. Improper application can obviously result in the imposition of higher eccentricities than has been

used in the analysis. However, for a given factor of safety, the all-hydraulic props, presented here are considered "safer" than those currently in use, in view of the more realistic mathematical analysis. Put in other words, a lowering of the factor of safety from four to, say, two, does not reduce the actual "safety" substantially. However, it should be pointed out that the corresponding reduction in weight is rather small, since the fluid mass is reduced only slightly. A factor of safety of three would probably result in a good compromise.

The analysis for both mechanical and hydraulic props has assumed pin-ends for the columns. Since the floor and roof of mines are generally rough, the friction at the ends, and consequently, the load capacity of the props, will be more than the numbers shown. Experimental data can be used to calculate more realistic values for the end-fixity. Designing of the props as fixed-end columns will increase the load capacity but may impose undesirable stresses on the roof and floor. All the props shown in Figures 4-1 through 4-15 have dished ends to approximate pin-ended columns. The projected surface area is large enough to result in an average stress less than 18 MPa on the bearing surface. By using suitably shaped pads, this stress can be reduced still further.

The occurrence of impact loads on the props has not been sufficiently documented. All the analysis reported herein is valid for gradually applied static loads. If the props are subjected to impact loads, the materials of construction would have to be chosen accordingly, and the load capacity would be different. If the cylinders were used as part of a centralized hydraulic system, fatigue life could be an important consideration. Fatigue strength data on steel and Al-alloys are fairly well-documented in the

literature. Such is, however, not the case with composites. Aramid-graphite is an exception, and Norman [24] presents the results of some fatigue tests.

EASE OF HANDLING:

The props shown in Figures 4-1 through 4-15 were designed with an upper size constraint of 175 mm. With appropriate handles, all of them can be transported from site to site. The mechanical and hydro-mechanical props can be dismantled into two sections.

POSSIBLE FAILURE MODES:

Failure modes can be classified as being repairable/unrepairable and gradual/catastrophic. Overstressing due to overloading or poor set-up can result in catastrophic and unrepairable damage. High bending stresses are most likely to damage the bearing surfaces of the cylinder/rod interface. High internal pressure can lead to rupture anywhere along the length of the cylinder. Studies have shown that pressurization of the (hollow) rod does not increase its strength. Hence, if a "dry" rod is used, rod failure will be due to bending stresses. Leakage of oil due to worn out seals and poor seating surfaces in relief valves is generally repairable and gradual, just as in the case of ordinary hydraulic cylinders and valves. Companies which manufacture props often furnish maintenance and testing instructions, and by using similar procedures, all designs of props can be kept in good working conditions for a finite life.

MATERIAL DURABILITY:

Apart from fatigue, environmental factors, i.e., temperature, humidity,

water and gases, are the ones most likely to influence the life of a prop. Since the atmosphere in mines can be more severe than above ground, and the water contains corrosive chemicals, special care should be taken in designing in the necessary protection. Table 4-5 summarizes significant physical and chemical properties of materials considered good candidates for prop manufacture. Corrosion can often be engendered by using highly dissimilar metals in close contact in a damp atmosphere (galvanic corrosion). Steel surfaces should be protected by painting, plating, galvanizing or equivalent processes. Since the temperatures in underground mines are expected to be relatively steady and in the 10°C to 40°C range, problems of creep are considered unlikely. Some composites have a tendency to absorb moisture and swell but can be protected by using sheaths and sealing the ends.

FLUID COMPATIBILITY AND SEALING ABILITY:

Steel and Al-alloys have been successfully employed in hydraulic systems for many years, and no problems of fluid compatibility or sealing have remained unsolved. Cylinders made of composites could have a permanent mandrel which can be internally finished before or after filament-winding. Since filament-wound cylinders have been successfully used as rotary and linear bearings, the surface finish, necessary to keep leakage within desirable limits, should be achievable. The design of seals is expected to be no different for composite cylinders than for metallic ones.

EASE OF MANUFACTURE:

The availability of tubular products in steels and Al-alloys makes the manufacture of metallic props a fairly straight-forward process. The

Table 4-5. Selected Physical and Chemical Properties of Metals and Composites.

Material	Tensile Strength 0° MPa	Comp. Strength 0° MPa	Young's Modulus GPa	Density kg/m ³	Fatigue Strength MPa @ 10 ⁶ cycles	Chemical Resistance					
						High Humidity	Absorption %	Oxidation	Organic Solvent	Hyd. Fluid	Weak Acid
² 4340 Steel	1193	--	206	8016	350-700	Fair	--	Poor	Exc	Exc	Poor
² Al-Alloy 7075 - T6	565	--	68	2801	161-259	Good	--	Good ³	Exc ³	Exc ³	Poor ³
² BeCu Alloy	814	--		8267			--				
Boron Epoxy	1586	2482	206	1997	980	Fair		Exc	High	High	
Graphite Epoxy	1448	1172	127	1526	1014 ¹	Poor	2.7	Poor	High	High	Good
Kevlar-49 Epoxy	1379	275	75	1387	980	Exc		--	High	High	
E-Glass	1103	620	39	1803	224	Fair	0.11-1.85	Exc	Good	Good	Exc
S-Glass	1510	827	43	1830	302	Fair	0.11-1.85	Exc	Good	Good	Exc
Kevlar-49 Graphite Hybrid	1034	772	108						High	High	

* All properties are for unidirectional composites with force directed along the fiber axis.

¹ Value at 10⁷ cycles. from Ref. 29.

² From Ref. 15.

³ From Ref. 30.

manufacture of composite props would have few industrial precedents, mostly in the aerospace industry, e.g., landing gear parts, spars, wing pands, etc. Some of the manufacturing techniques are briefly explained in reference thirty-one. Case histories of applications can be found in reference twenty. The important point to note is the necessity for an integrated approach to design and manufacture. Ports, bugs, handles, etc., cannot be incorporated as afterthoughts without seriously affecting the strength of a composite structure. This is especially true of continuous filament-wound structures, where the breaking of filaments by machining/cutting can reduce the load capacity of an entire layer. In the use of composites, special care has to be exercised in designing any composite-metal interfaces that may be needed. Conventional joining techniques, e.g., screws, rivets, welding, are often unapplicable. However, as the aerospace industry has demonstrated, the problems are not unsurmountable.

COST ANALYSIS:

The following factors exert significant influence on the cost of mine props:

1. materials of construction.
2. complexity of design.
3. manufacturing technology.
4. life expectancy.
5. MSHA, OSHA or other mandatory requirements.
6. Production/Sales volume.
7. standardization and interchangeability.

Material costs are often given on a weight basis. Comparison of such figures should be done with caution since the same material can be made available at different stages of manufacture, e.g., steel in ingots or tube-stock, glass as filament or chopped fiber or moulding compound. The estimation of manufacturing cost is especially difficult in the case of composites, since historical records on similar components may not exist. Obviously high volume production enables the distribution of the fixed costs, in any process, over a larger number of units.

Table 4-6 presents cost estimates of the alternative designs described in this chapter. The total cost is broken up into material and manufacturing costs. Rosen and Kulkarni [32] furnished the price of Al-alloy, \$4.48/kg, used in the estimation. This number is considerably higher than that given in other references. Since a strong and stiff alloy is needed, it is expected that cheaper commercial grades of Al-alloy may be unsuitable. The price of steel was taken as \$1.61/kg and is based on data furnished by Fox and Miller [30]. The cost of aramid-graphite was taken as \$1.03/kg from Ref. [33]. Significant reductions in cost of not only this material, but other filament materials can be expected [29].

The manufacturing costs for metallic cylinders/tubes were projected from price lists for hydraulic cylinders and jacks. The estimation of manufacturing costs for the aramid-graphite composites involved the use of a cost factor given by Rosato [34]. The factor is dependent upon the manufacturing technology being used to develop the finished product.

All costs in Table 4-6 are production unit costs and do not include and R & D, investment, nonrecurring or operation and maintenance. The production unit costs are strictly investment recurring costs.

TABLE 4-6
COST ESTIMATE FOR ALTERNATE DESIGNS

S. No.	Design Type	Material Cost \$	Mfg. Cost \$	Total Cost \$	Remarks
1	Mechanical Wedge (Al-Alloy)	100	500	600	
2	Mechanical Screw (Steel)	70	630	700	
3	Hydraulic-Int. Power, Int. Reservoir (Al-Alloy)	60	440	500	
4	Hydraulic-Ext. Power, Int. Reservoir (Al-Alloy)	50	400	450	
5	Hydraulic-Ext. Power, Ext. Reservoir (Al-Alloy)	50	350	400	
6	Hydro-Mech., Int. Power, Int. Reservoir (Al-Alloy)	50	600	650	
7	Hydro-Mech., Ext. Power, Int. Reservoir (Al-Alloy)	50	600	650	
8	Hydro-Mech., Ext. Power, Ext. Reservoir (Al-Alloy)	45	555	600	
9	Hydraulic-Int. Power, Int. Reservoir (Aramid Graphite)	1,000	3,000	4,000	
10	Hydraulic-Int. Power, Int. Reservoir (Steel)	30	420	450	
11	Hydraulic-Int. Power, Int. Reservoir (Al-Alloy)	30	420	450	
12	Hydraulic-Int. Power, Int. Reservoir (Aramid Graphite)	700	2,100	2,800	
13	Hydro-Mech., Int. Power, Int. Reservoir (Al-Alloy)	40	460	500	

Factor
of
Safety =
2

MAINTAINABILITY

All prop designs furnished above are considered repairable. A strict inspection, maintenance and record-keeping program will ensure the reliability of the components. Every re-conditioned prop must be subjected to load tests before it is released for usage. Seals, o-rings and moving parts in the hand pump and relief valve are the only parts expected to be replaced within the life of a prop.

ADDENDUM

A prop design which was conceived during discussions with the project monitor was the mechanical screw-pin prop, shown in Fig. 4-16. This design attempts to combine the best features of the telescopic tube and screw arrangements. The former provides the rough adjustment while the latter the fine adjustment. The screw need be only about 200 mm long for an effective travel of 100 mm, and hence there is a considerable weight saving over the screw-type prop.

A screw thread of the Acme type, with a pitch of 12.7 mm ($\frac{1}{2}$ inch), root diameter 108 mm and thread height 6.35 mm, and made of SAE 1040 steel, would be satisfactory. An internal bore of 100 mm would lighten the screw, which would still be capable of carrying a compressive load of 196 kN. Weight of the screw would be 3.9 kg. The nut with which the screw would engage would weigh approximately 4.2 kg.

The telescopic tubes are designed for buckling strength. If the inside tube has an O.D. of 133.35 mm, an I.D. of 114.3 mm and is made of Al-alloy of Young's modulus 68.97 GPa, the safe Euler load for a column

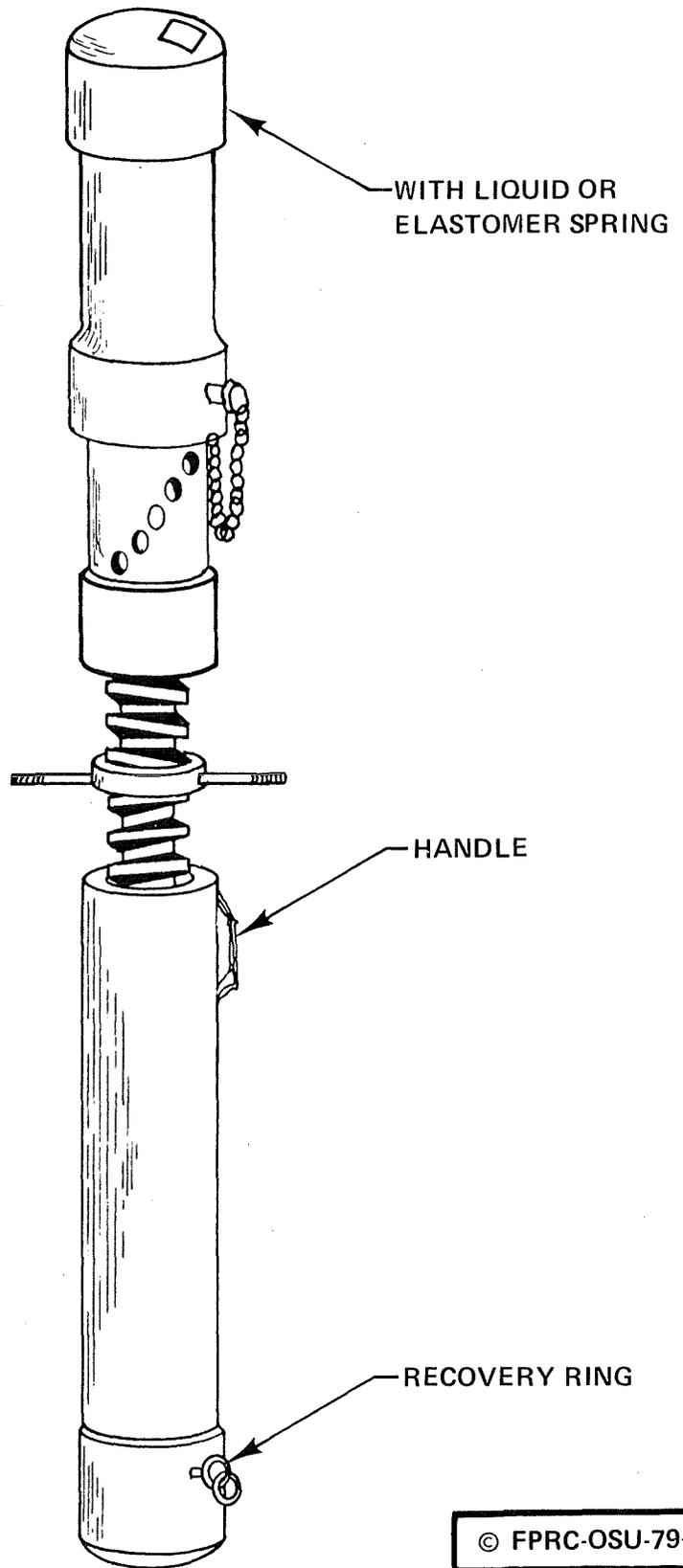


FIGURE 4-16
MECHANICAL-SCREW PIN DESIGN 9

2.44 metres is 818 kN which is more than that needed. The outside tube can have an I.D. of 133.4 mm and an O.D. of 146.1 mm. The combined weight of the tubes is approximately 21.5 kg.

A mechanical prop of the above design can also be provided with a liquid spring. The total weight of such a prop would be approximately 37 kg.

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

This project was aimed at developing alternative optimal designs of light-weight roof supports for underground coal mining operations. In the first phase of the project, existing mechanical and hydraulic temporary roof support-designs were to be reviewed; in the second phase alternate conceptual designs were to be evolved and no more than five to be finalized. Apart from load and extension requirements, the designs were to be evaluated from a number of considerations.

A study of existing mechanical and hydraulic prop designs has revealed the following:

1. Mechanical props, mostly of the screw jack on timber-post variety, are rarely of the requisite capacity (195 kN). They are not of the yielding variety and can be overloaded during set-up.
2. Hydraulic props are mostly of the integral reservoir, integral pump, i.e., self-contained, variety. Manual props in the 100-200 kN range are available. Most of them weigh about 60 kg for the height under consideration.

Conceptual designs developed by this project were categorized as being mechanical, hydraulic or hydro-mechanical. Mechanisms for prop extension were classified as being suitable for rough or fine adjustment. Of the mechanical designs, the telescopic tube arrangement was investigated and found to be excellent for rough adjustment. Using circular cross-section

tubes resulted in stiffness in all directions and an extremely light design. Transfer of load from one tube to another was through a shear pin, and fine adjustment could be provided either by wedges or a screw. Conventional metal springs were found to be kinematically capable of making the props yieldable, but increased the weight of the prop to an unacceptable value. A liquid spring was found to be capable of fulfilling the same needs and weighing substantially less. As an alternative to wedges, the use of a short screw was investigated and found to result in a viable design.

The mechanical screw design was found to weigh considerably more than the target value, mostly because of the steel screw. With the addition of overload protection, i.e., yielding characteristics, through springs, the design became heavier.

The hydraulic props were designed as hydraulic cylinders. Historically, they have been treated as uniform stepped columns. Using a computerized analysis technique developed at the Fluid Power Research Center, Oklahoma State University, a more realistic mathematical model has been used to design the hydraulic cylinders constituting the prop. Since the load capacity of such cylinders depends on both the tensile strength, as well as the Young's Modulus, material selection has to take both properties into consideration. Al-alloy was found to result in the lightest props, but steel was not far behind. Larger tube diameters generally yielded lighter "dry" props but beyond 170 mm the wall thickness became too small to sustain lateral loads. Additionally larger props would operate at a low pressure and weigh considerably more when filled with oil. The optimal all-hydraulic prop weighed 31.84 kg dry and 42.57 kg wet.

The main reasons for the seemingly excessive weight of all-hydraulic props are the following:

1. For the length under consideration, the props function as columns of intermediate length, and both, tensile strength and Young's modulus, are important. Increasing the former by itself does little to improve the capacity. For a given class of alloys, the latter is generally the same. I.E., use of high tensile strength material does not result in significant weight savings.

2. The prop carries "dead" weight in the form of fluid. The fluid column does not contribute to the stiffness of the prop. For a given weight of wall material, stiffer props are obtained by increasing the diameter, but this increases the fluid weight and thus defeats the original purpose.

It is concluded that the infinite adjustability, which the hydraulic prop provides is achieved at the cost of carrying "dead" fluid. Use of light and stiff material for cylinder fabrication results in less than proportionate weight saving in the final "wet" prop. Nevertheless, it should be emphasized that using the computerized analysis technique, the lightest possible cylinders are obtained.

The hydro-mechanical props combine the best aspects of telescopic tubes and hydraulic cylinders. The tubes, which provide rough adjustment, can be made lighter than a hydraulic cylinder, and the hydraulic cylinder, having a short stroke, need be designed only for internal pressure. Consequently the hydro-mechanical prop results in the lightest design, without compromises in performance.

The use of composite materials for manufacture of lightweight roof supports has been explored and found to be a viable option. Most of the commonly used composites, e.g., fiberglass, PRD-49 do not have a high stiffness. A hybrid composite, aramid-graphite, was found to have good stiffness and impact strength, in addition to high specific strength. It resulted in the lightest prop. Other composites of similar material properties could conceivably be made to order, but would be rather expensive.

Using a factor of safety of two saved appreciable weight in the hydraulic and hydro-mechanical props. Use of this lower factor of safety is considered to result in as "safe" a design using computer analysis as the use of a safety factor of four with Euler formulas.

RECOMMENDATIONS

1. Props of the following type should be detail-designed, manufactured and tested:

- a. Mechanical screw-pin type (Design No. 9, Fig. 4.16).
- b. Hydraulic, (Design No. 3, Fig. 4.9) of steel, Al-alloy and aramid-graphite.
- c. Hydro-mechanical (Design No. 6, Fig. 4.13) of Al-alloy.

Testing should specifically include measurement of eccentricity of loading and end-moments. Critical parts of the prop should be strain-gaged and prop deflection measured.

Tests should also be performed to evaluate:

- a. fatigue strength
- b. corrosion/temperature, humidity resistance

- c. lateral load resistance
- d. leakage characteristics and contaminant sensitivity (for hydraulic designs)

2. Efforts should be made to ascertain actual eccentricities, end-moments and yield in a variety of mines. These data can then be used to furnish input values for eccentricity and end-pin friction for the computerized analysis. By having reliable estimates for these quantities, the nominal factor of safety could conceivably be reduced without compromising genuine safety.

3. The optimization reported herein was based on a prop capacity of 196 kN. It is possible that a different capacity could result in the best capacity per unit weight. An investigation of the trade-off between capacity and specific capacity will reveal the optimal capacity.

4. Extensive stress analysis of composite materials should be performed to exploit their directional properties.

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APPENDIX A
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- Mining
Roof Support
- Graphite
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- A new lighter copolyester thermoplastic has very high heat deflection and contains 30% glass fibers. This plastic is expected to show lower warpage and higher gloss than previously used plastics. Applications would include automobile fenders and distributor housings.
- This paper discusses the problems inherent in controlling mine roofs that are composed of shale with plates of clay dispersed throughout. Waterproofing appears to be the answer to roof control but is also uneconomical. Both waterproofing and humidity control are discussed.
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A mathematical derivation for the determination of buckling stresses in cylinders exposed to uniform external pressure is given and numerical results are evaluated. The results agreed closely with those obtained by Flugge and also those obtained by Danielson and Simmonds.
- A11 Anonymous, "Detroit Puts Materials on a Diet," *Machine Design*, Vol. 49, No. 26, October, 1977, pp. 179-180. Plastics
This article discusses the increased use of plastics in the automotive industry. Future use and problems of finishing plastic body components are discussed. Improvement of existing metal fabrication techniques and development are treated briefly.
- A12 Amoco Chemicals Corporation, Trade Brochure Composite
A description of Torlon composite materials. Physical and mechanical properties are given for the different types of Torlon composite. Engineering applications as well as chemical resistance are also discussed.
- B1 Batkov, B. I., "Strength of Telescopic Hydraulic Jacks," *Russian Engineering Journal*, Vol. 53, No. 3, 1973, pp. 110-112. Buckling Cylinders

BIBLIOGRAPHY (Cont.)

- 100
- B1 (Cont.) An analytical method for analysis of telescoping hydraulic jacks is presented and demonstrated. *Key Words*
- B2 Broutman, L. J., **Composite Materials**, Academic Press Inc., New York, 1974. *Composites*
Buckling
Analysis of composite structures including buckling, shear, and stress-strain relations. Analysis is given for beams, columns, shells, and various other common geometrical shapes.
- B3 Beach, N. E., **Plastic Laminate Materials—Their Properties and Usage**, Foster Publishing Co., Long Beach, California, 1967. *Physical Properties*
Plastics
Composites
A listing of many reinforced plastic materials giving their composition and physical properties. Specific uses for laminates discussed range from electrical services to strong structural composites.
- B4 Bureau of Standards, "Abstracts of Government-Industry Workshop on Alternatives for Cadmium Electroplating in Metal Finishes, October 4-6, 1977." *Corrosion*
Approximately 30 abstracts of presentations given at the government-industry workshop on alternatives for cadmium electroplating. Topics discussed include ion implantation, electroplating, ion vapor depositing of aluminum coatings, solid film lubricant, and many others.
- B5 Brush Wellman Inc., "Machining Beryllium Copper," Trade Brochure, October, 1976. *Materials*
Beryllium
Machining
Pamphlet covers all aspects of machining beryllium including tools and coolant, turning, drilling, topping, reaming, and milling. Physical properties and chemical compositions of various beryllium alloys are also given.

BIBLIOGRAPHY (Cont.)

- 101
- B6 Babcock and Wilcox, "B & W Stroyloy 5C and 2A," Bulletin T-452.
- This bulletin describes different aspects of Stroyloy 5C and 2A. Both are new high strength and low alloy steel developed by Babcock and Wilcox. Capabilities of the alloys are discussed as well as availability, manufacturing and welding techniques. The chemical composition of both alloys is presented and tensile strength for selected specimens are presented in tabular form.
- Key Words*
Steel
- B7 Blake, Alexander, "Collapse of Out-of-Round Cylinders," *Machine Design*, Vol. 46, No. 19, August, 1974, pp. 129-130.
- A method for calculating the collapse strength of cylinders exposed to external pressure is presented. Thick and thin-walled cylinders failure is based on different criteria. Determination of the type of failure to be expected can be determined by use of the thinness factor. Methods for estimation of this thinness factor are given.
- Cylinders
- B8 Bacon, C. E., "Appliances and Equipment," *Composite Materials*, Vol. 3, by B. R. Noton (ed.), Academic Press Inc., New York, 1974, pp. 404-405.
- These two pages present a table listing mechanical and physical properties of various reinforced plastics, unreinforced plastic, and metals. Various properties listed include tensile and compressive strengths, flexure modulus, chemical resistance, and many others.
- Composites
Physical Properties
- C1 Cross, S. L., "Graphite Composite Materials Application in Aircraft Structures, SAE 750533.
- Basic physical properties and fabrication techniques for graphite composites are presented.
- Composite
Graphite
Physical Properties

BIBLIOGRAPHY (Cont.)

- 102
- C2 Corwine, J. W., "Review of Roof Control Technology Research," *Mining Congress Journal*, Vol. 62, No. 1, January, 1976, p. 25.
- Key Words*
Mining
- This report presents the basic principles of mine roof control. Causes of roof collapse are presented and a new research program is discussed.
- C3 Chang, George C. and Hirschfield, "For the Latest in Energy Storage, Try the Flywheel!" *Mechanical Engineering*, Vol. 100, No. 2, February, 1978, pp. 38-45.
- Composite
Physical Properties
- This article describes energy storage by use of flywheels housed in evacuated housings. Application to automotive industry is discussed in detail. A large part of the article is devoted to optimum design criteria. Materials that look promising in flywheel design are also treated. A table of various physical properties for composite materials is listed.
- C4 Commercial, Hy-Jax Trade Brochure.
- Cylinders
- A description of the Hy-Jax line of hydraulic mine props manufactured by Commercial. Specifications for various lengths and load capacities of props are given. Maintenance tips are also discussed.
- C5 Cloud, P. J. and B. Arkles, "Environmental, Thermal and Flammability Resistance of Glass Fortified Thermoplastics for Business Machine, Appliance and Tool Housings," 26th Annual Technical Conference, Reinforced Plastics/Composite Institute of the Society of the Plastic Industry, 1971.
- Glass
Corrosion
- A study of the corrosion resistance of glass fibers incorporated into many different resin matrices. Samples were subjected to acids, alkalines, and organic solvents. The effect of exposures to high temperature and flame resistance was also studied.

BIBLIOGRAPHY (Cont.)

Key Words

- D1 Dinda, S., "High-Strength Steel for Cost Effective Weight Reduction," SAE 760207.
- Economic justification for using high-strength steel and resulting fabrication problems as applied to automobile construction is discussed in detail.
- D2 Dreger, D. R., "Structural SMC Aims at Automotive Market," *Machine Design*, Vol. 49, No. 28, December, 1977.
- Report describes new higher strength sheet molded compounds with much higher glass fiber content. These sheet molded compounds can be formed in many more geometric shapes than earlier high-fiber content composites.
- D3 Dym, C. L., "On the Buckling of Cylinders in Axial Compression," *Journal of Applied Mechanics*, Vol. 40, No. 2, Transactions of ASME, Vol. 95, Series E, June, 1973, pp. 565-568.
- This paper presents buckling results for cylinders in axial compression based upon the shell equations of Hoiter and Budiansky. The limiting results for an infinitely long cylinder are given and numerical results are compared with the results obtained by Flugge and Donnell.
- D4 Donnell, L. H., "Analysis of Buckling Strength of Piston Rod," Internal Report to Benton Harbour Eng., Division of Koehring, 1958.
- A method is presented for determination of maximum allowable loads for hydraulic cylinders. The buckling of the rod and hoop stress in the cylinder are both dealt with. A method is outlined for determination of the cylinder pressure at which rod deformation begins.

Steel
Fabrication

Glass

Buckling

Cylinders

BIBLIOGRAPHY (Cont.)

- 104
- D5 Dixon Industries Corporation, Trade Brochures, 1976. *Key Words*
Plastics
Composites
- These trade brochures give approximate physical and mechanical properties for a number of injection molded thermoplastics and composites. Electrical and corrosive resistant properties are also presented. The working pressure and other properties of Dixon Industries line of Fiberglass pipe are also discussed.
- D6 Dym, C. L., "On Approximation of the Buckling Stresses of Axially Compressed Cylinders," *Journal of Applied Mechanics*, Vol. 41, No. 1, Transactions of ASME, Series E, March, 1974, pp. 163-167. *Buckling*
- An eighth-order equation for the buckling of cylinders subjected to axial compression is derived and used to approximate buckling stresses in a circular cylinder. The solution is compared to solutions obtained from various other types of analysis and the approximation is evaluated.
- E1 Enerpac, Catalog E-309, 1977 E. *Cylinders*
- This catalog gives a listing of many hydraulic tools manufactured and sold by Enerpac located in Butler, Wisconsin. A general description as well as specifications is given for hydraulic devices such as hydraulic cylinders, pumps, pump and cylinder sets, hoses, valves, and many others.
- E2 Enerpac, Idea Book I-1001, 1976. *Cylinders*
- This brochure describes various industrial applications of Enerpac hydraulic tools. A wide spectrum of industry is covered including metal fabrication, railroad applications, and manufacturing of mining equipment.
- E3 Enerpac, U.S. Net Price List No. SU-8039, August 22, 1977. *Cylinders*

BIBLIOGRAPHY (Cont.)

Key Words

- E3 (cont.) This is a price list of various devices sold by Enerpac of Butler, Wisconsin. Prices and shipping weights are listed alphabetically according to part numbers.
- G1 Gaylord, M. W., **Reinforced Plastics—Theory and Practice**, Cahners Publishing Co., Boston, Mass., 1974.
- A presentation dealing with all aspects of composite materials from fabrication to training of shop personnel.
- G2 General Electric Co., **Lexan Resin—New Name on the Bottom Line**, Trade Brochure.
- A semi-technical paper describing the advantages and disadvantages of Lexan resin application. Cost of Lexan application is discussed along with ease of production. Various applications are shown and discussed economically.
- G3 General Electric Company, **Lexan Polycarbonate Resin — Design**, Trade Brochure.
- Technical paper presenting the physical properties of Lexan resins. Design of various components is discussed in detail. A problem analysis section is also included.
- G4 Gullick Dobson International Ltd., **Trade Brochures**, September, 1977.
- Descriptions of the many different types of roof support systems manufactured by Gullick Dobson Ltd. Specifications are given for both hand set and multi-cylinder models. A description of hydraulic power systems is also given.

Fabrication
Composites

Plastic

Plastic

Roof Support

BIBLIOGRAPHY (Cont.)

- H1 Hunter, Donald W., "Trends in Coal Mining Systems—Conventional, Continuous, Longwall and Shortwall," *Mining Congress Journal*, Vol. 62, No. 7, July, 1976, pp. 56-60.
- Future of coal mining in the U.S. is discussed and evaluated. Various mining techniques such as longwall, shortwall, and continuous are also evaluated. A small section of the report is devoted to discussion of new equipment such as roof supports.
- H2 Huwood-Irwin Co., Irwin, Pennsylvania, Longwall Systems to Handle the Unique Conditions of Any Mine, Trade Brochure.
- This brochure gives the capabilities and specification of various longwall roof support systems manufactured by Huwood-Irwin Co., Irwin, Pennsylvania.
- H3 Hemscheidt America, Trade Brochure.
- This brochure provides a general description of the various roof support systems offered by the Hemscheidt America Corporation.
- H4 Hemscheidt America, Hemscheidt Support System, Trade Brochure.
- This brochure lists specifications for various shield type props, single props, and hydraulic power packs manufactured by Hemscheidt America Corporation.
- Key Words*
- Mining
Roof Support
- Mining
Roof Support
- Mining
Roof Support
- Mining
Roof Support

BIBLIOGRAPHY (Cont.)

- Key Words*
- H5 Hart, G. L. and Geoffrey Pritchard, "Role of the Interface in the Corrosion of Graphite Fiber Composites," 29th Annual Technical Conference, Reinforced Plastic/Composite Institute of the Society of Plastics Industry, 1974. Corrosion
- It is shown that graphite composites can withstand a variety of acids, bases, and organic solvents. The adhesion between fiber and matrix and its effects on corrosion properties are also discussed.
- I1 Isham, A. B., "Glass Fiber Reinforced Elastomers for Automobile Applications—A Comparison of RIM Urethanes and Alternative Material Systems," SAE 76033. Composites
- Physical properties of flexible materials reinforced by glass fibers are determined by experiment and presented in tabular form.
- I2 Impact Plastics Inc., Trade Brochure Plastics
- These pamphlets provide basic information concerning the implementation of Impact Plastic's line of plastics in design of mechanical components. The various physical properties of these plastics are presented in tabular form.
- J1 Jackson, Daniel, "York Canyon Researches in Depth for Optimum in Mining Preparation," *Coal Age*, May, 1976. Mining
- A report on experimental work being carried out in Kaiser Steel's York Canyon mine. Different mining methods and problems of roof control are discussed.
- K1 Kary, J. J., "Weight Reduction of Automobile Bumper Systems," SAE 760012 Steel
- Methods for reducing weight of vehicle bumpers are presented.

BIBLIOGRAPHY (Cont.)

- 108
- K2 Kusy, P. F., "Plastic Materials Selection Guide," SAE 760663.
- Guidelines for determining if a plastic material should be considered and approximate ranges of physical properties of plastics are presented.
- Key Words
Plastics
Physical Properties
- K3 Khoddam-Mohammadi, H. and R. K. King, "Composite Materials—A Design Alternative," *The BFPR Journal*, 1979, Vol. 12, pp. 11-14.
- This paper is a discussion of the advantages and disadvantages of composite materials. Many properties of composites are discussed including fatigue strength, strength-to-weight ratios and cost of the advanced composites. Also included is a section on manufacturing techniques.
- Composites
Cost
- K4 Korb, L. J., "Engineering Applications of Composites," in B. R. Noton (ed.), *Composite Materials*, Academic Press, New York, 1974, Vol. 3, pp. 70-73.
- This table presents a comparison of S-glass, boron, graphite and organic fibers. Tables listing both relative and specific properties, and the general cost and availability of each type of composite are presented.
- Composites
Physical Properties
- M1 Meir, J. H., "Buckling of Uniform and Stepped Columns—I," *Product Engineering*, Vol. 20, No. 10, October, 1949, pp. 119-123.
- Analytical method of predicting critical loading and buckling behavior of columns is presented in detail. This procedure gives more accuracy than formulas given in handbooks of that date. Curves are presented to aid in determining buckling loads for various geometric shapes.
- Buckling
Cylinders
- M2 Mills, B. D., "The Fluid Column," *American Journal of Physics*, Vol. 28, 1960, pp. 353-356.
- Cylinders
Buckling

BIBLIOGRAPHY (Cont.)

Key Words

M2 The buckling behavior of pressurized cylinders without axial loading is
(cont.) presented including a mathematical analysis of this system.

M3 Molded Fiberglas Companies, Story of.

Past accomplishments of the molded Fiberglas companies are noted as well as plans for the future. Present manufacturing techniques are presented and displayed pictorially. Boating, farm machinery, and construction applications are presented along with a discussion of various testing techniques.

Glass
Fabrication
Composite

M4 Meir, J. H., "Buckling of Uniform and Stepped Columns—II," **Product Engineering**, Vol. 20, No. 11, November, 1949, pp. 116-118.

This study of buckling of columns deals with such parameters as factor of safety, initial eccentricity, and fatigue stress in determining the critical load. A study is also made of the buckling of cylinders and piston rods using the method developed for stepped columns. Example problems illustrate the techniques described.

Buckling
Cylinders

M5 Marrel Hydro-Mines, Trade Brochures, June, 1974.

A description of various roof support mechanisms manufactured by Marrel Hydro-Mines. All props discussed are of the multi-cylinder hydraulic type.

Roof Support

M6 Minex, Trade Brochures

A description of the wide variety of mining equipment produced by Minex Industries. Specifications are given for a wide variety of equipment such as self-advancing hydraulic roof support systems, shearers, and roof bolts.

Roof Support

BIBLIOGRAPHY (Cont.)

- 110
- M7 Morrison Molded Fiberglass Co., Trade Brochures.
- A complete description of glass-reinforced materials produced by Morrison Fiberglass Co. An engineering manual is included which presents physical properties, tolerances and availability of various beams and columns. Corrosion resistance and other design details are also dealt with.
- Key Words
Composites
Glass
Corrosion
- N1 Norman, J. C., "Damage Resistance of High Modulus Aramid Fiber Composites in Aircraft Applications," SAE 750532.
- The resistance of graphite, glass, and aramid fiber composites to impact and fracture are compared. Physical properties of all three are presented. Hybrid fibers are also discussed.
- Physical Properties
Aramid Fiber
Composites
- O1 Office of Technology, "Assessment, Material Substitution: Availability, Energy, and Environmental Factors," Conference, September 27-28, 1977.
- The conference proceedings include substitution of ceramics for high-temperature alloys, new steel alloys, and many others.
- Materials
- P1 Polentz, L. M., "Hydraulic Cylinders—New Performance for Old Reliable," *Machine Design*, Vol. 46, No. 10, April, 1974, pp. 131-137.
- This article discusses the advancements made in hydraulic cylinder design in the past few years. The problem of proper cylinder mounting and a list of do's and don'ts applying to their problem are presented. Determining proper size of hydraulic cylinders and selection of proper cylinder sealing are also discussed.
- Cylinders
- P2 Polygon Company, Trade Brochures.
- Plastics
Composites

BIBLIOGRAPHY (Cont.)

Key Words

- P2 (cont.) These brochures give listings of physical properties and specifications for Teflon and Fiberglass products sold by the Polygon Company of Walkerton, Indiana including Teflon bearings and woven Fiberglass tubes.

- R1 Ralston, J. A., *Ralston's Index—S.P.I.R.P./C., 1970-1976*, Robert Wilson Publishing Co., Alexandria, Virginia, 1977.

Composites

An index to publications pertinent to composite materials and designs incorporating these materials.

- R2 Rehfield, L. W., "Design of Stiffened Cylinders to Resist Axial Compression," *Journal of Spacecraft and Rockets*, Vol. 10, No. 5, May, 1973, pp. 346-349.

Cylinders

A design procedure is presented for compressed cylindrical shells stiffened by stringers and rings. A new method for prediction of proper sizing and spacing of rings is outlined and illustrated. Emphasis is placed on attainment of a minimum weight design within the limits of the design parameters.

- R3 Reese, Charles D. and Charles W. Bert, "Buckling of Orthotropic Sandwich Cylinders Under Axial Compression and Bending," *Journal of Aircraft*, Vol. 11, No. 4, April, 1974.

Buckling

The buckling analysis in this report is for finite-length sandwich cylinders clamped at both edges and loaded in pure bending, axial compression, or a combination of these loadings. Love's first-approximation shell theory is used and a Rayleigh-Ritz solution is obtained. A comparison of predicted and experimental results is given.

- R4 Rao, S. N., "Development of Lightweight Hydraulic Prop for Underground Coal Mining Industry," Master's Thesis, Oklahoma State University, Stillwater, Oklahoma, 1974.

Aluminum Alloy
Cylinder

BIBLIOGRAPHY (Cont.)

- R4 (cont.) This thesis gives a description of props used in the mining industry. A mathematical analysis of the various stresses in loaded props and the results of the study are presented in tabular form. A program for computer analysis of mining props is listed.

Key Words

- R5 Renton, W. J., **Hybrid and Select Metal Matrix Composites: A State of the Art Review**, American Institute of Aeronautics and Astronautics, New York, 1977.

Composites

Many facets of composite materials design are discussed in detail. A short section is devoted to a discussion of various different types of composite materials. Analytical and design methods applicable to hybrid composites are also presented. Testing methods and mechanical and physical properties of both metallic and non-metallic matrix composites are listed for many different composite materials. Separate sections are devoted to problems involved with the design of boron-aluminum and graphite-aluminum composites.

- R6 Roller Bearing Company of America, Trade Brochure.

Bearings

A description of bearings and keyless connecting shafts manufactured by RBC. Bearings permit precise adjustment of interference fit and are chemically inert.

- R7 Rosata, D. V., "Cost Comparison of Plastic Parts for the Transportation Industry Based on Different Manufacturing Processes," 28th Annual Technical Conference, Reinforced Plastics/Composite Institute of the Society of Plastics Industry, 1973.

Cost

A summary of different manufacturing techniques used in the construction of composite structures. The advantages and disadvantages of each type of fabrication are discussed. A method for estimating costs of composite structures after manufacturing is presented. The total manufacturing cost is broken down into several categories such as direct labor and overhead. The amount of the manufacturing costs going into each of these categories is estimated.

BIBLIOGRAPHY (Cont.)

Key Words

- 113
- S1 Sperring, R. F., "Glass Materials: 1974-1980," SAE 750344.
Report deals with the increasing importance of glass production. The economics and problems of producing glass are also dealt with.
Glass
- S2 Schwartz, R. T., **Fundamental Aspects of Fiber Reinforced Plastic Composites**, Interscience Publishers, New York, 1968.
Book presents mathematical analysis of composite structures and ways of bonding fiber and matrix.
Composites
Design Criteria
- T1 Thomson, W. T., "Critical Load of Columns of Varying Cross-Section," **Journal of Applied Mechanics**, Transactions of the ASME, Vol. 72, June, 1950, p. 132.
A method is developed for determining the critical load of a pin-ended column of several different cross-sections and lengths.
Buckling
Cylinders
- U1 Union Carbide Corporation, Udel Polysulfane for Injection Molding and Extrusion, Trade Brochure, 1976.
Technical paper covering the basic aspects of the family of Udel polysulfanes. The plastics can be formulated in virtually any color and can be reinforced with glass filler. The plastic has good thermal qualities and has applications ranging from medical instruments to astronauts' face mask shields. The plastic is good for any application requiring a relatively low tensile strength.
Plastic
- U2 Unistrut Corporation, Wayne, Michigan, Designing with Telespar, Trade Brochure, 1977.
This brochure explains the manufacturing techniques used in making Telespar square tubing. Various applications of this tubing are illustrated and design data are given in tabular form.
Steel

BIBLIOGRAPHY (Cont.)

- W1 Wimber, R. T., "A Preliminary Materials Selection Guide," SAE 760139.
General ranges of physical properties are given for virtually all common structural materials.
- W2 Woodruff, S. D., *Methods of Working Coal and Metal Mines*, Pergamon Press Ltd., Oxford, England, 1966.
A detailed look into the processes of underground mining. Many topics are covered including roof support.
- W3 Whitney, J. M. and Sun Chang-Tun, "Buckling of Composite Cylindrical Characterization Specimens," *Journal of Composite Materials*, Vol. 9, April, 1975, pp. 138-148.
The determination of critical buckling loads for unidirectional and laminated cylinders subjected to torsion and axial compression.
- Y1 Yeaple, F., "Tie Rod Stress in Cylinders Can Be Accurately Calculated," *Product Engineering*, Vol. 44, No. 12, December, 1973, pp. 40-43.
Report presents a method for analysing stresses present in pressurized hydraulic cylinders. Equations are derived to determine tension, expansion, compression, torque, and stress for the various parts of the cylinder.
- Z1 Zweben, C. and P. G. Riewald, "Kevlar-49 Hybrid Composites for Commercial and Aerospace Applications," 30th Annual Technical Conference, Reinforced Plastics/Composite Institute of the Society of Plastics Industry, 1975.
A listing of mechanical and physical properties for graphite-Kevlar-49 hybrid composites of varying compositions. The cost per pound of raw materials is presented in tabular form for varying graphite to Kevlar-49 ratios.

Key Words

Physical Properties

Mining
Cylinders

Cylinders
Buckling

Cylinders

Physical Properties
Cost

CROSS REFERENCE

Aluminum Alloy -- A2, A5, R4

Aramid Fiber Composites -- N1

Beryllium -- B5

Bearings -- R6

Buckling -- A10, B1, B2, D6, M1, M2, M4, R3, T1, W3

Composites -- A3, A6, A8, A12, B2, B3, B8, C1, C3, D5, G1, I1, K3, K4, M3, M7, N1, P2, R1, R5, S2

Corrosion -- B4, C5, H5, M7

Cost -- K3, R7, Z1

Cylinders -- A5, B1, B7, C4, D4, E1, E2, E3, M1, M2, M4, P1, R2, R4, T1, W2, W3, Y1

Design -- S2

Fabrication -- A3, D1, G1, M3

Glass -- C5, D2, M3, M7, N1, S1

Graphite -- A8, C1

Machining -- B5

Jacks -- A9

Materials -- B5, O1

Mining -- A7, C2, H1, H2, H3, H4, J1, W2

Physical Properties -- A3, A4, B3, B8, C1, C3, K2, K4, N1, W1, Z1

Plastics -- A11, B3, D5, G2, G3, I2, K2, P2, U1

Roof Support -- A7, G4, H1, H2, H3, H4, M5, M6

Safety -- A9

Steel -- A1, B6, D1, K1, U2

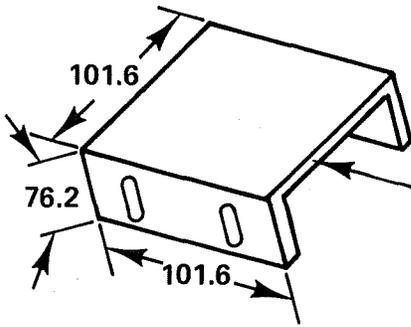
Thermoplastics -- A6

Titanium -- A4

APPENDIX B
SIZE AND WEIGHT CALCULATIONS

MECHANICAL WEDGE

Part No. 1

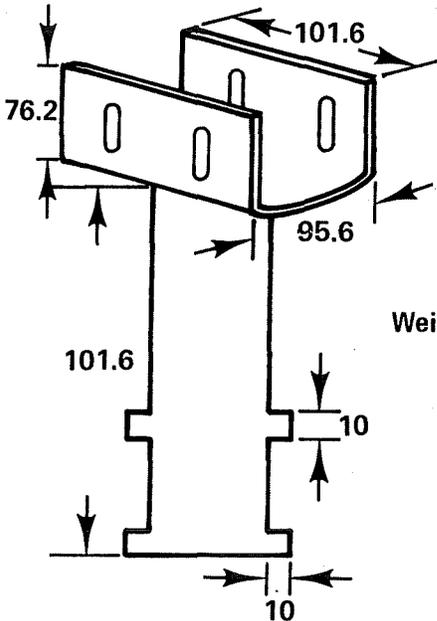


Top Plate, Aluminum Alloy
Density = 2801 kg/m³

Thickness is 3 mm for entire component.

$$\text{Weight} = [(0.106)^2 (0.003) + 2(0.0762) (0.1016) (0.003)] (2801) = 0.1 \text{ kg}$$

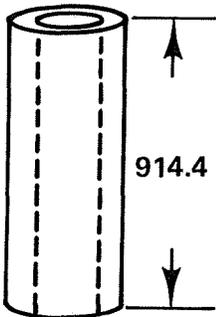
Part No. 2



Aluminum Alloy
Inside diameter = 114.5 mm
Outside diameter = 127 mm
Thickness of plate = 3 mm

$$\begin{aligned} \text{Weight} = & \pi/4 \left[\left((0.127)^2 - (0.1145)^2 \right) (0.1016) + 2 \left((0.147)^2 - (0.127)^2 \right) \right. \\ & \left. (0.01) \right] (2801) + \left[2(0.1016) (0.0762) + (0.1016) (0.095) \right] \\ & (2801) (0.003) = 1.13 \text{ kg} \end{aligned}$$

Part No. 3

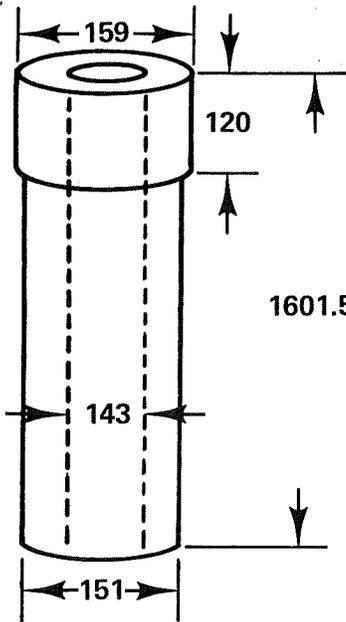


Aluminum Alloy
Outside diameter = 143 mm
Inside diameter = 127 mm

$$\text{Weight} = (0.9144) \pi/4 [(0.143)^2 - (0.127)^2] (2801) = 8.69 \text{ kg}$$

MECHANICAL WEDGE

Part No. 4



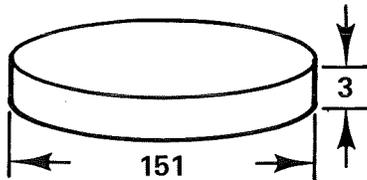
Large Tube, Aluminum Alloy
Density = 2801 kg/m³

Outside diameter = 159 mm for top 120 mm
151 mm for remainder of tube
Inside diameter = 143 mm

$$\text{Weight} = \pi/4 \left[\left((.159)^2 - (.143)^2 \right) (.120) + \left((.151)^2 - (.143)^2 \right) (1.6015 - .120) \right] (2801) = 8.94 \text{ kg}$$

Part No. 5

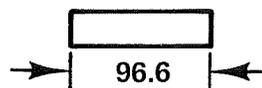
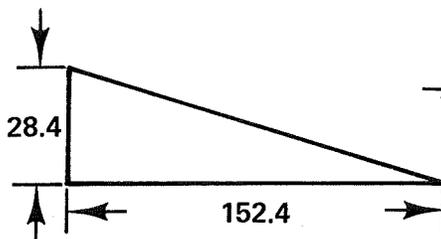
Bottom Plate, Aluminum Alloy



$$\text{Weight} = \pi/4 (.151)^2 (.003) (2801) = .15 \text{ kg}$$

Part No. 6

Wedges, Wood Density = 600 kg/m³



Two wedges required

$$\text{Weight} = (0.0254) (0.1524) (0.0966) (600) = .22 \text{ kg}$$

MECHANICAL SCREW

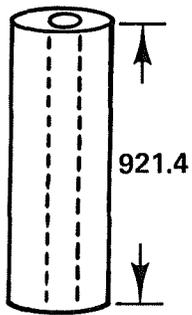
Buckling load for screw (2438.4 mm long, 114.3 mm outside diameter, 104.78 mm inside diameter)

$$F_e = \frac{\pi^2 E I}{L^2}$$

$$E = 2.07 \times 10^{11} \text{ Pa}$$

$$F_e = \frac{(3.1416)^3 (2.07 \times 10^{11}) [(0.1143)^4 - (0.10478)^4]}{(2.4384)^2 (64)} = 8.46 \times 10^5 \text{ N}$$

$7.85 \times 10^5 \text{ N}$ is required.



Top Tube,

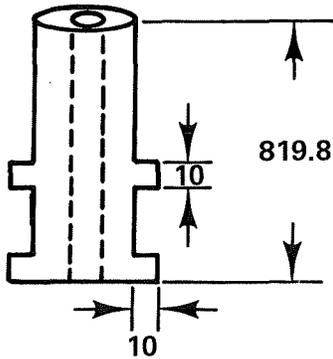
Steel Density = 8016 kg/mm^3

Part No. 1

Outside diameter = 133.35 mm

Inside diameter = 127.0 mm

$$\text{Weight} = \pi/4 \left((0.13335)^2 - (0.127)^2 \right) (0.9214) (8016) = 9.59 \text{ kg}$$



Bottom Tube,

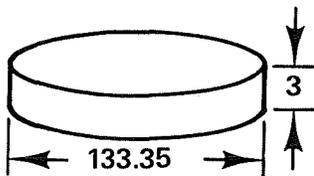
Steel

Part No. 2

Outside diameter = 133.35 mm

Inside diameter = 127.0 mm

$$\text{Weight} = \pi/4 \left[\left((0.13335)^2 - (0.127)^2 \right) (819.8) + 2 \left((0.15335)^2 - (0.13335)^2 \right) (0.01) \right] \times 8016 \text{ kg/m}^3 = 9.25 \text{ kg}$$



Top Plate,

Steel

Part No. 3

Thickness = 3 mm

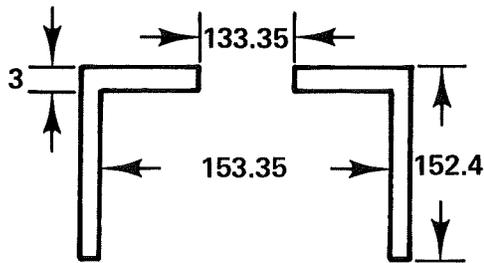
Diameter = 133.35 mm

$$\text{Weight} = \pi/4 (0.13335)^2 (0.003) (8016) = .34 \text{ kg}$$

MECHANICAL SCREW

Bottom Support

Part No. 4



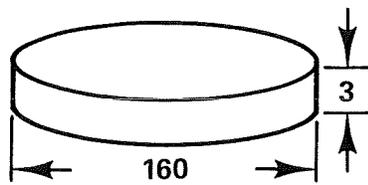
Inside diameter = 153.35 mm
Outside diameter = 160 mm

$$\text{Weight} = \pi/4 [(0.160)^2 - (0.15335)^2] (.1524) + (0.160)^2 - (0.13335) (.003) (8016) = 2.73 \text{ kg}$$

Part No. 5

Bottom Plate,

Steel

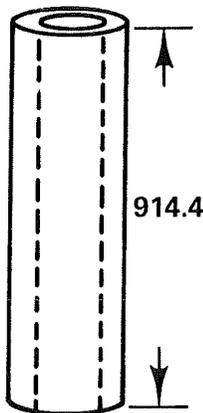


$$\text{Weight} = \pi/4 (.16)^2 (.003) (8016) = 0.48 \text{ kg}$$

Part No. 6

Screw,

Steel



Length = 914.4 mm
Outside diameter = 114.3 mm
Inside diameter = 104.77 mm

$$\text{Weight of cylinder} = \pi/4 [(.1143)^2 - (.10477)^2] (.9144) (8016) = 12.02 \text{ kg}$$

For Acme thread screw of above dimensions:

$$\text{Pitch} = 12.7 \text{ mm} \quad \text{Cross sectional area of 1 thread} \approx \left(\frac{\text{pitch}}{2} \right)^2$$

Weight of threads = (Number of threads) (π) (Mean thread diameter) x (Area of one thread)

$$\text{Number of threads} = 914.4 \times 1/12.7 = 72$$

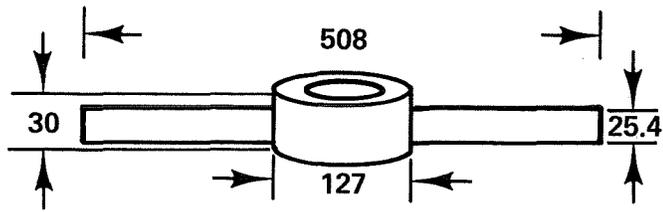
$$\text{Weight of threads} = 72\pi (.1175) (.0127/2)^2 (8016) = 8.6 \text{ kg}$$

MECHANICAL SCREW

Handle,

Steel

Part No. 7



Outside diameter = 25.4 mm
Inside diameter = 15.4 mm

$$\text{Weight} = 8016 \pi/4 \left[(.381) \left((.0254)^2 - (.0154)^2 \right) + \left((.127)^2 - (.1143)^2 \right) (.03) \right] = 1.56 \text{ kg}$$

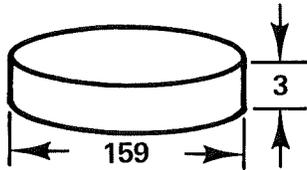
TOTAL PROP WEIGHT:

$$9.59 + 9.25 + 0.34 + 2.73 + 0.48 + 8.6 + 1.56 = 32.55 \text{ kg}$$

Top Plate,

Al-Alloy

Density = 2801 kg/m³

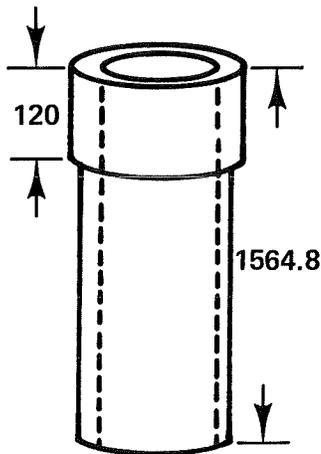


$$\text{Weight} = \pi/4 (.159)^2 (.003) (2801) = 0.17 \text{ kg}$$

Part No. 2

Top Tube,

Al-Alloy



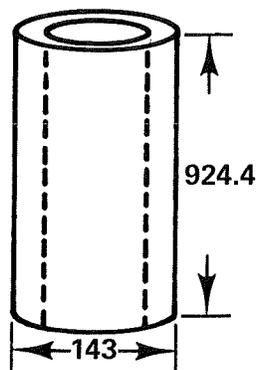
Outside diameter = 159 mm for top 120 mm of tube
151 mm for bottom 1555.8 mm
Inside diameter is 143 mm throughout

$$\text{Weight} = \pi/4 \left[\left((0.159)^2 - (0.143)^2 \right) (0.120) + \left((0.151)^2 - (0.143)^2 \right) (1.4448) \right] (2801) = 8.75 \text{ kg}$$

Part No. 3

Bottom Tube,

Al-Alloy

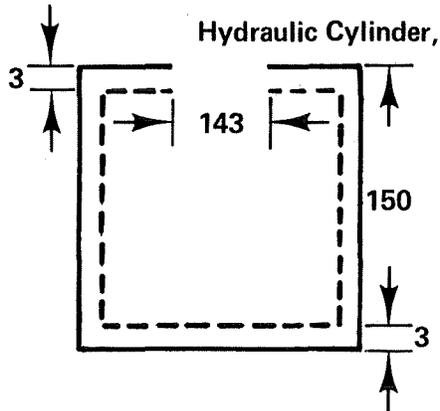


Inside diameter = 127 mm
Outside diameter = 143 mm

$$\text{Weight} = \pi/4 \left[(0.143)^2 - (0.127)^2 \right] (0.9244) (2801) = 8.78 \text{ kg}$$

HYDRO-MECHANICAL (External Power, Internal Reservoir)

Part No. 4



Hydraulic Cylinder,

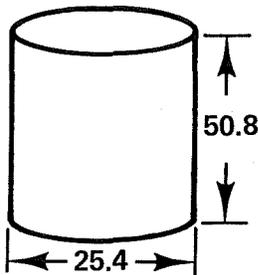
Al-Alloy

Outside diameter = 168.6 mm
Inside diameter = 155.0 mm

$$\text{Weight} = \pi/4 \left[\left((0.1686)^2 - (0.155)^2 \right) (0.15) + (0.003) \left((0.1686)^2 + (0.1686)^2 - (0.143)^2 \right) \right] (2801) = 1.69 \text{ kg}$$

Part No. 5

Pump (Gear Pump, 0.25 gpm)



Assume solid steel for estimation purposes.
Density = 8016 kg/m³

$$\text{Weight} = \pi/4 (0.0254)^2 (0.0508) (8016) = 0.21 \text{ kg}$$

Part No. 6

Reservoir, Al-Alloy
(must contain .0016 m³ fluid)

$$x^3 = .0016$$

$$x = .1177 \text{ m}$$

Wall thickness = 2 mm

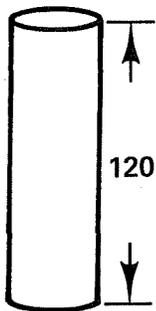
If reservoir is mounted on cylinder wall, only 5 sides are needed.

$$\text{Weight} = 5(0.1177)^2 (.002) (2801) = .4 \text{ kg}$$

Part No. 7

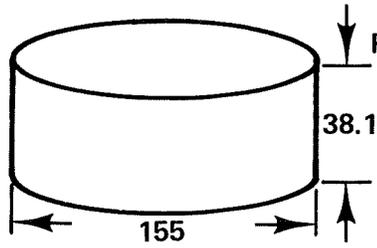
Valve Release Rod,

Al-Alloy



Outside diameter = 12.7 mm
Inside diameter = 7.6 mm

$$\text{Weight} = \pi/4 \left[(0.127)^2 - (.0076)^2 \right] (.12) (2801) = .03 \text{ kg}$$



Piston Head, Al-Alloy

38.1

155

$$\text{Weight} = \pi/4 (0.0381) (0.155)^2 (2801) = 2.0 \text{ kg}$$

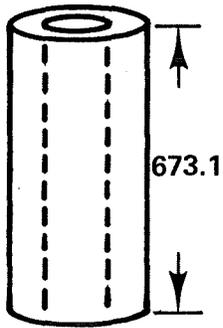
Fluid (Calculated on a basis of 152.4 mm stroke)

$$\text{Density} = 901.5 \text{ kg/m}^3$$

$$\text{Weight} = \pi/4 (0.155)^2 (.1524) (901.5) = 2.6 \text{ kg}$$

TOTAL PROP WEIGHT:

$$0.17 + 8.75 + 8.78 + 1.69 + 0.21 + 0.4 + 0.03 + 2.0 + 2.6 = 24.63 \text{ kg}$$

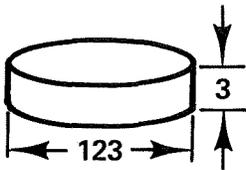


Rod, Al-Alloy

Outside diameter = 123 mm
Inside diameter = 105 mm

$$\text{Weight} = \pi/4 \left[(0.123)^2 - (0.105)^2 \right] (0.6731) (2801) = 6.08 \text{ kg}$$

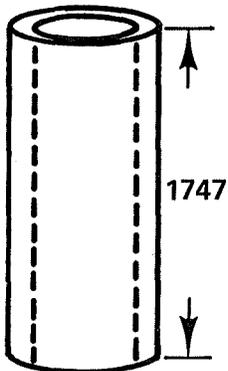
Part No. 2



Top Plate, Al-Alloy

$$\text{Weight} = \pi/4 (0.123)^2 (.003) (2801) = 0.1 \text{ kg}$$

Part No. 3

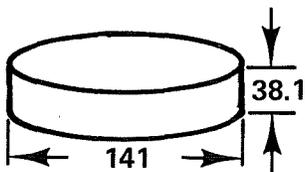


Cylinder, Al-Alloy

Outside diameter = 160 mm
Inside diameter = 141 mm

$$\text{Weight} = \pi/4 \left[(0.16)^2 - (0.141)^2 \right] (2801) (1.747) = 22.0 \text{ kg}$$

Part No. 4



Piston Head, Al-Alloy

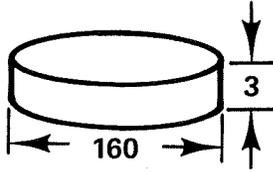
$$\text{Weight} = \pi/4 (0.141)^2 (0.0381) (2801) = 1.67 \text{ kg}$$

HYDRAULIC (External Power, Internal Reservoir)

Part No. 5

Bottom Plate,

Al-Alloy

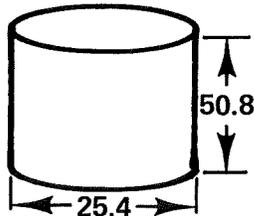


$$\text{Weight} = \pi/4 (0.16)^2 (.003) (2801) = 0.17 \text{ kg}$$

Part No. 6

Pump,

Steel



$$\text{Density} = 8016 \text{ kg/m}^3$$

$$\text{Weight} = \pi/4 (.0254)^2 (.0508) (8016) = 0.21 \text{ kg}$$

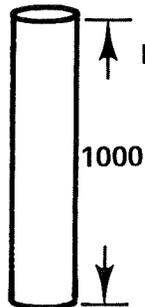
Reservoir, (must be capable of holding 0.0073 m³ of fluid)

Part No. 7

2 mm wall thickness Al-Alloy
5 sides needed

$$\text{Weight} = 5(0.1935)^2 (.002) (2801) = 1 \text{ kg}$$

Part No. 8



Pressure Release Rod,

Al-Alloy

Outside diameter = 12.7 mm
Inside diameter = 7.6 mm

$$\text{Weight} = \pi/4 \left[(0.0127)^2 - (0.0076)^2 \right] (1) (2801) = 0.23 \text{ kg}$$

Fluid

Stroke = 696.6 mm

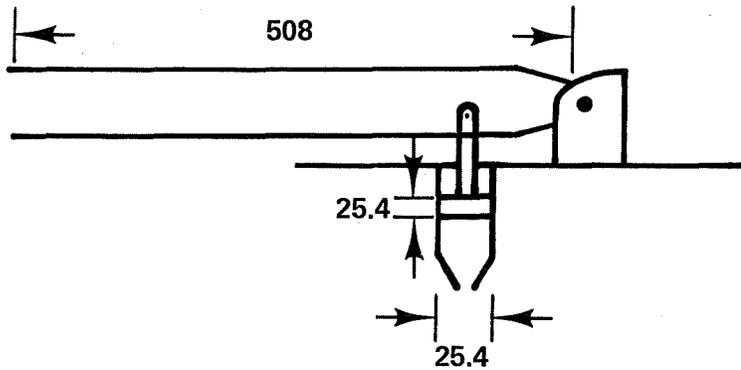
Part No. 9

$$\text{Weight} = \pi/4 (0.141)^2 (0.762) (901.5) = 10.73 \text{ kg}$$

TOTAL PROP WEIGHT:

$$6.08 + 0.1 + 22.0 + 1.67 + 0.17 + 0.21 + 1 + 0.23 + 10.73 = 42.19 \text{ kg}$$

INTERNAL POWER



Inside cylinder diameter = 25.4 mm
Outside cylinder diameter = 28.6 mm Steel

Outside rod diameter = 25.4
Inside rod diameter = 21.6 Al-Alloy

$$\text{Rod weight} = \pi/4 (.508) \left[(0.0254)^2 - (0.0216)^2 \right] (2801) = 0.2 \text{ kg}$$

$$\text{Cylinder weight} = \pi/4 \left[(0.0286)^2 - (0.0254)^2 \right] (.0381) (8016) = 0.04 \text{ kg}$$

Piston Weight, steel piston

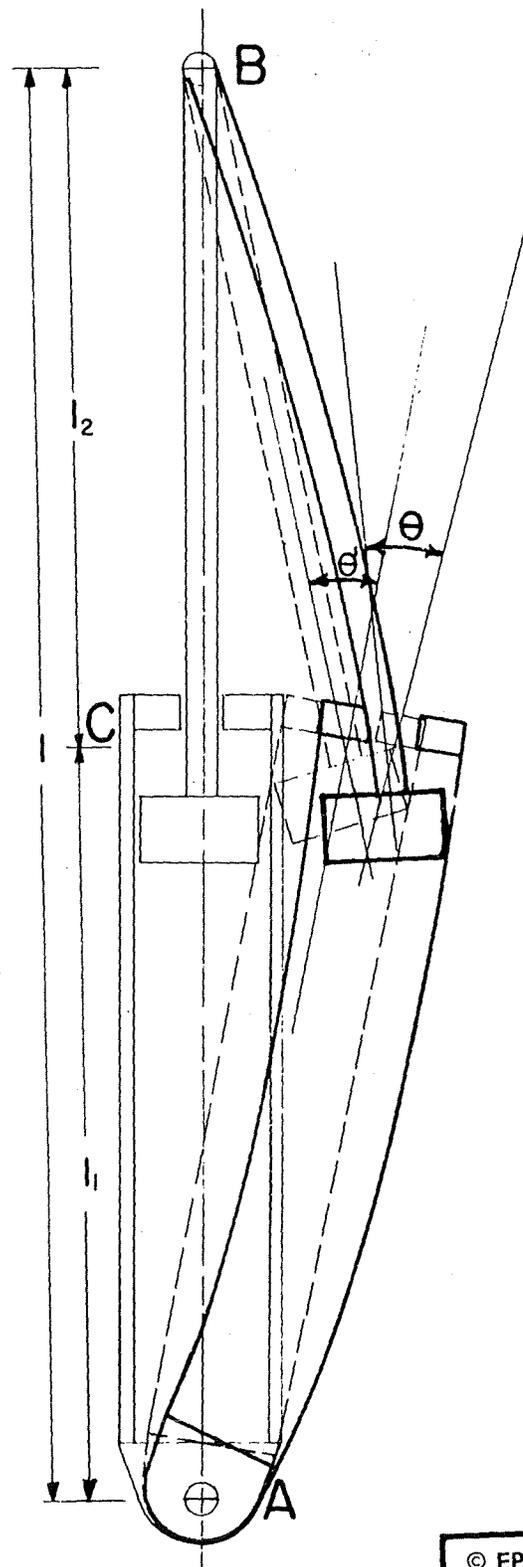
$$\text{Weight} = \pi/4 (0.0254)^2 (.0254) (8016) = 0.1 \text{ kg}$$

$$\text{Assembly weight} = 0.2 + 0.04 + 0.1 = 0.34 \text{ kg}$$

APPENDIX C
COMPUTER PROGRAM FOR CYLINDER ANALYSIS
(SACREG)

This brief description is included to enable readers to gain an appreciation of the joint-flexure theory of hydraulic cylinder structural analysis and the computer program for *Static Analysis of Cylinder, REGular* (SACREG). The program, which was used for all-hydraulic prop optimization was developed for a Technology Development Project, sponsored jointly by the U.S. Army Mobility Equipment Research and Development Command and industrial companies [16]. The computer analysis has been verified for deflection accuracy [17] and industrial users are in the process of load capacity verification [18].

The mathematical model for the cylinder relies upon continuous beam theory to establish the deflection curves for the cylinder and rod in the loaded condition. It differs significantly from earlier efforts in incorporating, at the beginning of the analysis, the effect of the clearances between the piston rod and the cylinder. Figure C-1 shows in light lines the configuration of a symmetrical (ideal) cylinder. Due to eccentric loading, constructional irregularities, self-weight, etc., the unloaded cylinder will assume the shape shown by the dotted lines. The center lines of the cylinder and rod (which may be a tube) intersect at a point away from the line joining the end-pins. Consequently, there exists an initial deflection and a "crookedness angle" at the piston rod-cylinder interface; even in the unloaded condition when external loads and moments are applied the cylinder and rod deflect still further, until the material builds up sufficient stresses to develop restraining



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FIGURE C-1

STRAIGHT AND FLEXED SHAPES OF HYDRAULIC CYLINDER

moments. Note that the points of contact of the piston head and rod on one hand, and the cylinder wall and gland on the other develop into areas of contact. In the mathematical model, average contact stresses at different areas of contact are calculated by using the moduli of elasticity of the softer material. Details of the model are provided in reference sixteen.

In the rod, at any cross-section, there will be compressive stress due to the end loading and bending stresses due to deflection, eccentric loading and end moments. In the cylinder, at any cross-section, there will be bending stress and hoop stress due to internal fluid pressure. It should be noted that the cylinder walls, in contra-distinction to a cylindrical pressure vessel, will not have longitudinal stresses caused by internal pressure. Also, *all* the bending moments at any section have to be restrained by stresses in cylinder wall, since the internal fluid cannot resist distortion.

The principal stresses, as well as the shear stress at any cross-section, can be calculated once the deflection curve of the cylinder and rod are determined. Since the equations for these curves are transcendental, an alternative solution scheme is used to establish the points of maximum deflection in the cylinder and the rod.

The actual cylinder loading is simulated on the computer by incrementing the load by small amounts, a process which is stopped when the principal stress in either the rod or the cylinder attains the prescribed limit. It is important to note that:

1. "Short", as well as "long" cylinders can be analyzed with equal

ease and accuracy since the failure criterion is one of overstressing.

2. Deflection increases gradually until the material is overstressed, i.e., there is no "buckling" in the classical meaning of the term.

Apart from calculating the critical load for a given configuration, the mathematical model has been computerized to

(i) calculate the load and stresses for a given operating pressure, and

(ii) calculate the length of the stop tube required to limit the crookedness angle and the forces on the bearings to prescribed values. These options would not be normally used in hydraulic prop design.

Even though the mathematical model can, in principle, be used to calculate any set of variables/parameters entering into the equations, so long as they equal the number of equations; the solution algorithm, and therefore the computer program, would have to be different for each set. Consequently, if a specific parameter, say cylinder wall thickness, is to be optimized, the computer program would have to be used iteratively and the performance index for optimization calculated separately after each stage.

APPENDIX D

USER'S GUIDE TO SACREG

The computer program for cylinder structural analysis, SACREG, is written in FORTRAN IV and requires approximately 90 kilo-bytes of storage on an IBM 370 digital computer. Input is in the form of 80-column cards and output is formulated for 132 character-width lines. Input information is categorized into a set of tables, which may run on to multiple cards. The program is set up for repetitive runs and data furnished for one run can be retained for the next by exercising the "KEEP" option in Table I. In the following pages, fields for the various quantities are indicated on a card by card basis. Input data must adhere strictly to these requirements as well as standard FORTRAN conventions. It is specially important that real numbers include the decimal point and, if scientific notation is used, the numbers be right-justified in the fields. The total number of cards for a problem depends on the number and arrangement of piston rings and load-bearing seals at the gland. The last card must always be blank, and there is no harm in including extra blank cards. Special care should be taken in assigning consistent units and numerical values to parameters. Thus, if the metre is the unit of length, and Newton is the unit of load, the tensile strength and Young's modulus should be in Pascals (N/m^2). If some numbers are very large or very small, the exponential form of notation may have to be used. The program operates in double precision but will accept input data in simple precision as well. For mine prop appraisal, the value of LPRTP in Table I should be always one.

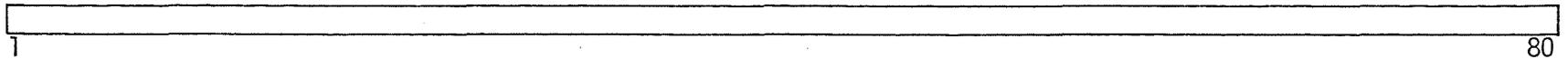
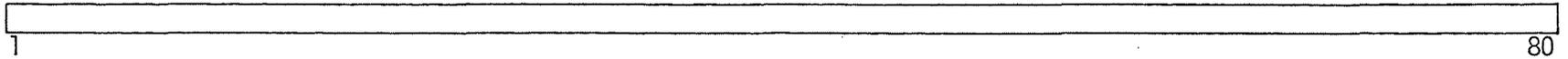
Beginning users should take note of the following:

1. The program checks to see if the lengths of the various constituents add up to the extended length furnished by the user. If a discrepancy is detected, a diagnostic message is issued and no further calculations performed. By referring to Figure 4.4 in this chapter, the user can easily compare the extended length as obtained by calculation and compare it with the input extended length. In some cases the constituent lengths may have to be altered slightly to yield consistent results.

2. The mathematical model for the cylinder consists of a set of nonlinear equations. Consequently, the application of a factor of safety on the load will not result in the same load capacity as application of the same factor of safety on stress.

Program SACREG--Guide for Data Input

PROGRAM IDENTIFICATION (Two alphanumeric cards at the beginning of run)

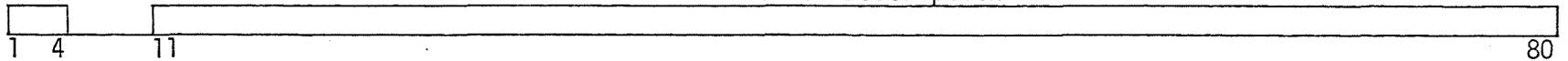


Format--20A4

PROBLEM IDENTIFICATION (One card at the beginning of each problem)

Prob.
Name
NPROB

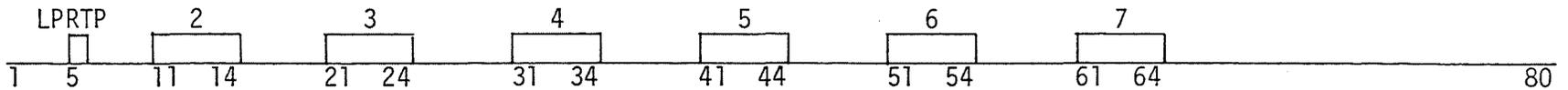
Problem Description



Format--20A4

Program stops if NPROB is blank

TABLE 1: CONTROL DATA (One card for each problem)



Format--LPRTP - I1; 2 to 7 - A4

LP RTP = 1--Critical load analysis and analysis for a factored load using given factor of safety

2--Analysis for a particular fluid pressure

3--Analysis to determine a stop tube length for given limiting values of crookedness angle and lateral force at the sliding connection at a given fluid pressure

If any of the following tables are same as in the previous problem and are to be retained for this problem, enter "KEEP" in the corresponding blocks 2 to 7

Enter only LP RTP for the first problem

TABLE 2: UNITS OF MEASUREMENTS (No card if TABLE 2 is retained from previous problem)

	LNTU	LODU	LPREU	LANGU	
1	11 14	21 24	31 34	41 44	80

Format--A4 for all

LNTU - Unit of lengths (ex: INCH, FEET, CM, MET, etc.)

LODU - Unit of loads (ex: LBS, KIPS, KGS, etc.)

LPREU - Unit of pressures (ex: KSI, PSI, KSCM, etc.)

LANGU - Unit of angles (enter DEG or RAD starting in column 41)

TABLE 3: CYLINDER DIMENSIONS (No cards if TABLE 3 is retained from previous problem; see Figure 36 for details)

Card No. 1--Lengths

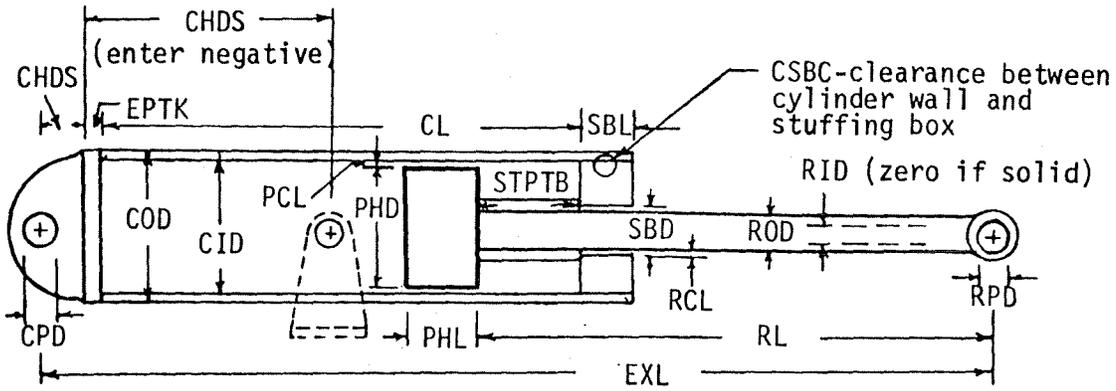


Figure 36. Cylinder Dimensions for SACREG

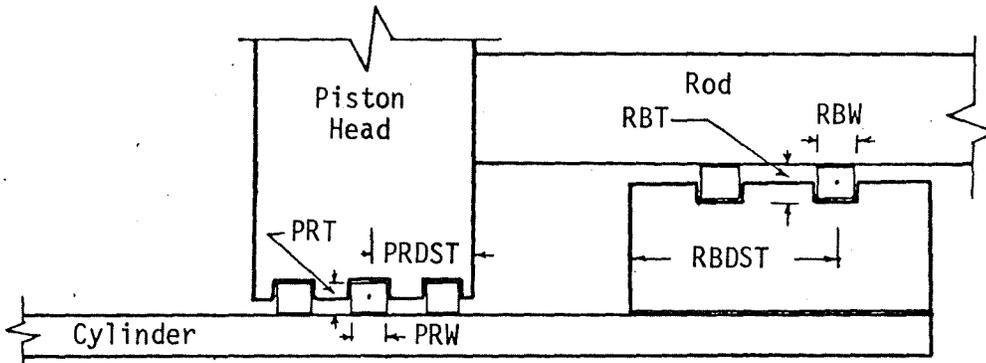


Figure 37. Dimensions of Bearings and Seals for SACREG

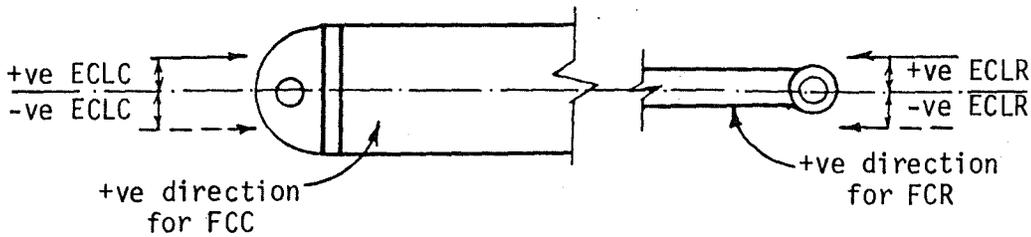
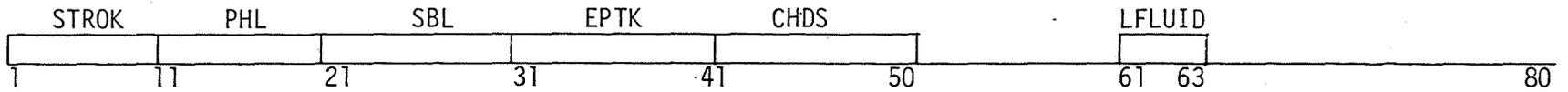


Figure 38. Sign Conventions for Eccentricities of Loading and Friction Coefficients for SACREG



Format--LFLUID--A3; E10.3 for the rest

LFLUID - Enter "YES", for hollow rod with fluid

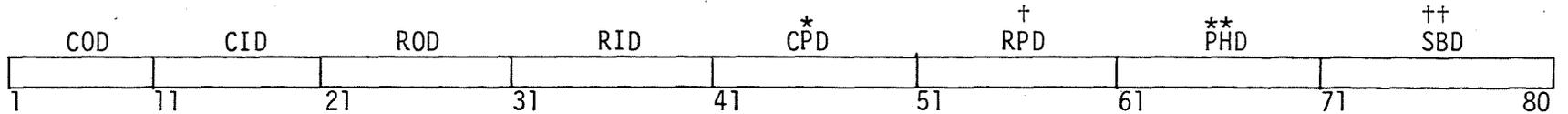
- Enter "NO" or blank, for hollow rod without fluid or solid rod

Card No. 2--Lengths (card No. 2 is not input for LPRTP = 3)



Format--E10.3 for all

Card No. 3--Diameters



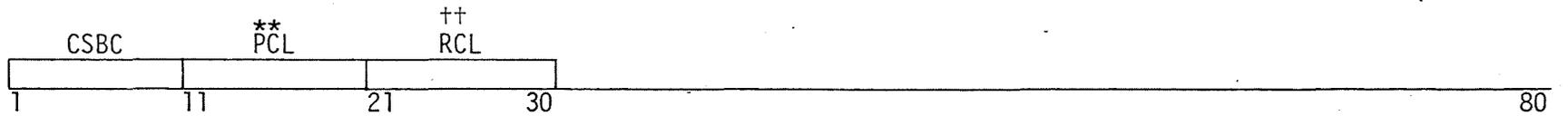
Format--E10.3 for all

* CPD - Leave blank if cylinder support is fixed

+ RPD - Leave blank if rod support is fixed

** and ++ - See next card

Card No. 4--Clearances



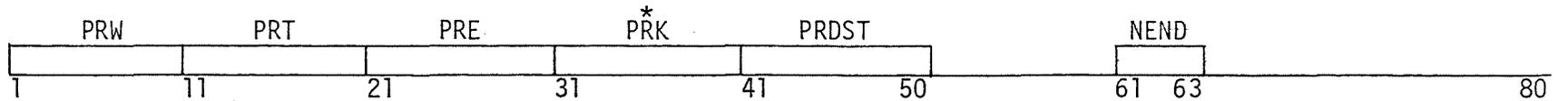
Format--E10.3 for all

**Input either PHD or PCL; if both are input, PCL will be used and PHD will be ignored

††Input either SBD or RCL; if both are input, RCL will be used and SBD will be ignored

TABLE 4: BEARINGS AND SEALS (No cards if TABLE 4 is retained from previous problem; see Figure 37 for details)

Piston Head Bearing Cards: (one card for each bearing)

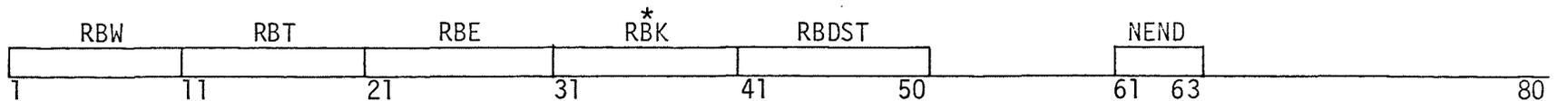


Format--NEND--A3; E10.3 for the rest

NEND - Enter "END" on the last piston head bearing card

* Input either (PRW, PRT and PRE) or (PRK); if PRK and some or all of PRW, PRT, and PRE are input, PRK will be used and the rest ignored

Rod Bearing Cards: (one card for each bearing)



Format--NEND--A3; E10.3 for the rest

NEND - Enter "END" on the last rod bearing card

* Input either (RBW, RBT, and RBE) or (RBK); if RBK and some or all of RBW, RBT and RBE are input, RBK will be used and the rest ignored

PRE and RBE - Young's modulus of piston head bearings and rod bearings

PRK and RBK - Stiffnesses of piston head bearings and rod bearings per unit length (force required to compress a unit length of bearing by one unit)

TABLE 5: WEIGHTS AND MATERIAL PROPERTIES (No card, if TABLE 5 is retained from previous problem)

WC	WR	WPH	WSB	ECYL	EROD	FYCYL	FYROD	
1	11	21	31	41	51	61	71	80

Format--E10.3 for all

WC and WR - Weight of cylinder and rod per unit length

WPH - Weight of piston head

WSB - Weight of stuffing box

ECYL and EROD - Modulus of elasticity of cylinder and rod, respectively

FYCYL and FYROD - Yield stresses of cylinder and rod, respectively

TABLE 6: INCLINATION, FIXITY, FRICTION COEFFICIENTS, AND LOADING ECCENTRICITIES (No card if TABLE 6 is retained from previous problem)

CINCL	LCEND	LTEND	*FCC	*FCR	*ECLC	*ECLR	
1	31	36	41	51	61	71	80
10	33	38					

Format--LCEND and LREND--A3; E10.3 for the rest

CINCL - Inclination of the cylinder with horizontal (always positive and between 0° and 90°)

LCEND - Enter FIX for fixed or PIN for pinned cylinder support

LREND - Enter FIX for fixed or PIN for pinned rod support

FCC - Friction coefficient at cylinder pin. Leave blank if LCEND is FIX

FCR - Friction coefficient at rod pin. Leave blank if LREND is FIX

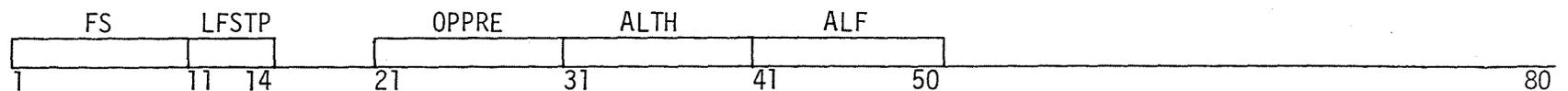
ECLC - Eccentricity of loading at cylinder end

ECLR - Eccentricity of loading at rod end

* See Figure 38 for sign convention

The direction of friction moments at the pins should be visualized by the user depending on the direction of rotation of the pins in the case of rotating pins, and depending on the predicted direction of the slopes at ends of the system for that particular loading, and accordingly proper signs should be assigned for FCC and FCR

TABLE 7: FACTOR OF SAFETY, OPERATING PRESSURE, ALLOWABLE θ AND F (No card if TABLE 7 is retained from previous problem)



Format--LFSTP--A4; E10.3 for the rest

If LPRTP = 1 - enter only FS and LFSTP

= 2 - enter only OPPRE

= 3 - enter only OPPRE, ALTH and ALF

FS - Factor of safety

LFSTP - Factor of safety type (enter LOAD if FS is to be applied to the critical load obtained; enter STRS if FS is to be applied to the limiting stresses)

If only critical load analysis is required and no factored load analysis is required, leave this card blank or enter FS \leq 1.0, and LFSTP--LOAD or STRS or blank

OPPRE - Particular operating pressure for which analysis is required

ALTH - Allowable crookedness angle at the sliding connection

ALF - Allowable total lateral force on bearings (total force on piston head bearings or total force on rod bearings which are equal to each other)

NEXT PROBLEM

Start from "PROBLEM IDENTIFICATION" card

END OF RUN

At the end of last problem data set, insert a blank card (only first 4 columns need to be blank, the rest of the card may be used for comments)

APPENDIX E
FORTRAN LISTING OF SACREG

C---- >>> MAIN PROGRAM

C

```

IMPLICIT REAL*8 ( A-H, C-Z )
COMMON EXL, P
COMMON / ID / IDCARD(40), NPRUB, IPRUB(19), LPRTP
COMMON / UNITS / LNTU, LDDU, LPREU, LANGU
COMMON / LENGTS / STROK, PHL, SBL, EPTK, CHDS, LFLUID
COMMON / STPTBS / CL, RL, STPTB
COMMON / DIAMTS / COD, CID, ROD, RID, CPD, RPD, PHD, SBD
COMMON / CLERNC / CSBC, PCL1, RCL1
COMMON / CLEAR / PCL, RCL
COMMON / PISTON / PRW(5), PRT(5), PKE(5), PRK(5), PRDST(5), NPHBR
COMMON / RODBRS / RBW(5), RBT(5), RBE(5), RBK(5), RBDST(5), NRDBR
COMMON / WGTINI / WC1, WR1, WPH1, WSB1
COMMON / WGTVER / WC, WR
COMMON / WGTCON / WPH, WSB
COMMON / PROPTS / ECYL, ERUD, FYCYL, FYROD
COMMON / ENDS / LCEND, LREND
COMMON / INCLFR / CINCL, FCC, FCR
COMMON / ECCTRI / ECLC, ECLR
COMMON / FSOPTF / OPPRE, ALTH, ALF, FS, LFSTP
COMMON / CONSTS / EEL, FFL, AKTHC, AKTHR
COMMON / CANDDS / C1, C2, D1, D2
COMMON / GLDFOR / FX(5), FY(5), F1, F2, F3, F4
COMMON / FRCCNS / FCCY, FCRD, CCNM
COMMON / BRGSTF / PRKX, SPRK, RBKY, SRBK
COMMON / CRPROP / RDZ, CYZ, RDZI, CYZI, HSCCI, HSCCC, HSCRI,
*           HSCRO, BAREAC, BAREAR, CAREAC, CAREAR

```

C

```

DATA IBLNK, LOAD / 4H      , 4HLOAD /
DATA PI, LDEG, H180 / 3.141593D00, 4HDEG , 180.0D00 /
DATA ZERO, ONE / 0.0D00, 1.0D00 /

```

C

```

          NPROB = IBLNK
10 CALL INPECO ( IBLNK, GC )
          FYCYLT = FYCYL
          FYRCDT = FYROD
CALL CONST1
I      ( ECYL, ERUD, LCEND, CHDS, LANGU,
O      RDI, CYK, RDK )
CALL TRIALP
I      ( ERCD, LPRTP, RDI, FYCYL, BAREAC, HSCCI, LCEND,
O      PINCRI, PINCR2 )

```

C

C---- >>> KEY-- KWIT IS SETUP TO APPLY FACTOR OF SAFETY AND REPEAT
ANALYSIS FOR FACTORED LOAD

C

KWIT = 1

C

C---- >>> KEY-- KEYF IS SETUP TO QUIT LOOP AT FINAL ITERATION

C

```

2C      KEYF = 1
        KEYST = 1

```

C

C---- >>> KEY'S-- KEYT AND KEYP ARE SETUP TO MAKE PROPER LOAD INCREMENTS

C

```

        KEYT = 1
        KEYP = 1

```

```

      TETA = ZERO
      BMG   = ZERC
30 CALL EQBRIM
   I      ( PRK, PRDST, RBK, RBDST, NPHBR, NRDBR, CYK, RDK, LPRTP, GC,
   I      RPD, BMG,
   O      CLT, RLT, CK, RK, BBI, BB2, TETA, KASE, DEFG, THC )
      IF ( LPRTP .NE. 3 ) GO TO 40
      IF ( KEYST .EQ. 1 ) GO TO 40
      ALTH1 = ALTH
      TEMP  = PI / H180
      IF ( LANGU .EQ. LDEG ) ALTH1 = ALTH * TEMP
      ITERAT = 1
      CALL STOPTB
   I      ( TETA, NPHBR, NRDBR, ALTH1, ALF, STROK, ITERAT,      GC )
      IF ( ITERAT .NE. 1 ) GO TO 30
40 CALL XATYMX
   I      ( CLT, CK, RK, DEFG,
   O      XCY, XRD, CSLPC, CSLPR )
      CALL YMAXS
   I      ( XCY, XRD, CK, RK, CSLPC, CSLPK, BBI, BB2,
   O      YCMAX, YRMAX )
      CALL STRCHS
   I      ( KEYF, KEYT, KEYP, XCY, XRD, YCMAX, YRMAX, RID, CHDS,
   I      OPPRE, FYRODT, FYCYLT, PINCR1, PINCR2, KEYST, LFLUID,
   O      HSC, HSR, AXTEN, CSTR, CSTRP, RSTR, RSTRP, CSS, NCSS, RSS, NRSS)
C
C---- >>> IS IT FINAL ITERATION? ? ?
C
      IF ( KEYF .NE. 3 ) GO TO 30
      IF ( KEYST .NE. 1 ) GO TO 50
      IF ( LPRTP .EQ. 1 ) GO TO 50
      PRES = P / BAREAC
      IF ( CPPRE .GT. PRES ) GO TO 100
      P    = OPPRE * BAREAC
      KEYST = 2
      GO TO 30
50 CALL OUTPUT
   I      ( KWIT, BAREAC, XCY, XRD, YCMAX, YRMAX, CHDS, GC, TETA,
   I      FYCYL, FYROD, CSTR, CSTRP, RSTR, RSTRP, HSC, HSR, AXTEN,
   I      NPHBR, NRDBR, THC, CSS, NCSS, RSS, NRSS )
C
C---- >>> IF THIS PROBLEM IS COMPLETE GO TO NEXT PROBLEM
C
      IF ( KWIT .NE. 1 ) GO TO 10
      IF ( LPRTP .NE. 1 ) GO TO 10
      IF ( FS .EQ. ZERO .OR. FS .EQ. ONE ) GO TO 10
C
C---- >>> FACTOR OF SAFETY IS TO BE APPLIED TO STRESS OR LOAD
C
      IF ( LFSTP .EQ. LOAD ) GO TO 60
      FYCYLT = FYCYL / FS
      FYRODT = FYROD / FS
      P      = P / FS
      KWIT   = 2
      GO TO 20
60      P      = P / FS
      KWIT   = 2
      GO TO 30

```

10C PRINT 210, PRES
GO TO 10

210 FORMAT (1H1, 20(//), 10(10X, 21H*** ** ERROR *** ** /), //, //,
1 10X, 38HOPERATING PRESSURE IS GREATER THAN THE /
2 10X, 42HCAPACITY(CRITICAL LOAD) OF THE CYLINDER. //
3 10X, 37HTHE MAXIMUM PRESSURE FOR THE CYL IS =, 1PD10.3, //)
END

SUBROUTINE INPECO (IBLNK, GC)

C

C----- >>> SUBROUTINE TO READ AND ECHC INPUT DATA FOR SACREG

C

IMPLICIT REAL*8 (A-H, O-Z)
 COMMON EXL, F
 COMMON / ID / IDCARD(40), NPROB, IPROB(19), LPRT P
 COMMON / UNITS / LNTU, LODU, LPREU, LANGU
 COMMON / LENGTHS / STROK, PHL, SBL, EPTK, CHDS, LFLUID
 COMMON / STPTBS / CL, RL, STPTB
 COMMON / DIAMTS / COD, CID, ROD, RID, CPD, RPD, PHD, SBD
 COMMON / CLERNC / CSBC, PCL1, RCL1
 COMMON / PISTON / PRW(5), PRT(5), PRE(5), PRK(5), PRDST(5), NPHBR
 COMMON / RODERS / RBW(5), RBT(5), RBE(5), RBK(5), RBDST(5), NRDBR
 COMMON / WGTINI / WC1, WR1, WPH1, WSBI
 COMMON / PROPTS / ECYL, EROD, FYCYL, FYROD
 COMMON / ENDS / LCEND, LREND
 COMMON / INCLFR / CINCL, FCC, FCR
 COMMON / ECCTRI / ECLC, ECLR
 COMMON / FSOFTF / OPPRE, ALTH, ALF, FS, LFSTP

C

DATA KEEP, IEND / 4HKEEP, 3HEND /
 DATA ZERC, TWO / 0.0000, 2.0000 /
 DATA LYES / 3HYES /

C

C----- >>> FORMATS

C

10 FORMAT (20A4)
 20 FORMAT (A4, 19A4)
 30 FORMAT (4X, I1, 5X, 6(A4, 6X))
 40 FORMAT (4X, 4(6X, A4))
 50 FORMAT (8F10.0)
 60 FORMAT (5F10.0, 10X, A3)
 70 FORMAT (3F10.0, A3, 2X, A3, 2X, 4F10.0)
 80 FORMAT (F10.0, A4, 6X, 3F10.0)
 90 FORMAT (1H1, //, 5X, 36HPROGRAM SACREG - STRESS ANALYSIS OF ,
 1 21HCYLINDERS (REGULAR), ///, 2(7X, 20A4/), /)
 100 FORMAT (/, 5X, 8HPROBLEM , A4, //, 1X, 19A4)
 110 FORMAT (/, 5X, 11HINPUT DATA:, //, 5X, 8HTABLE 1:, 5X,
 1 12HCONTROL DATA, /)
 120 FORMAT (/, 10X, 41HPROBLEM TYPE = 1 - CRITICAL LOAD ANALYSIS,
 1 31H & ANALYSIS FOR A FACTORED LOAD, /)
 130 FORMAT (/, 10X, 27HPROBLEM TYPE = 2 - ANALYSIS,
 1 26H FOR A PARTICULAR PRESSURE, /)
 140 FORMAT (/, 10X, 27HPROBLEM TYPE = 3 - ANALYSIS,
 1 39H TO DETERMINE SUITABLE STOP-TUBE LENGTH, /)
 160 FORMAT (/, 18X, 37HTABLES RETAINED FROM PREVIOUS PROBLEM, //,
 1 20X, 2H 2, 4X, 2H 3, 4X, 2H 4, 4X, 2H 5, 4X, 2H 6,
 2 4X, 2H 7, /, 17X, 6(2X, A4))
 190 FORMAT (23X, 25HNO KEEP OPTIONS EXERCISED, /)
 210 FORMAT (/, 5X, 33HTABLE 2: UNITS OF MEASUREMENT, //, 17X,
 1 6HLENGTH, 6X, 4HLOAD, 5X, 8HPRESSURE, 3X, 7HANGULAR,
 2 //, 11X, 4(7X, A4))
 220 FORMAT (//, 5X, 32HTABLE 3: CYLINDER DIMENSIONS, //, 10X,
 1 8HLENGTHS:, //, 19X, 23HSTROKE PISTON HEAD, 2X,
 2 40HSTUFFING BOX END PLATE HINGE DIST., //, 14X,
 3 5(2X, 1PD12.5))
 230 FORMAT (//, 18X, 20HCYLINDER ROD, 8X,

```

1      23HEXTENDED      STOP TUBE      )
240 FORMAT ( /, 14X, 4( 2X, 1PD12.5 ) )
250 FORMAT ( /, 15X, 23HTHESE NCT INPUT BECAUSE,
1      35H STOP TUBE LENGTH ANALYSIS IS ASKED, / )
260 FORMAT ( /, 10X, 10HDIAMETERS:, //, 17X, 10HCYL. CUTER, 4X,
1      38HCYL. INNER      ROD CUTER      ROD INNER, / )
270 FORMAT ( /, 14X, 4( 2X, 1PD12.5 ), 2X, 9HSOLID ROD, / )
280 FORMAT ( /, 14X, 4( 2X, 1PD12.5 ), 2X, 10HHOLLOW ROD )
282 FORMAT ( /, 72X, 10HWITH FLUID, / )
284 FORMAT ( /, 72X, 13HWITH NO FLUID, / )
290 FORMAT ( /, 18X, 22HCYL. PIN      ROD PIN, 5X,
1      25HPISTON HEAD STUFFING BOX, //, 14X, 4( 2X, 1PD12.5 ),
2      //, 16X, 26H(IF ZERO THE END IS FIXED),
3      32H (IF ZERO OTHER OPTION IS INPUT), / )
310 FORMAT ( /, 10X, 19HCLEARANCES BETWEEN:, //, 12X,
1      2( 6X, 8HCYLINDER ), 8X, 3HROD, /, 9X, 3( 11X, 3HAND ),
2      /, 16X, 12HSTUFFING BCX, 2X, 11HPISTON HEAD, 3X,
3      12HSTUFFING BOX, //, 14X, 3( 2X, 1PD12.5 ), /, 29X,
4      31H(IF ZERO OTHER OPTION IS INPUT), / )
320 FORMAT ( /, 5X, 31HTABLE 4:      BEARINGS AND SEALS, //, 10X,
1      16HPISTON BEARINGS:, / )
330 FORMAT ( /, 21X, 3( 2H A, 12X ), 2H B, 7X, 13HDISTANCE FROM, /,
1      20X, 5HWIDTH, 7X, 9HTHICKNESS, 2X, 14HYOUNGS MODULUS, 3X,
2      9HSTIFFNESS, 5X, 9HBACK FACE, //,
3      5( 14X, 5( 2X, 1PD12.5 ), / ) )
340 FORMAT ( /, 10X, 13HROD BEARINGS:, / )
350 FORMAT ( /, 15X, 48H(A IS USED TO CALCULATE B - HENCE, EITHER A OR
1      , 10HB IS INPUT, /, 28X,
2      46HZERO'S ABOVE INDICATE THAT THEY ARE NCT INPUT), / )
360 FORMAT ( /, 5X, 44HTABLE 5:      WEIGHTS AND MATERIAL PROPERTIES,
1      //, 10X, 17HWEIGHTS OF PARTS:, //, 18X, 8HCYLINDER, 9X,
2      3HROD, 7X, 11HPISTON HEAD, 2X, 12HSTUFFING BOX, /, 21X,
3      17H(PER UNIT LENGTH), //, 14X, 4( 2X, 1PD12.5 ), / )
370 FORMAT ( /, 10X, 2CHMATERIAL PRCPERTIES:, //, 22X,
1      14HYOUNGS MODULUS, 15X, 12HYIELD STRESS, /, 10X,
2      2( 8X, 8HCYLINDER, 9X, 3HROD ),//, 14X, 4( 2X, 1PD12.5)//)
380 FORMAT ( /, 5X, 43HTABLE 6:      INCLINATION, FIXITY, FRICTION ,
1      34HCOEFFICIENTS, LOADING ECCENTRICITY, //, 10X,
2      34HCYL INCLINATION WITH HORIZONTAL = , 1PD12.5, //, 10X,
3      45HSUPPORT CONDITIGNS - CYLINDER END      RCD END, //,
4      25X, 2( 11X, A3 ), //, 10X, 25HFRICITION COEFFICIENTS AT
5      9HSUPPORTS:, //, 32X, 8HCYLINDER, 9X, 3HROD, //, 28X,
6      2( 2X, 1PD12.5 ), /, 34X, 19H(ZERO IF FIXED END), //, 10X
7      , 21HLOADING ECCENTRICITY:, //, 32X, 8HCYLINDER, 9X,
8      3HROD, //, 28X, 2( 2X, 1PD12.5 ), / )
390 FORMAT ( /, 5X, 44HTABLE 7:      FACTOR OF SAFETY OR OPERATING,
1      38H PRESSURE AND/OR ALLOWABLE THETA & F, /, 18X,
2      25HCEPENDING ON PROBLEM TYPE, / )
410 FORMAT ( /, 10X, 19HFACTOR OF SAFETY = , F6.3, 4H ON , A4, / )
420 FORMAT ( /, 10X, 21HOPERATING PRESSURE = , 1PD12.5, / )
430 FORMAT ( /, 10X, 32HOPERATING CYLINDER PRESSURE = , 1PD12.5, //,
1      10X, 32HALLOWABLE CROOKEDNESS ANGLE = , 1PD12.5,
2      10H AT GLAND, //, 10X, 32HALLOWABLE TOTAL LATERAL FORCE =
3      1PD12.5, 13H ON BEARINGS, / )
450 FORMAT ( ///, 10X, 30H***** ERROR IN LENGTHS ***** , / )
460 FORMAT ( ///, 10X, 7H***** ,
1      32HERROR : PHD IS GREATER THAN CID , 6H ***** , / )
470 FORMAT ( ///, 10X, 7H***** ,

```

```

1
48C FORMAT ( //, 10X, 6H***** , 19HPROGRAM TERMINATED , 5H***** )
49C FORMAT ( 1H1 )
C
C---- >>> READ AND ECHO RUN AND PROBLEM IDENTIFICATION
C
      IF ( NPRCB .NE. IBLNK ) GO TO 500
      READ ( 5, 10 ) ( IDCARD( I ), I = 1, 40 )
500 READ ( 5, 20 ) NPRCB, ( IPKCB( I ), I = 1, 19 )
C
C---- >>> TEST FOR END OF RUN
C
      IF ( NPRCB .EQ. IBLNK ) GO TO 1700
      PRINT 90, ( IDCARD( I ), I = 1, 40 )
      PRINT 100, NPRCB, ( IPKCB( I ), I = 1, 19 )
C
C---- >>> READ TABLE 1: PROBLEM TYPE AND TABLES TO BE RETAINED FROM
C                          PREVIOUS PROBLEM
C
      READ ( 5, 30 ) LPRTP, KEEP2, KEEP3, KEEP4, KEEP5, KEEP6, KEEP7
      IF ( KEEP2 .EQ. KEEP ) GO TO 510
C
C---- >>> READ TABLE 2: UNITS OF MEASUREMENT
C
      READ ( 5, 40 ) LNTU, LODU, LPREU, LANGU
510      IF ( KEEP3 .EQ. KEEP ) GO TO 540
      READ ( 5, 60 ) STROK, PHL, SBL, EPTK, CHDS, LFLUID
      IF ( LPRTP .EQ. 3 ) GO TO 520
C
C---- >>> READ TABLE 3: LENGTHS AND DIAMETERS
C
      READ ( 5, 50 ) CL, RL, EXL, STPTB
520 READ ( 5, 50 ) COD, CID, ROD, RID, CPD, RPD, PHD, SBD
      READ ( 5, 50 ) CSBC, PCL1, RCL1
C
C---- >>> CALCULATE GLAND CLEARANCE
C
      GC = ZERO
      IF ( LPRTP .NE. 3 ) GC = CL - STROK - PHL
C
C---- >>> TEST FOR PROPER INPUT
C
      IF ( LPRTP .EQ. 3 ) GO TO 530
      A = EXL - CHDS - EPTK - CL - SBL - RPD / TWO
      IF ( A .LT. STROK ) GO TO 1000
530      IF ( ( PCL1 .GT. ZERO ) .OR. ( RCL1 .GT. ZERO ) ) GO TO 540
      IF ( PHD .GT. CID ) GO TO 1100
      IF ( ROD .GT. SBD ) GO TO 1200
540      IF ( KEEP4 .EQ. KEEP ) GO TO 580
C
C---- >>> READ TABLE 4: PISTON RINGS AND ROD BEARINGS DETAILS
C
      I = 1
      J = 1
550 READ ( 5, 60 ) PRW(I), PRT(I), PRE(I), PRK(I), PRDST(I), NEND
      NPHBR = I
      IF ( NEND .EQ. IEND ) GO TO 560
      I = I + 1

```

```

      GO TO 55C
56C PEAD ( 5, 60 ) RBW(J), RBT(J), RBE(J), RBK(J), RBCST(J), NEND
      NRDBR = J
      IF ( NEND .EQ. IEND ) GO TO 580
      J = J + 1
      GO TO 56C
580 CONTINUE
      IF ( KEEP5 .EQ. KEEP ) GO TO 590
C
C---- >>> READ TABLE 5: WEIGHTS OF PARTS AND MATERIAL PROPERTIES
C
      READ ( 5, 50 ) WC1, WR1, WPH1, WSB1, ECYL, EROD, FYCYL, FYROD
59C      IF ( KEEP6 .EQ. KEEP ) GO TO 600
C
C---- >>> READ TABLE 6: INCLINATION, END FIXITY, FRICTION COEFFICIENTS
C                          AND ECCENTRICITY OF LOADING
C
      READ ( 5, 70 ) CINCL, CSSTF, RSSTF, LCEND, LREND, FCC, FCR, ECLC,
*      ECLR
600      IF ( KEEP7 .EQ. KEEP ) GO TO 610
C
C---- >>> READ TABLE 7; FACTOR OF SAFETY AND ITS TYPE, OPERATING
C                          PRESSURE AND ALLOWABLE THETA AND LATERAL FORCE
C                          DEPENDING ON THE PROBLEM TYPE
C
      READ ( 5, 80 ) FS, LFSTP, OPPRE, ALTH, ALF
61C CONTINUE
      PRINT 110
      IF ( LPRTP .EQ. 1 ) PRINT 120
      IF ( LPRTP .EQ. 2 ) PRINT 130
      IF ( LPRTP .EQ. 3 ) PRINT 140
      PRINT 160, KEEP2, KEEP3, KEEP4, KEEP5, KEEP6, KEEP7
      IF ( KEEP2 .NE. IBLNK ) GO TO 620
      IF ( KEEP3 .NE. IBLNK ) GO TO 620
      IF ( KEEP4 .NE. IBLNK ) GO TO 620
      IF ( KEEP5 .NE. IBLNK ) GO TO 620
      IF ( KEEP6 .NE. IBLNK ) GO TO 620
      IF ( KEEP7 .EQ. IBLNK ) PRINT 190
62C CONTINUE
C
C---- >>> PRINT ALL THE TABLES READ
C
      PRINT 210, LNTU, LCDU, LPREU, LANGU
      PRINT 220, STROK, PHL, SBL, EPTK, CHDS
      PRINT 230
      IF ( LPRTP .EQ. 3 ) GO TO 640
      PRINT 240, CL, RL, EXL, STPTB
      GO TO 650
64C PRINT 250
65C PRINT 260
      IF ( RID .GT. ZERO ) GO TO 670
      PRINT 270, CCD, CID, ROD, RID
      GO TO 680
67C PRINT 280, COD, CID, ROD, RID
      IF ( LFLUID .NE. LYES ) GO TO 675
      PRINT 282
      GO TO 680
675 PRINT 284

```

```
680 PRINT 290, CPO, RPD, PHD, SBD
PRINT 310, CSBC, PCL1, RCL1
PRINT 320
PRINT 330, ( PRW(I), PRT(I), PRE(I), PRK(I), PRDST(I), I=1, NPHBR )
PRINT 340
PRINT 330, ( RBW(I), RBT(I), RBE(I), RBK(I), RBDST(I), I=1, NRDBR )
PRINT 350
PRINT 360, WCI, WRI, WPHI, WSBI
PRINT 370, ECYL, EROD, FYCYL, FYROD
PRINT 380, CINCL, LCEND, LREND, FCC, FCR, ECLC, ECLR
PRINT 390
      IF ( LPRTP .EQ. 1 ) PRINT 410, FS, LFSTP
      IF ( LPRTP .EQ. 2 ) PRINT 420, OPPRE
      IF ( LPRTP .EQ. 3 ) PRINT 430, OPPRE, ALTH, ALF
GO TC 1500
C
C---- >>> PRINT ERROR MESSAGES FOR INPUT ERROR
C
1000 PRINT 450
      GO TO 1600
1100 PRINT 460
      GO TC 1600
1200 PRINT 470
      GO TC 1600
1500 CCNTINUE
      RETURN
1600 PRINT 480
1700 PRINT 490
C
C---- >>> END OF RUN IF ERROR IN INPUT IS ENCOUNTERED
C
      STOP
      END
```

```

SUBROUTINE CONST1
  I      ( ECYL, EROD, LCEND, CHDS, LANGU,
  O      PDI, CYK, ROK )
C
C----- >>> SUBROUTINE TO CALCULATE CONSTANT TERMS FOR CONVENIENCE
C
      IMPLICIT REAL*8 ( A-H, O-Z )
      COMMON / DIAMTS / COD, CID, ROD, RID, CPD, RPD, PHD, SBD
      COMMON / CLEARC / CSBC, PCL1, RCL1
      COMMON / CLEAR / PCL, RCL
      COMMON / PISTON / PRW(5), PRT(5), PRE(5), PRK(5), PRDST(5), NPHBR
      COMMON / RCDBRS / RBW(5), RBT(5), RBE(5), RBK(5), RBDST(5), NRDBR
      COMMON / WGTINI / WCL, WR1, WPH1, WSB1
      COMMON / WGTVER / WC, WR
      COMMON / WGTCON / WPH, WSB
      COMMON / INCLFR / CINCL, FCC, FCR
      COMMON / FRCCNS / FCCY, FCPD, CCNM
      COMMON / BRGSTF / PRKX, SPRK, RBKY, SRBK
      COMMON / CRPROP / RDZ, CYZ, RDZI, CYZI, HSCCI, HSCCC, HSCRI,
*      HSCRO, BAREAC, BAREAR, CAKEAC, CAREAR
C
      DATA ZERO, TWO, FOUR / 0.0000, 2.0000, 4.0000 /
      DATA SXTFOR, PI / 64.0000, 3.141593000 /
      DATA H180, LDEG, LFIX / 180.0000, 4HDEG, 3HFIX /
C
C----- >>> CALCULATE STIFFNESSES OF BEARINGS AND SEALS IF NOT INPUT
C
      SPRK = ZERO
      PRKX = ZERO
      DO 110 I = 1, NPHBR
        IF ( PRK(I) .GT. ZERO ) GO TO 100
      100   PRK(I) = CID * PRW(I) * PRE(I) / PRT(I)
      110   PRKX = PRK(I) * PRDST(I) + PRKX
          SPRK = PRK(I) + SPRK
          SRBK = ZERO
          RBKY = ZERO
      DO 130 I = 1, NRDBR
        IF ( RBK(I) .GT. ZERO ) GO TO 120
      120   RBKY = RBK(I) * RBDST(I) + RBKY
      130   SRBK = RBK(I) + SRBK
C
C----- >>> CALCULATE CROSS SECTIONAL PROPERTIES
C
      RCD2 = ROD * ROD
      RID2 = RID * RID
      COD2 = COD * COD
      CID2 = CID * CID
      ROD4 = ROD2 * ROD2
      RID4 = RID2 * RID2
      COD4 = COD2 * COD2
      CID4 = CID2 * CID2
      RDI = PI * ( ROD4 - RID4 ) / SXTFOR
      CYI = PI * ( COD4 - CID4 ) / SXTFOR
      RDZ = RDI * TWO / ROD
      CYZ = CYI * TWO / COD
      RDZI = 1.00+20
      IF ( RID .GT. ZERO ) RDZI = RDI * TWO / RID

```

```

C
C----- >>> BORE AREAS AND CROSS SECTICNAL AREAS OF CYLINDER AND ROD
C
      BAREAC = PI * CID2 / FOUR
      BAREAR = PI * RID2 / FOUR
      CAREAC = PI * ( COD2 - CID2 ) / FOUR
      CAREAR = PI * ( ROD2 - RID2 ) / FOUR
C
C----- >>> CALCULATE HOOP STRESS CCEFFICIENT
C
      RDENC = COD2 - CID2
      HSCCI = ( COD2 + CID2 ) / RDENC
      HSCCO = TWO * CID2 / RDENC
      RDEAR = ROD2 - RID2
      HSCRI = ( ROD2 + RID2 ) / RDEAR
      HSCRO = TWO * RID2 / RDEAR
C
      CYK   = DSQRT ( ECYL * CYI )
      RDK   = DSQRT ( EROD * RDI )
C
C----- >>> CALCULATE FRICTION MOMENT COEFFICIENTS
C
      FCCY  = FCC * CPD / TWO
      FCRD  = FCR * RPD / TWO
C
C----- >>> CALCULATE CLEARANCES AT PISTON HEAD AND STUFFING BOX
C
      PCL   = ( CID - PHD ) / TWO
      IF ( PCL1 .GT. ZERO ) PCL = PCL1
      RCL   = ( SBD - RCD ) / TWO + CSBC
      IF ( RCL1 .GT. ZERO ) RCL = RCL1 + CSBC
C
C----- >>> CALCULATE VERTICAL COMPCNENTS OF WEIGHTS
C
      TEMP  = PI / HI80
      BETA  = CINCL
      IF ( LANGU .EQ. LDEG ) BETA = CINCL * TEMP
      CB    = DCOS( BETA )
      WC    = WC1 * CB
      WR    = WR1 * CB
      WPH   = WPH1 * CB
      WSB   = WSB1 * CB
C
C----- >>> MOMENT DUE TO OVER HANG
C
      CONM  = WC * CHDS * CHDS / TWO
      IF ( CHDS .GT. ZERO ) CONM = ZERO
      IF ( LCEND .EQ. LFIX ) CONM = ZERO
RETURN
END

```

```

SUBROUTINE TRIALP
  I      ( EROD, LPRTP, RDI, FYCYL, BAREAC, HSCONC, LCEND,
  O      PINCR1, PINCR2 )
C
C---- >>> SUBROUTINE TO CALCULATE TRIAL LOAD AND LOAD INCREMENTS
C
      IMPLICIT REAL*8 ( A-H, O-Z )
      COMMON EXL, P
      COMMON / LENGTS / STROK, PHL, SBL, EPTK, CHDS, LFLUID
C
      DATA TWO, PI, FIFTY, FIVHUN / 2.0000, 3.141593000, 50.000, 500.000 /
      DATA LFIX / 3HFIX /
C
      IF ( LPRTP .EQ. 3 ) EXL = STROK + STROK + PHL + SBL + CHDS + EPTK
C
C---- >>> CALCULATE FIRST TRIAL LOAD
C
      PI2      = PI * PI
      EXL2     = EXL * EXL
C
C---- >>> TRIAL LOAD (1) AS PER EULERS BUCKLING, CONSIDERING FULL
C          LENGTH STIFFNESS AS THAT OF ROD ONLY
C
      P1      = PI2 * EROD * RDI / EXL2
      IF ( LCEND .EQ. LFIX ) P1 = TWO * P1
C
C---- >>> TRIAL LOAD (2) AS REQUIRED BY EXCESSIVE HOOP STRESS RESTRICTION
C
      P2      = FYCYL * BAREAC / HSCONC
C
C---- >>> TRIAL LOAD, SMALLER OF (1) AND (2)
C
      IF ( P2 .LT. P1 ) P1 = P2
      PINCR1 = P1 / FIFTY
      PINCR2 = P1 / ( FIVHUN * TWO )
      P      = P1
RETURN
END

```

```

SUBROUTINE EQBRIM
I      ( PRK, PRDST, RBK, RBDST, NPHBR, NRDBR, CYK, RDK, LPRTP, GC,
I      RPD, BMG,
J      CLT, RLT, CK, RK, BB1, BB2, TETA, KASE, DEFG, THC )
C
C----- >>> SUBROUTINE TO CALCULATE CONSISTENT VALUES OF DEFLECTIONS AND
C          CROOKEDNESS ANGLE BY ITERATING, FOR A PARTICULAR VALUE OF LOAD
C
      IMPLICIT REAL*8 ( A-H, O-Z )
      COMMON EXL, P
      COMMON / LENGTS / STROK, PHL, SBL, EPTK, CHDS, LFLUID
      COMMON / STPTBS / CL, RL, STPTB
      COMMON / CLEAR / PCL, RCL
      COMMON / WGTVER / WC, WR
      COMMON / WGTCON / WPH, WSB
      COMMON / ENDS / LCEND, LREND
      COMMON / ECCTRI / ECLC, ECLR
      COMMON / CONSTS / EEL, FFL, AKTHC, AKTHR
      COMMON / CANCDS / C1, C2, D1, D2
      COMMON / GLDFOR / FX(5), FY(5), F1, F2, F3, F4
      COMMON / FRCCNS / FCCY, FCRO, CCNM
      COMMON / REATNS / REC, RER
      COMMON / BRGSTF / PRKX, SPRK, RBKY, SRBK
C
      DIMENSION PRK( NPHBR ), PRDST( NPHBR ), RBK( NRDBR ), RBDST( NRDBR )
C
      DATA ZERO, TWO, TEN, HUNDRD / 0.000, 2.000, 10.000, 50.000 /
      DATA LFIX / 3HFIX /
C
      N      = 1
      CALL THETA
I      ( PRK, PRDST, RBK, RBDST, NPHBR, NRDBR, GC, BMG, PHL, SBL,
O      X, Y, TETA, KASE )
      IF ( LPRTP .NE. 3 ) GO TO 20
C
C----- >>> CALCULATE CYLINDER, ROD AND EXTENDED LENGTHS IN CASE OF
C          STOP-TUBE LENGTH DETERMINATION ANALYSIS
C
      CL      = STROK + PHL + GC
      RL      = STROK + SBL + RPD / TWO + GC
      EXL     = CL + EPTK + CHDS + RL - GC
C
C----- >>> CALCULATE TRANSFORMED CYLINDER AND ROD LENGTHS
C
20      CLT    = CL + EPTK + CHDS - Y
      RLT     = RL - X
      W       = WPH + WSB + WC * Y + WR * X
      CLT2    = CLT * CLT
      RLT2    = RLT * RLT
C
C----- >>> CALCULATE LATERAL REACTIONS AT SUPPORTS
C
      REC1    = W * RLT + CONM
      REC2    = WC * CLT * ( RLT + CLT / TWO ) + WR * RLT2 / TWO
      REC     = ( REC1 + REC2 ) / EXL
      TOTW    = W + WC * CLT + WR * RLT
      RER     = TOTW - REC
C

```

```

C---- >>> ESTABLISH STIFFNESSES FOR ROTATIONAL SPRINGS AT SUPPORTS AS
C          PER INPUT SUPPORT CONDITIONS
C

```

```

          CSSTF = ZERO
          RSSTF = ZERO
          IF ( LCEND .EQ. LFIX ) CSSTF = TEN * CYK * CYK
          IF ( LREND .EQ. LFIX ) RSSTF = TEN * RDK * RDK
          CSSTF = CSSTF * CSSTF * CSSTF
          RSSTF = RSSTF * RSSTF * RSSTF
          CALL THCDS
          I      ( CYK, RDK, CLT, RLT, CSSTF, RSSTF, TETA, W,
          O      BB1, BB2, CK, RK, CK11, SK11, THC )
          DEFG1 = ( AKTHC - AKTHR ) * CLT / EXL
C

```

```

C---- >>> CALCULATE DEFLECTION AT SLIDING CONNECTION
C

```

```

          DEFG1 = DEFG1 - WC * CLT * CLT / ( TWO * P )
          DEFG  = C1 * CK11 + C2 * SK11 - EEL * CLT + FFL * CLT
          DEFG  = DEFG - DEFG1 + BB1 - REC * CLT / P
C

```

```

C---- >>> CALCULATE BENDING MOMENT AT INTERFACE
C

```

```

          BMG    = P * CLT * ( EEL - FFL ) + DEFG1 * P + P * ECLC
          BMG    = BMG + REC * CLT - CONM - P * FCCY - AKTHC * P
          BMG    = BMG + P * DEFG
          CALL THETA
          I      ( PRK, PRDST, RBK, RBDST, NPHBR, NRDBR, GC, BMG, PHL, SBL,
          O      X, Y, TETA, KASE )
          DTETA  = DABS( TETA / HUNDRD )
          DIFF   = DABS( TETA - TETA )
          TETA   = TETA
C

```

```

C---- >>> ARE INITIAL AND FINAL THETAS CLOSE?
C

```

```

          IF ( DIFF .LT. DTETA ) RETURN
          N    = N + 1
          IF ( N .GT. 10 ) RETURN
          GO TO 20

```

```

END

```

SUBROUTINE THETA

```

I      ( PRK, PRDST, RBK, RBDST, NPHBR, NRDBR, GC, BMG, PHL, SBL,
O      X, Y, TETA, KASE )

```

C

```

C---- >>> SUBROUTINE TO CALCULATE CROCKEDNESS ANGLE AND FORCES AT
C      INTERFACE
C

```

```

      IMPLICIT REAL*8 ( A-H, O-Z )
      COMMON / CLEAR / PCL, RCL
      COMMON / GLDFOR / FX(5), FY(5), F1, F2, F3, F4
      COMMON / BRGSTF / PRKX, SPRK, RBKY, SRBK

```

C

```

      DIMENSION PRK( NPHBR ), PRDST( NPHBR ), RBK( NRDBR ), RBDST( NRDBR )

```

C

```

      DATA ZERC, ONE, TWO / 0.0000, 1.0000, 2.0000 /

```

C

```

          SIGN = ONE
          IF ( BMG .EQ. ZERO ) GO TO 100
          SIGN = BMG / DABS( BMG )
          SRPBK = SRBK + SPRK

```

100

C

```

C---- >>> CASE 1: NO CONTACT
C

```

```

          X      = ( RBKY + GC * SRBK - PRKX ) / SRPBK
          Y      = GC - X
          IF ( PCL .EQ. ZERO .AND. RCL .EQ. ZERO ) GO TO 690
          CF     = ZERO

```

```

DO 130 I = 1, NPHBR

```

130

```

          CF     = CF + PRK( I ) * ( X + PRDST( I ) ) * ( X + PRDST( I ) )

```

```

DO 140 I = 1, NRDBR

```

140

```

          CF     = CF + RBK( I ) * ( Y + RBDST( I ) ) * ( Y + RBDST( I ) )
          TETA   = BMG / CF

```

```

      CALL GFORCE ( PRK, PRDST, TETA, X, NPHBR,          FX )

```

```

      CALL GFORCE ( RBK, RBDST, TETA, Y, NRDBR,          FY )

```

```

          D1     = ( X + PHL ) * DABS( TETA )

```

```

          D2     = ( Y + SBL ) * DABS( TETA )

```

```

      IF ( ( D1 .GE. PCL ) .AND. ( D2 .GE. RCL ) ) GO TO 170

```

```

      IF ( D1 .GE. PCL ) GO TO 200

```

```

      IF ( D2 .GE. RCL ) GO TO 300

```

```

          F1     = ZERO

```

```

          F2     = ZERO

```

```

          F3     = ZERO

```

```

          F4     = ZERO

```

```

          KASE   = 1

```

```

      RETURN

```

170

```

          D22    = PCL * ( Y + SBL ) / ( X + PHL )

```

```

      IF ( D22 .GE. RCL ) GO TO 300

```

C

```

C---- >>> CASE 2: CONTACT AT FRONT FACE OF PISTON HEAD
C

```

C

200

```

          XNUM   = ZERO

```

```

          XDEN   = ZERO

```

```

          A      = PHL + GC

```

```

          B      = DABS( BMG ) / PCL

```

```

DO 210 I = 1, NPHBR

```

```

          TEMP   = PRK( I ) * ( PHL - PRDST( I ) )

```

```

          XDEN   = XDEN + TEMP

```

210

```

          XNUM   = XNUM + TEMP * PRDST( I )

```

```

XNUM = - XNUM
CO 220 I = 1, NRDBR
TEMP = RBK(I) * ( A + RBDST(I) )
XDEN = XDEN + TEMP
220 XNUM = XNUM + TEMP * ( GC + RBDST(I) )
XNUM = XNUM - B * PFL
XDEN = B + XDEN
X = XNUM / XDEN
Y = GC - X
TETA = PCL / ( X + PHL ) * SIGN
F1 = TETA * ( RBKY + GC * SRBK - PRKX - X * SRPBK )
CALL GFORCE ( PRK, PRDST, TETA, X, NPHBR, FX )
CALL GFORCE ( RBK, RBDST, TETA, Y, NRDBR, FY )
D3 = DABS( TETA ) * X
D2 = ( Y + SBL ) * DABS( TETA )
IF ( ( D3 .GE. PCL ) .AND. ( D2 .GE. RCL ) ) GO TO 270
IF ( D3 .GE. PCL ) GO TO 400
IF ( D2 .GE. RCL ) GO TO 600
F2 = ZERO
F3 = ZERO
F4 = ZERO
KASE = 2

RETURN
27C TETA = PCL * TWO / PHL
D2 = ( PHL / TWO + GC + SBL ) * DABS( TETA )
IF ( D2 .GE. RCL ) GO TO 600
GO TO 40C

```

C

C---- >>> CASE 3: CONTACT AT FRONT FACE OF STUFFING BOX

C

```

300 XNUM = ZERO
XDEN = ZERO
A = SBL + GC
B = DABS( BMG ) / RCL
CO 310 I = 1, NPHBR
TEMP = PRK(I) * ( A + PRDST(I) )
XDEN = XDEN + TEMP
31C XNUM = XNUM + TEMP * PRDST(I)
XNUM = - XNUM
CO 320 I = 1, NRDBR
TEMP = RBK(I) * ( SBL - RBDST(I) )
XDEN = XDEN + TEMP
320 XNUM = XNUM + TEMP * ( GC + RBDST(I) )
XNUM = XNUM + B * A
XDEN = B + XDEN
X = XNUM / XDEN
Y = GC - X
TETA = RCL / ( Y + SBL ) * SIGN
F2 = TETA * ( X * SRPBK - RBKY - GC * SRBK + PRKX )
CALL GFORCE ( PRK, PRDST, TETA, X, NPHBR, FX )
CALL GFORCE ( RBK, RBDST, TETA, Y, NRDBR, FY )
D1 = ( X + PHL ) * DABS( TETA )
D4 = Y * DABS( TETA )
IF ( ( D1 .GE. PCL ) .AND. ( D4 .GE. RCL ) ) GO TO 370
IF ( D1 .GE. PCL ) GO TO 600
IF ( D4 .GE. RCL ) GO TO 500
F1 = ZERO
F3 = ZERO

```

```

      F4      = ZERO
      KASE    = 3
RETURN
37C      TETA  = RCL * TWO / SBL
      D1     = ( PHL + GC + SBL / TWO ) * DABS( TETA )
      IF ( D1 .GE. PCL ) GO TO 600
      GO TO 500
C
C----- >>> CASE 4: CONTACT AT FRONT AND BACK FACES OF PISTON HEAD
C
40C      X      = - PHL / TWO
      TETA     = TWO * PCL / PHL * SIGN
      Y      = GC - X
      CALL GFORCE ( PRK, PRDST, TETA, X, NPHBR,          FX )
      CALL GFORCE ( RBK, RBDST, TETA, Y, NRDBR,          FY )
      F1     = ZERO
      F3     = ZERO
      DO 470 I = 1, NPHBR
      F3     = F3 + FX(I)
47C      F1     = F1 + FX(I) * PRDST(I)
      DO 480 I = 1, NRDBR
      F3     = F3 - FY(I)
48C      F1     = F1 + FY(I) * ( GC + RBDST(I) )
      F1     = ( BMG - F1 ) / PHL
      F3     = F1 + F3
      F2     = ZERO
      F4     = ZERO
      KASE    = 4
RETURN
C
C----- >>> CASE 5: CONTACT AT FRONT AND BACK FACES OF STUFFING BOX
C
50C      X      = GC + SBL / TWO
      TETA     = TWO * RCL / SBL * SIGN
      Y      = GC - X
      CALL GFORCE ( PRK, PRDST, TETA, X, NPHBR,          FX )
      CALL GFORCE ( RBK, RBDST, TETA, Y, NRDBR,          FY )
      F2     = ZERO
      F4     = ZERO
      DO 570 I = 1, NPHBR
      F2     = - FX(I) * ( GC + PRDST(I) ) + F2
57C      F4     = - FX(I) + F4
      DO 580 I = 1, NRDBR
      F2     = F2 - FY(I) * RBDST(I)
58C      F4     = F4 + FY(I)
      F2     = ( BMG + F2 ) / SBL
      F4     = F2 + F4
      F1     = ZERO
      F3     = ZERO
      KASE    = 5
RETURN
C
C----- >>> CASE 6: CONTACT AT FRONT FACE OF PISTON HEAD AND FRONT FACE
C      OF STUFFING BOX
C
60C      TETA  = ( PCL + RCL ) / ( PHL + GC + SBL ) * SIGN
      X      = PCL / DABS( TETA ) - PHL
      Y      = GC - X

```

```

CALL GFORCE ( PRK, PRDST, TETA, X, NPHBR,      FX )
CALL GFORCE ( RBK, RBDST, TETA, Y, NRDBR,      FY )
      F1      = ZERO
      F2      = ZERO
DO 670 I = 1, NPHBR
      F1      = - FX(I) * ( SBL + GC + PRDST(I) ) + F1
670      F2      = FX(I) + F2
DO 68C I = 1, NRDBR
      F1      = F1 + FY(I) * ( SBL - RBDST(I) )
68C      F2      = - FY(I) + F2
      F1      = ( BMG + F1 ) / ( PHL + GC + SBL )
      F2      = F1 + F2
      F3      = ZERO
      F4      = ZERO
      KASE    = 6
      RETURN
690      TETA   = ZERO
      F1      = ZERO
      F2      = ZERO
      F3      = ZERO
      F4      = ZERO
CALL GFORCE ( PRK, PRDST, TETA, X, NPHBR,      FX )
CALL GFORCE ( RBK, RBDST, TETA, Y, NRDBR,      FY )
RETURN
END

```

```
      SUBROUTINE GFORCE
      I ( AK, DST, TETA, X, N,
      0 F )
C
C---- >>> SUBROUTINE TO CALCULATE FORCES ON EACH BEARING
C
      IMPLICIT REAL*8 ( A-H, G-Z )
      DIMENSION AK(N), DST(N), F(N)
C
      DO 10 I = 1, N
          F(I) = AK(I) * ( X + DST(I) ) * TETA
10 CONTINUE
      RETURN
      END
```

```

SUBROUTINE THCDS
I      ( CYK, RDK, CLT, RLT, CSSTF, RSSTF, TETA, W,
O      BB1, BB2, CK, RK, CK1L1, SK1L1, THC )
C
C----- >>> SUBROUTINE TO CALCULATE SLOPES AT SUPPORTS AND CONSTANTS IN
C          DEFLECTION EQUATIONS
C
      IMPLICIT REAL*8 ( A-H, O-Z )
      COMMON EXL, P
      COMMON / WGTVER / WC, WR
      COMMON / ECCTRI / ECLC, ECLR
      COMMON / CONSTS / EEL, FFL, AKTFC, AKTHR
      COMMON / CANDDS / C1, C2, D1, D2
      COMMON / FRCCNS / FCCY, FCRD, CCNM
      COMMON / REATNS / REC, RER
C
      DATA ZERO, ONE / 0.000, 1.000 /
C
      SQP      = DSQRT ( P )
      CK       = SQP / CYK
      RK       = SQP / RDK
      CK2      = CK * CK
      RK2      = RK * RK
      AK1L1    = CK * CLT
      AK2L2    = RK * RLT
      AK2L1    = RK * CLT
      AK2EL    = RK * EXL
      TK1L1    = DTAN( AK1L1 )
      SK1L1    = DSIN( AK1L1 )
      CK1L1    = DCOS( AK1L1 )
      TK2L2    = DTAN( AK2L2 )
      SK2L2    = DSIN( AK2L2 )
      CK2L2    = DCOS( AK2L2 )
      CK2L1    = DCOS( AK2L1 )
      TK2L1    = DTAN( AK2L1 )
      CK2EL    = DCOS( AK2EL )
      SK2EL    = DSIN( AK2EL )
      TK2EL    = DTAN( AK2EL )
      AKCCN    = CK / TK1L1 + RK / TK2L2
      BKCCN    = CK * TK2L2 + RK * TK1L1
      CKCCN    = RK - CK * TK1L1 * TK2L2
      DKCCN    = RK * TK1L1 * TK2L1 + CK
      TT       = TK1L1 * TK2L2
      SS       = SK1L1 * SK2L2
      CC       = CK1L1 * CK2L2
      ST       = SK1L1 * TK2L2
      CCEL     = CK2L2 * CK2EL
      TS       = TK1L1 * SK2L2
      WRPK2    = WR / ( P * RK2 )
      WCPK2    = WC / ( P * CK2 )
      WCWR     = WRPK2 - WCPK2
      AKCBP    = CSSTF / P
      AKRBP    = RSSTF / P
      EEL      = ( ECLR - ECLC ) / EXL
      FFL      = ( FCRD - FCCY ) / EXL
      BB1      = CONM / P + FCCY - ECLC - WCPK2
      BB2      = FCRD - ECLR - WRPK2
      BB3      = TETA + W / P

```

C
C---- >>> CALCULATE SLOPES AT CYLINDER AND ROD SUPPORTS
C

A11 = ONE - AKCBP * CK * CKCON / BKCON + AKCBP / EXL
 A12 = AKRBP * CK * RK / BKCON / CC - AKRBP / EXL
 B1 = BB1 * CKCCN / TT - WCWR * RK / ST
 B1 = B1 - BB2 * RK / SS + BB3 / SK1L1
 B1 = B1 * CK / AKCCN - EEL + FFL - REC / P
 A21 = - AKCBP * CK * RK / BKCON / CC + AKCBP / EXL
 A22 = ONE + AKRBP * CK2L1 * DKCON * RK / CCEL / BKCON
 A22 = A22 - AKRBP * RK * TK2EL - AKRBP / EXL
 B2 = BB1 * CK / CC - BB2 * DKCON * CK2L1 / CCEL
 B2 = B2 - WCWR * CK / CK2L2 - BB3 * TK1L1 / CK2L2
 B2 = B2 * RK / BKCCN + BB2 * RK * TK2EL
 B2 = B2 - EEL + FFL + RER / P
 THDEN = A11 * A22 - A12 * A21
 THC = (B1 * A22 - A12 * B2) / THDEN
 THR = (A11 * B2 - A21 * B1) / THDEN
 AKTHC = AKCBP * THC
 AKTHR = AKRBP * THR
 BB1 = BB1 + AKTHC
 BB2 = BB2 + AKTHR

C
C---- >>> CALCULATE CONSTANTS IN DEFLECTION EQUATIONS
C

C1 = -BB1
 C2 = BB1 * CKCON / TT - WCWR * RK / ST - BB2 * RK / SS
 C2 = (C2 + BB3 / SK1L1) / AKCON
 D2 = BB1 * CK * CK2EL / SS - BB2 * DKCON * CK2L1 / TS
 D2 = D2 - WCWR * CK * CK2EL / TS - BB3 * CK2EL / SK2L2
 D2 = D2 / AKCON
 D1 = (- D2 * SK2EL - BB2) / CK2EL

RETURN
END

```

SUBROUTINE STOPTB
  I ( TETA, NPHBR, NRDBR, ALTH, ALF, STROK, ITERAT, GC )
C
C---- >>> SLBROUTINE TO CALCULATE THE REQUIRED LENGTH OF STOP-TUBE
C
  IMPLICIT REAL*8 ( A-H, O-Z )
  COMMON / GLDFOR / FX(5), FY(5), F1, F2, F3, F4
C
  DATA TWO, HUNDRD / 2.0000, 100.0000 /
C
C---- >>> INCREMENT FOR STOP-TUBE LENGTH
C
      GCI      = STROK / HUNDRD
      TF       = DABS( F1 ) + DABS( F2 ) + DABS( F3 ) + DABS( F4 )
10  DO 10 I = 1, NPHBR
      TF       = TF + DABS( FX(I) )
20  DO 20 I = 1, NRDBR
      TF       = TF + DABS( FY(I) )
      F        = TF / TWO
C
C---- >>> CHECK TOTAL FORCE AND CROOKEDNESS ANGLE LIMITS
C
      IF ( F .LT. ALF .AND. DABS( TETA ) .LT. ALTH ) RETURN
      GC       = GC + GCI
      ITERAT   = 2
RETURN
END

```

```

      SUBROUTINE XATYMX
      I      ( CLT, CK, RK, DEFG,
      D      XCY, XRD, CSLPC, CSLPR )
C
C----- >>> SUBROUTINE TO CALCULATE THE DISTANCES AT WHICH MAXIMUM
C          DEFLECTIONS OCCUR
C
      IMPLICIT REAL*8 ( A-H, O-Z )
      COMMON EXL, P
      COMMON / WGTVER / WC, WR
      COMMON / CCNSTS / EEL, FFL, AKTHC, AKTHR
      COMMON / CANDDS / C1, C2, D1, D2
      COMMON / REATNS / REC, RER
C
      DATA ZERO, HUNDRD / 0.000, 100.000 /
C
C----- >>> INCREMENT FOR X
C
      AINCR = CLT / HUNDRD
      CSLP  = - EEL + FFL - ( AKTHC - AKTHR ) / EXL
      CSLPC = CSLP - REC / P
      CSLPR = CSLP + RER / P
      XCY   = CLT
      XRD   = CLT
C
C----- >>> POINT AT WHICH MAXIMUM DEFLECTION OCCURS IN CYLINDER
C
      10      CANG = CK * XCY
              CSLOP = - C1 * CK * DSIN( CANG ) + C2 * CK * DCOS( CANG )
              CSLCP = CSLOP + CSLPC + WC * XCY / P
              IF ( DEFG .LE. ZERO ) GO TO 20
              IF ( CSLCP .GT. ZERO ) GO TO 50
              GO TO 30
      20      IF ( CSLOP .LT. ZERO ) GO TO 50
      30      XCY = XCY - AINCR
              GO TO 10
C
C----- >>> POINT AT WHICH MAXIMUM DEFLECTION OCCURS IN ROD
C
      50      RANG = RK * XRD
              RSLCP = - D1 * RK * DSIN( RANG ) + D2 * RK * DCOS( RANG )
              RSLCP = RSLCP + CSLPR - WR * ( EXL - XRD ) / P
              IF ( DEFG .LE. ZERO ) GO TO 60
              IF ( RSLCP .LT. ZERO ) GO TO 100
              GO TO 70
      60      IF ( RSLCP .GT. ZERO ) GO TO 100
      70      XRD = XRD + AINCR
              GO TO 50
      100 RETURN
          END

```

```

SUBROUTINE YMAXS
  I ( XCY, XRD, CK, RK, CSLPC, CSLPP, BB1, BB2,
  O YCMAX, YRMAX )
C
C---- >>> SUBROUTINE TO CALCULATE MAX. DEFLECTIONS IN CYLINDER AND ROD
C
  IMPLICIT REAL*8 ( A-H, O-Z )
  COMMON EXL, P
  COMMON / WGTVER / WC, WR
  COMMON / CANDDS / C1, C2, D1, D2
C
  DATA TWO / 2.0000 /
C
      CANG = XCY * CK
      RANG = XRD * RK
      TWOP = TWO * P
      XRDP = EXL - XRD
C
C---- >>> MAXIMUM DEFLECTION IN CYLINDER
C
      YCMAX = C1 *DCOS( CANG )+C2 *DSIN( CANG ) + CSLPC * XCY
      YCMAX = YCMAX + BB1 + WC * XCY * XCY / TWOP
C
C---- >>> MAXIMUM DEFLECTION IN ROD
C
      YRMAX = D1 *DCOS( RANG ) + D2 *DSIN( RANG ) - CSLPR*XRDP
      YRMAX = YRMAX + BB2 + WR * XRDP * XRDP / TWOP
RETURN
END

```

SUBROUTINE STRCHS

```

I      ( KEYF, KEYT, KEYP, XCY, XRD, YCMAX, YRMAX, RID, CHDS,
I      OPPE, FYRODT, FYCYLT, PINCR1, PINCR2, KEYST, LFLUID,
O      HSC, HSR, AXTEN, CSTR, CSTRP, RSTR, RSTRP, CSS, NCSS, RSS, NRSS)

```

```

C
C---- >>> SUBROUTINE TO CHECK THE MAXIMUM STRESSES WITH THE LIMITING
C      STRESSES
C

```

```

IMPLICIT REAL*8 ( A-H, O-Z )

```

```

COMMON EXL, P

```

```

COMMON / WGTVER / WC, WR

```

```

COMMON / ECCTRI / ECLC, ECLR

```

```

COMMON / CONSTS / EEL, FFL, AKTHC, AKTHR

```

```

COMMON / FRCCNS / FCCY, FCRD, CCNM

```

```

COMMON / REATNS / REC, RER

```

```

COMMON / CRPROP / RDZ, CYZ, RDZI, CYZI, HSCCI, HSCCO, HSCRI,

```

```

*      HSCRO, BAREAC, BAREAR, CAREAC, CAREAR

```

```

DATA ZERO, TWO / 0.0000, 2.0000 /

```

```

DATA LYES / 3HYES /

```

```

C
C---- >>> CALCULATE BENDING MOMENTS AT MAXIMUM DEFLECTION POINTS
C

```

```

BMC1 = ( EEL - FFL + ( AKTHC - AKTHR ) / EXL ) * P * XCY

```

```

BMC  = BMC1 + REC * XCY - CCNM - P * FCCY - AKTHC * P

```

```

BMC  = DABS( BMC - WC * XCY * XCY / TWO + P * ( ECLC + YCMAX ) )

```

```

XRDP = EXL - XRD

```

```

BMR  = - BMC1 * XRDP / XCY + RER * XRDP - P * FCRD

```

```

BMR  = BMR - AKTHR * P - WR * XRDP * XRDP / TWO

```

```

BMR  = DABS( BMR + P * ( ECLR + YRMAX ) )

```

```

C
C---- >>> CALCULATE BENDING MOMENTS AT SUPPORTS
C

```

```

BMCP = DABS( - CCNM - P * ( FCCY + AKTHC - ECLC ) )

```

```

BMRP = DABS( - P * ( FCRD + AKTHR - ECLR ) )

```

```

C
C---- >>> CALCULATE ALL STRESSES
C

```

```

PRE  = P / BAREAC

```

```

HSC  = HSCCI * PRE

```

```

HSCC = HSCCO * PRE

```

```

HSR  = ZERO

```

```

IF ( LFLUID .EQ. LYES ) HSR = HSCRI * PRE

```

```

HSRC = ZERO

```

```

IF ( LFLUID .EQ. LYES ) HSRC = HSCRO * PRE

```

```

AXTEN = ZERO

```

```

IF ( CHDS .LT. ZERO ) AXTEN = P / CAREAC

```

```

CSTR = BMC / CYZ

```

```

CSTR1 = BMC / CYZI

```

```

CSTRP = BMCP / CYZ

```

```

CSTRPI = BMRP / CYZI

```

```

PR  = P

```

```

IF ( LFLUID .EQ. LYES ) PR = P - PRE * BAREAR

```

```

AXRST = PR / CAREAR

```

```

RSTR = BMR / RDZ + AXRST

```

```

RSTR1 = BMR / RDZI + AXRST

```

```

RSTRP = BMRP / RDZ + AXRST

```

```

RSTRPI = BMRP / RDZI + AXRST

```

```

      CSSPO = HSCO + CSTRP
      CSSPI = HSC + CSTRPI
      CSSC  = HSCO + CSTR
      CSSI  = HSC + CSTR
IF (CSSPC.GT.CSSPI.AND.CSSPO.GT.CSSO.AND.CSSPO.GT.CSSI)GO TO40
IF ( CSSPI .GT. CSSO .AND. CSSPI .GT. CSSI ) GO TO 30
IF ( CSSC .GT. CSSI ) GO TO 20
      CSS  = CSSI
      NCSS = 2
2C   GO TO 50
      CSS  = CSSO
      NCSS = 1
3C   GO TO 50
      CSS  = CSSPI
      NCSS = 2
4C   GO TO 50
      CSS  = CSSPO
      NCSS = 1
50   RSS  = ZERO
      NRSS = 0
IF ( LFLUID .NE. LYES ) GO TO 90
      RSSPO = HSR0 + RSTRP
      RSSPI = HSR + RSTRPI
      RSSC  = HSR0 + RSTR
      RSSI  = HSR + RSTR
IF (RSSPC.GT.RSSPI.AND.RSSPC.GT.RSSO.AND.RSSPO.GT.RSSI)GO TO80
IF ( RSSPI .GT. RSSO .AND. RSSPI .GT. RSSI ) GO TO 70
IF ( RSSC .GT. RSSI ) GO TO 60
      RSS  = RSSI
      NRSS = 2
6C   GO TO 90
      RSS  = RSSO
      NRSS = 1
7C   GO TO 90
      RSS  = RSSPI
      NRSS = 2
8C   GO TO 90
      RSS  = RSSPO
      NRSS = 1
90  CONTINUE
      IF ( KEYF .NE. 1 ) GO TO 500
      IF ( KEYST .NE. 1 ) GO TO 500
C----- >>> CHECK WITH LIMITING STRESSES
C
      IF ( RSTR .GT. FYRODT ) GO TO 100
      IF ( HSC .GT. FYCYLT ) GO TO 100
      IF ( (CSTR .GT. FYCYLT ) .OR. ( CSTRP .GT. FYCYLT ) )GO TO 100
      IF ( ( HSR .GT. FYRODT ) .OR. ( RSTRP .GT. FYRODT ) )GO TO 100
      IF ( ( CSS .GT. FYCYLT ) .OR. ( RSS .GT. FYRODT ) ) GO TO 100
      IF ( KEYT .EQ. 2 ) GO TO 400
C----- >>> CHANGE THE TRIAL LOAD CORRESPONDINGLY
C
      P      = P + PINCR1
RETURN
100  IF ( KEYP .EQ. 2 ) GO TO 200
      P      = P - PINCR1 + PINCR2

```

RELEASE 2.0

STRCHS

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```
      RETURN      KEYT = 2
200  RETURN      P     = P - PINCR2
      KEYF       = 2
      RETURN
400  P           = P + PINCR2
      KEYT       = 2
      KEYP       = 2
      RETURN
500  KEYF       = 3
      RETURN
      END
```

```

SUBRCUTINE OUTPUT
I      ( KWI, BAREAC, XCY, XRD, YCMAX, YRMAX, CHDS, GC, TETA,
I      FYCYLT, FYRODT, CSTR, CSTRP, RSTR, RSTRP, HSC, HSR, AXTEN,
I      NP+BR, NRDBR, THC, CSS, NCSS, RSS, NRSS )
C
C---- >>> SUBRCUTINE TO PRINT ALL THE RESULTS
C
      IMPLICIT REAL*8 ( A-H, O-Z )
      COMMON EXL, F
      COMMON / ID / IDCARD(40), NPROB, IPROB(19), LPRTP
      COMMON / UNITS / LNTU, LODU, LPREU, LANGU
      COMMON / ENDS / LCEND, LREND
      COMMON / FSOPTF / OPPRE, ALTH, ALF, FS, LFSTP
      COMMON / GLDFOR / FX(5), FY(5), F1, F2, F3, F4
C
      DATA ZERO, PI, H18C / 0.000, 3.141593000, 180.0000 /
      DATA LDEG, LFIX / 4HDEG , 3HFIX /, TWO / 2.0000 /
C
C---- >>> FORMATS
C
100 FORMAT ( 1H1, //, 5X, 36HPROGRAM SACREG - STRESS ANALYSIS OF ,
1          21HCYLINDERS ( REGULAR ), ///, 2( 7X, 20A4//), / )
110 FORMAT ( /, 5X, 8HPROBLEM , A4, //, 1X, 19A4 )
120 FORMAT ( //, 5X, 35HRESULTS:      CRITICAL LOAD ANALYSIS, / )
140 FORMAT ( //, 5X, 45HRESULTS:      ANALYSIS TO DETERMINE STOP-TUBE ,
1          6HLENGTH, / )
150 FORMAT ( //, 14X, 25HCRTICAL LOAD           = ,1PD10.3, 4X, A4,
1          //, 14X, 25HMAXIMUM FLUID PRESSURE = ,1PD10.3, 4X, A4,
2          //, 14X, 25HCROOKEDNESS ANGLE      = ,1PD12.5, 2X, A4, / )
160 FORMAT ( //, 14X, 25HOPERATING PRESSURE     = ,1PD10.3, 4X, A4,
1          //, 14X, 25HLOAD                     = ,1PD10.3, 4X, A4,
2          //, 14X, 25HCROOKEDNESS ANGLE      = ,1PD12.5, 2X, A4, / )
180 FORMAT ( //, 5X, 34HREQUIRED LENGTH OF STOP-TUBE = 1PD10.3, 4X, A4
1          //, 5X, 34HCORRESPONDING EXTENDED LENGTH = 1PD10.3, 4X, A4
2          //, 5X, 32HRESULTS WITH THIS STOP-TUBE ARE:, / )
130 FORMAT ( //, 5X, 44HRESULTS:      ANALYSIS FOR A GIVEN OPERATING ,
1          8HPRESSURE, / )
250 FORMAT ( 1H1, //, 5X, 40HANALYSIS AFTER APPLYING GIVEN FACTOR OF ,
1          10HSAFETY OF , F6.3, 2X, 2HON, 2X, A4, 2H: , / )
260 FORMAT ( //, 14X, 25HLOAD           = ,1PD10.3, 4X, A4,
1          //, 14X, 25HFLUID PRESSURE = ,1PD10.3, 4X, A4,
2          //, 14X, 25HCROOKEDNESS ANGLE = ,1PD12.5, 2X, A4, / )
300 FORMAT ( //, 5X, 9HCYLINDER:,
1          //, 10X, 29HMAXIMUM DEFLECTION           = ,1PD10.3, 4X, A4,
2          //, 10X, 29HMAXIMUM LONGITUDINAL STRESS = ,1PD10.3, 4X, A4,
3          //, 10X, 29HAT A DISTANCE FROM CYL SUP = ,1PD10.3, 4X, A4,
4          //, 10X, 29HFACTOR OF SAFETY ON CYL     = ,1PD10.3, / )
310 FORMAT ( //, 10X, 29HMAX LONG STRESS AT CYL END = ,1PD10.3, 4X, A4,
1          //, 10X, 29HFACTOR OF SAFETY ON CYL     = ,1PD10.3, / )
312 FORMAT ( //, 10X, 29HMAX SHEAR STRESS IN CYL   = ,1PD10.3, 4X, A4,
1          /, 10X, 31HOCCURS AT MAX LONG STRESS POINT )
314 FORMAT ( 10X, 21HAND AT OUTER SURFACE )
316 FORMAT ( 10X, 21HAND AT INNER SURFACE )
318 FORMAT ( /, 10X, 29HFACTOR OF SAFETY ON CYL   = ,1PD10.3, / )
320 FORMAT ( //, 10X, 29HMAXIMUM HOOP STRESS IN CYL = ,1PD10.3, 4X, A4,
1          //, 10X, 29HFACTOR OF SAFETY ON CYL     = ,1PD10.3, / )
325 FORMAT ( //, 10X, 29HAXIAL TENSION IN OVER HANG = ,1PD10.3, 4X, A4,
1          //, 10X, 29HFACTOR OF SAFETY ON CYL     = ,1PD10.3,

```

```

2
330 FORMAT ( //, 10X, 29HEND DEFLECTION IN OVERHANG = ,1PD10.3, 4X, A4 )
1 //, 5X, 9HROD :
2 //, 10X, 29HMAXIMUM DEFLECTION = ,1PD10.3, 4X, A4,
3 //, 10X, 29HMAXIMUM LONGITUDINAL STRESS= ,1PD10.3, 4X, A4,
4 //, 10X, 29HAT A DISTANCE FROM CYL SUP = ,1PD10.3, 4X, A4,
340 FORMAT ( //, 10X, 29HFACTOR OF SAFETY ON ROD = ,1PD10.3, / )
1 //, 10X, 29HMAX LONG STRESS AT ROD END = ,1PD10.3, 4X, A4,
342 FORMAT ( //, 10X, 29HFACTOR OF SAFETY ON ROD = ,1PD10.3, / )
1 //, 10X, 29HMAX SHEAR STRESS IN ROD = ,1PD10.3, 4X, A4,
348 FORMAT ( /, 10X, 31HOCCURS AT MAX LONG STRESS POINT )
350 FORMAT ( //, 10X, 29HFACTOR OF SAFETY ON ROD = ,1PD10.3, / )
1 //, 10X, 29HMAXIMUM HOCP STRESS IN ROD = ,1PD10.3, 4X, A4,
360 FORMAT ( 1H1, //, 5X, 29HFORCES AT SLIDING CONNECTION:, //, 15X,
1 23HPISTON BEARINGS (SEALS), 6X, 5HFORCE, /, 25X, 2HNO, / )
370 FORMAT ( /, 5( 25X, 12, 12X, 1PD10.3, 2X, A4, // ) )
380 FCRMAT ( /, 18X, 20HROD BEARINGS (SEALS), 6X, 5HFORCE, /, 25X, 2HNO/)
390 FORMAT ( //8X38HF1- FORCE AT PISTON HEAD FRONT FACE =1PD11.3, 2XA4
1 //8X38HF2- FORCE AT STUFFING BOX FRONT FACE =1PD11.3, 2XA4
2 //8X38HF3- FORCE AT PISTON HEAD EACH FACE =1PD11.3, 2XA4
3 //8X38HF4- FORCE AT STUFFING BOX INNER FACE =1PD11.3, 2XA4
4 /, 11X, 33H(ZERO FORCES INDICATE NO CONTACT) , / )
400 FORMAT ( ///, 10X, 39HTHETA EQUAL TO ZERO IMPLIES CONTINUOUS ,
1 17HCONTACT AT GLAND. , /, 10X,
2 34HABOVE FORCES CANNOT BE CALCULATED.
3 33HHENCE, ARE PRINTED OUT AS ZERO'S. , // )

```

C

```

TEMP = H180 / PI
IF ( LANGU .EQ. LDEG ) TETA = TETA * TEMP
PRE = P / BAREAC
IF ( KWIT .NE. 1 ) GO TO 540
PRINT 100, ( IDCARD(I), I = 1, 40 )
PRINT 110, NPROB, ( IPROB(I), I = 1, 19 )
GO TO ( 510, 520, 530 ), LPRTP
510 PRINT 120
PRINT 150, P, LODU, PRE, LPREU, TETA, LANGU
GO TO 550
520 PRINT 130
PRINT 160, OPRE, LPREU, P, LODU, TETA, LANGU
GO TO 550
530 PRINT 140
PRINT 180, GC, LNTU, EXL, LNTU
PRINT 150, P, LODU, OPRE, LPREU, TETA, LANGU
GO TO 550
540 PRINT 250, FS, LFSTP
PRINT 260, P, LODU, PRE, LPREU, TETA, LANGU

```

C

```

C---- >>> CALCULATE THE FACTOR OF SAFETY'S ON MAXIMUM STRESSES WITH
C LIMITING STRESSES
C

```

```

550 FCSF = FYCYLT / CSTR
PRINT 300, YCMAX, LNTU, CSTR, LPREU, XCY, LNTU, FCSF
IF ( CSTRP .GT. ZERO ) FCSF = FYCYLT / CSTRP
IF ( LCEND .EQ. LFIX ) PRINT 310, CSTRP, LPREU, FCSF
FCSF = FYCYLT / CSS
CSS = CSS / TWO
PRINT 312, CSS, LPREU
IF ( NCSS .EQ. 1 ) PRINT 314

```

```

      IF ( NCSS .EQ. 2 ) PRINT 316
PRINT 318, FCSF
      FCSF = FYCYLT / HSC
PRINT 320, HSC, LPREU, FCSF
      IF ( CHDS .GE. ZERO ) GO TO 560
      FCSF = FYCYLT / AXTEN
      ENDDF = THC * CHDS
PRINT 325, AXTEN, LPREU, FCSF, ENDDF, LNTU
56C   FCSF = FYRODT / RSTR
PRINT 330, YRMAX, LNTU, RSTR, LPREU, XRD, LNTU, FCSF
      IF ( RSTRP .GT. ZERO ) FCSF = FYRODT / RSTRP
      IF ( LREND .EQ. LFIX ) PRINT 340, RSTRP, LPREU, FCSF
      IF ( HSR .EQ. ZERO ) GO TO 570
      FCSF = FYRODT / RSS
      RSS = RSS / TWO
PRINT 342, RSS, LPREU
      IF ( NRSS .EQ. 1 ) PRINT 314
      IF ( NRSS .EQ. 2 ) PRINT 316
PRINT 348, FCSF
      FCSF = FYRODT / HSR
PRINT 350, HSR, LPREU, FCSF
570  PRINT 360
PRINT 370, ( I, FX(I), LODU, I = 1, NPHBR )
PRINT 380
PRINT 370, ( I, FY(I), LODU, I = 1, NRDBR )
PRINT 390, F1, LODU, F2, LODU, F3, LODU, F4, LODU
      IF ( TETA .EQ. ZERO ) PRINT 400
RETURN
END

```

APPENDIX F

PROTECTION OF LIGHT-WEIGHT HYDRAULIC PROPS AGAINST ACCIDENTAL DAMAGE BY LATERAL LOADING

Since the cylinder and rod of the all hydraulic prop are long in comparison to their diameter stress, calculations for lateral loadings will be performed assuming that their lengths are infinite. The results will be conservative since the effect of end caps is to strengthen the assembly. Figure F-1 shows a concentrated lateral load acting on a small circular area of a cylinder. Such a load will result in longitudinal bending stresses (σ_l), as well as circumferential bending stresses (σ_c). These stresses vary with distance from the load and radial location of the point. Formulas for maximum stresses have been given by Roark [35] as follows:

Longitudinal bending stress (σ_l) =

$$\frac{P}{t^2} \left[0.42 \lambda \eta \frac{0.215R}{b} + \frac{6\nu}{4\pi} \right] \quad (1)$$

Circumferential bending stress (σ_c) =

$$\frac{P}{t^2} \left[0.42 \lambda \eta \frac{0.215R}{b} + \frac{6}{4\pi} \right] \quad (2)$$

Where: P = concentrated load

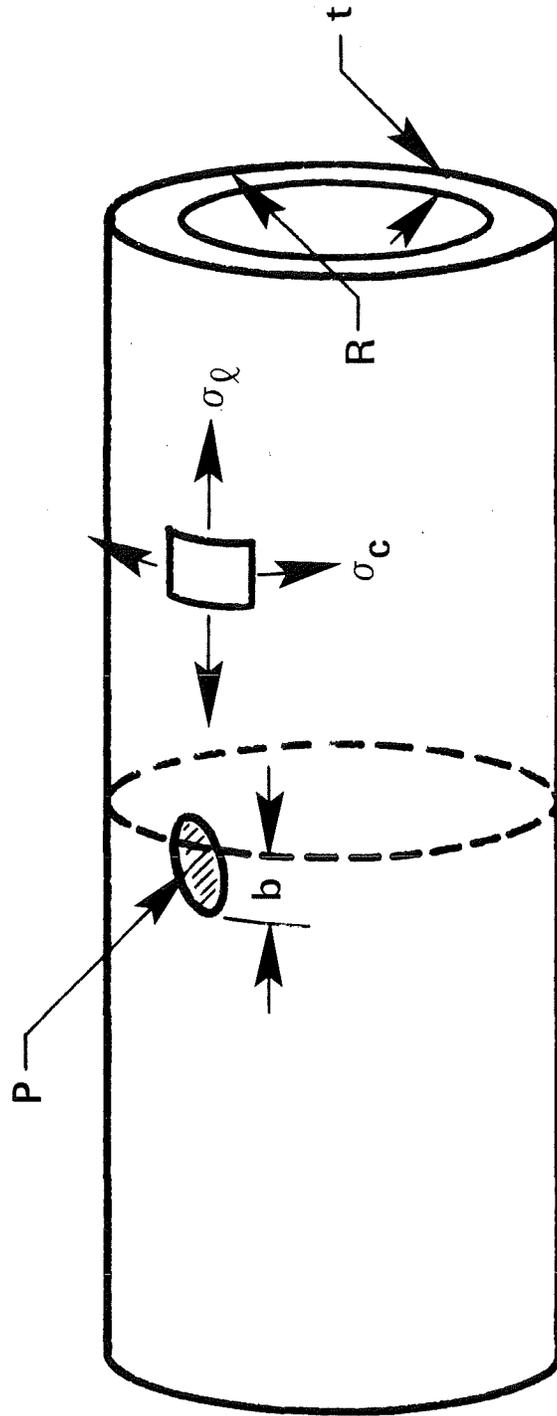
b = radius of area on which concentrated load acts

t = thickness of cylinder

R = mean radius of cylinder

ν = Poisson's ratio

Since Poisson's ratio is less than one for most materials σ_c is greater than σ_l .



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FIGURE F-1
LATERAL LOADING OF CYLINDER

Equation two can be rewritten to calculate the minimum wall thickness of a given cylinder so as not to exceed a prescribed stress, as follows:

$$t = \left[\frac{P(0.42 \ln \frac{0.215R}{b} + \frac{6}{4\pi})}{\sigma_c} \right]^{\frac{1}{2}} \quad (3)$$

Lateral loads on mine props may occur due to placing of objects on the prop, running wheeled or tracked vehicles over them. There are no "design" loads to such occurrences, and any numbers value assigned to them is only an estimate. A value of 1000 lbf (4.45 kN) is taken for the following calculations, which are performed for steel, Al-alloy and graphite-aramid composite props.

(a) Steel Prop

$$R = 3.15" \text{ (80 mm)}$$

$$b = 0.1" \text{ (2.5 mm)}$$

$$\sigma_c = 170,000 \text{ psi (1172 MPa)}$$

$$t = \left[\frac{1000 (0.42 \ln \frac{0.215 \times 3.15}{0.1} + \frac{6}{4\pi})}{170,000} \right]^{\frac{1}{2}}$$

$$= 0.087" \text{ (2.21 mm)}$$

(b) Al-alloy prop

$$R = 3.35" \text{ (85 mm)}$$

$$b = 0.1" \text{ (2.5 mm)}$$

$$\sigma_c = 72,000 \text{ psi (496 MPa)}$$

$$t = \left[\frac{1000 (0.42 \ln \frac{0.215 \times 3.35}{0.1} + \frac{6}{4\pi})}{72,000} \right]^{\frac{1}{2}}$$

$$= 0.135" \text{ (3.42 mm)}$$

(c) Aramid-graphite Composite

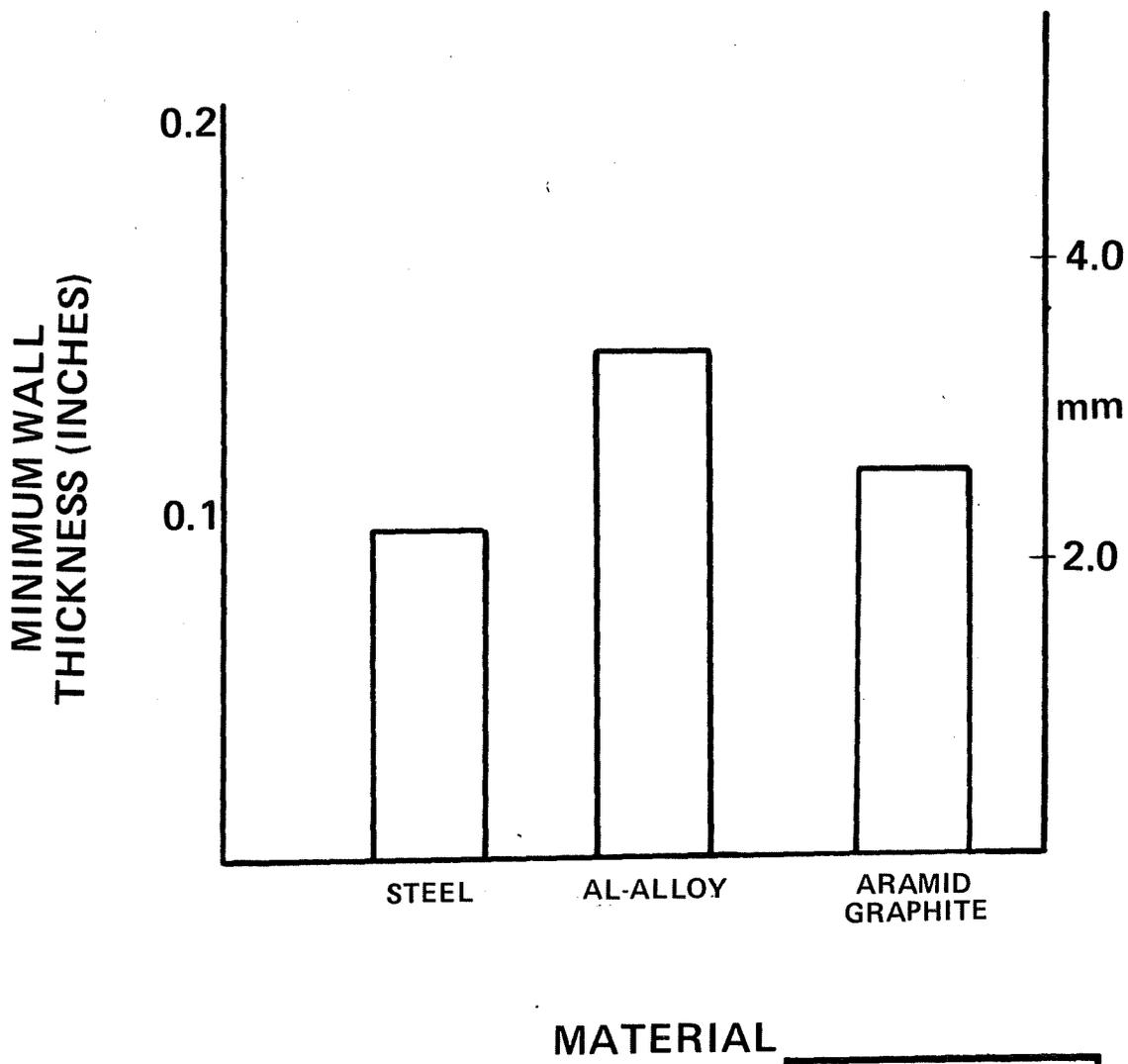
$$R = 2.56'' \text{ (65 mm)}$$

$$b = 0.1'' \text{ (2.5 mm)}$$

$$\sigma_c = 112,000 \text{ psi (772 MPa)}$$

$$t = \left[\frac{1000 \left(0.42 \times \frac{0.215 \times 2.56}{0.1} + \frac{6}{4\pi} \right)}{112,000} \right]^{\frac{1}{2}}$$
$$= 0.103'' \text{ (0.26 mm)}$$

These values of wall thickness should be maintained even though other considerations may be met by using lesser values. Figure F-2 shows the wall thickness for steel, Al-alloy and aramid-graphite.



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FIGURE F-2
 MATERIAL THICKNESS FOR LATERAL STRENGTH

APPENDIX G

OVERLOAD PROTECTION IN MECHANICAL PROPS

One of the problems with mechanical props, whether of the post and wedge design or the screw type, is that they do not yield appreciably on overloading. Such overloading may occur during setting up, or subsequently due to roof sag or floor heave. In hydraulic props the provision of a relief valve ensures that the jack does not become over-pressurized.

In mechanical designs, yielding at a prescribed load, and at a given rate, can be provided, theoretically at least, by suitably preloaded springs. Initially it was visualized that a coil spring could be incorporated in the top tube of a telescopic-tube prop and preloaded to the rated capacity, i.e., 196 kN. A yield of 10 mm corresponding to an overload of 196 kN was considered a reasonable design objective. Hence the coil spring would have to have a rate of 19.6 kN/mm. On using spring design charts [36], it was found that a coil spring of 25 mm wire diameter and 100 mm mean coil diameter and three turns would have a capacity of only 35.6 kN and deflect approximately 20 mm. Such a spring would weigh 3.8 kg. Higher stiffness would necessitate larger wire and coil diameters and would weigh proportionately more. Mean coil diameters greater than 100 mm are considered unsuitable since they would not fit inside the telescopic tubes. The conclusions drawn from the above calculations is that coil springs are unsuitable for overload protection in lightweight mechanical props.

The next arrangement investigated was that of Belleville springs

which are used where large load capacity, small deflection and a limited closed height are constraining factors [36]. The design parameters for Belleville springs are

- (i) Outside diameter (D_o),
- (ii) Inside Diameter (D_i),
- (iii) Dish angle (α), and
- (iv) Spring thickness (t).

The dish angle is constrained to values between 4° and 6° , and the outside diameter to be between $12t$ and $24t$ [36]. Increasing the spring thickness and decreasing the ratio D_o/D_i , increases the load capacity but reduces the maximum deflection. After considerable trial and error, using the charts in reference 36, the following dimensions were obtained.

$$D_o = 127 \text{ mm}$$

$$D_i = 76.2 \text{ mm}$$

$$t = 10.7 \text{ mm}$$

$$\alpha = 4^\circ$$

$$\text{Number of springs} = 33$$

$$\text{Material} = \text{SAE 6145}$$

$$\text{Weight} = 21.92 \text{ kg}$$

$$\text{Height} = 378 \text{ mm}$$

$$\text{Free solid height} = 353 \text{ mm}$$

$$\text{Load capacity} = 473 \text{ kN}$$

It is seen that not only does the spring take up considerable volume, but it also weighs a substantial amount.

The liquid spring differs from conventional solid springs in utilizing the compressibility of the material rather than flexural resistance. As

explained in reference 37, liquid springs are more compact than coil springs and Belleville springs of the same capacity.

The following design parameters were used to size a liquid spring for the mechanical props, in conjunction with the charts in reference 36.

Load Capacity (W) = 390 kN

Desired deflection (δ) = 25.4 mm

Fluid = Dow Corning F4029 or equivalent

Inside Diameter of Container (D_i) = 76.2 mm

Allowable pressure (p) = 85.5 MPa

Desired Compression (ΔV) = 8.8%

Fluid Volume, V, is given by

$$V = \frac{\delta \frac{\pi}{4} (D_i)^2}{\Delta V}$$

$$\begin{aligned} \text{I.e., } V &= \frac{0.0254 \times \frac{\pi}{4} (0.0762)^2}{0.088} \\ &= 1.132 \times 10^{-3} \text{ m}^3 \end{aligned}$$

The length of the liquid column, l, is given by:

$$\begin{aligned} L &= \frac{V}{\frac{\pi}{4} (D_i)^2} \\ &= \frac{1.132 \times 10^{-3}}{\frac{\pi}{4} (0.0762)^2} \\ &= 0.289 \text{ m} \end{aligned}$$

Assuming that the material of the pressure vessel is steel of a tensile strength σ , of 416 MPa, the outer diameter D_o , is given by:

$$D_o = D_i \left[\frac{\frac{\sigma}{p} + 1}{\frac{\sigma}{p} - 1} \right]$$

$$= 76.2 \left[\frac{\frac{416 \times 10^6}{85.5 \times 10^6} + 1}{\frac{416 \times 10^6}{85.5 \times 10^6} - 1} \right]$$

$$= 94 \text{ mm}$$

Assuming a material density of 8016 kg/m^3 , the container mass can be calculated as:

$$m_c \triangleq \frac{\pi}{4} (D_o^2 - D_i^2) L \times 8016$$

$$= \frac{\pi}{4} (0.094^2 - 0.0762^2) \times 0.289$$

$$\qquad \qquad \qquad \times 8016$$

$$= 5.5 \text{ kg}$$

The rod mass, m_r , is estimated by assuming it to be solid and 100 mm long, as follows:

$$m_r = \frac{\pi}{4} (0.0762^2) \times 0.1 \times 8016$$

$$= 3.17 \text{ kg}$$

Mass of fluid in spring, using a density of 900 kg/m^3 is obtained as:

$$m_f \triangleq = 1.132 \times 10^{-3} \times 900$$

$$= 1.2 \text{ kg}$$

Total mass of spring is 10.4 kg.

Reducing the volume of a liquid spring raises its operating pressure and capacity. Consider a spring made to the following specifications:

$$D_i = 57.15 \text{ mm}$$

$$p = 153 \text{ MPa}$$

$$\Delta V = 12.8 \%$$

Other parameters are as given earlier. Using the same formulas, the dimensions and weights are obtained to be:

$$V = 0.51 \times 10^{-3} \text{ m}^3$$

$$L = 0.199 \text{ m}$$

$$D_o = 70.4 \text{ mm}$$

$$m_c = 2.1 \text{ kg}$$

$$m_r = 1.6 \text{ kg}$$

$$m_f = 0.46 \text{ kg}$$

Total mass of spring is 4.16 kg.

Thus, a 60% reduction in mass has been achieved by raising the pressure by 80%. Further reductions in size might jeopardize the design by leading to (a) sealing problems and (b) thermal problems.

It is concluded that the liquid spring is a viable option for providing mechanical props with yielding action.

TABLE H-1
NOMENCLATURE AND DIMENSIONS

SYMBOL	DESCRIPTION	VALUE	
		S. I. UNITS	U. S. UNITS
F	Critical Buckling Load	783.5 kN	176 kips
E	Youngs Modulus		
	(a) Steel (SAE 1040)	106.9 GPa	30,000 ksi
	(b) Al Alloy (7075-T6)	68.97 GPa	10,000 ksi
A	Cross-sectional Area	sq mm	sq ins
L	Length	2.44 m	96 in
k	Radius of gyration	mm	ins
D_o	Outer Diameter of cylinder	mm	ins
D_i	Inner diameter of cylinder	mm	ins
M	Mass/unit length	kg/mm	lb/in
ρ_m	Density of metal		
	(a) Steel	kg/m ³	0.284 lb/in ³
	(b) Al Alloy	kg/m ³	0.1 lb/in ³
ρ_f	Density of fluid	kg/m ³	0.035 lb/in ³
I	Moment of inertia	mm ⁴	in ⁴
σ	Ultimate tensile strength		
	(a) Steel (SAE 1040)	689.7 MPa	100 ksi
	(b) Al Alloy (7075-T6)	496.6 MPa	72 ksi
P	Internal pressure	MPa	ksi
F_h	Maximum load for internal pressurization	kN	kips

APPENDIX H

WEIGHT OPTIMIZATION OF ALL-HYDRAULIC PROPS

The following assumptions will be made for analysis:

1. The prop is a uniform column of the same size as the cylinder portion, for buckling failure analysis. It is kept vertical, has pinned ends and is loaded concentrically.
2. The prop is treated as a uniform thick cylinder for over-pressurization failure.
3. The weight of the prop in the fully-extended position is obtained by treating it as a uniform cylinder filled with fluid. End caps and fittings are neglected in weight estimation.

Table H-1 gives the symbols and material properties used in the analysis.

The Euler formula for critical buckling load for a pinned uniform column loaded concentrically is

$$F = \frac{\pi^2 EA}{(L/k)^2} \quad (H-1)$$

We will design the prop for a load of 22 tons (195.9 kN) and a factor of safety of four; i.e., the maximum load is 176 kips (783.5 kN).

For a hollow cylinder

$$k = \sqrt{\frac{D_o^2 + D_i^2}{16}} \quad (H-2)$$

and

$$A = \pi/4(D_o^2 - D_i^2) \quad (H-3)$$

giving

$$F = \frac{\pi^3 E (D_o^4 - D_i^4)}{64L^2} \quad (H-4)$$

The mass of the prop per unit length is given by

$$M = \pi/4 \rho_m (D_o^2 - D_i^2) + \pi/4 \rho_f (D_i^2) \quad (H-5)$$

Rewrite equation H-4 to obtain

$$D_o^4 = \frac{64FL^2}{\pi^3 E} + D_i^4$$

Let

$$z = D_o^2 = \sqrt{\frac{64FL^2}{\pi^3 E} + D_i^4} \quad (H-6)$$

Also let

$$D_i^2 = y \quad \& \quad \alpha = \frac{64FL^2}{\pi^3 E}$$

Substituting for D_o and D_i in H-5:

$$M = \pi/4 \left[\rho_m \left\{ \sqrt{\alpha + y^2} - y \right\} + \rho_f y \right] \quad (H-7)$$

Minimizing M with respect to y:

$$\frac{dM}{dy} = \frac{\pi}{4} \left[\rho_m \left\{ \frac{y}{\sqrt{\alpha + y^2}} - 1 \right\} + \rho_f \right] = 0 \quad (H-8)$$

Let $\rho_m/\rho_f = B$. Then

$$B \left\{ \frac{y}{\sqrt{\alpha + y^2}} - 1 \right\} + 1 = 0$$

or

$$\frac{By}{\sqrt{\alpha + y^2}} = B - 1$$

or

$$\left(\frac{By}{B-1} \right)^2 = \alpha + y^2 \quad (H-9)$$

From equation H-9 we get

$$y = \sqrt{\frac{\alpha(B-1)^2}{2B-1}} \quad (\text{H-10})$$

CASE A

For steel: $B = \frac{0.284}{0.035} = 8.11$

$$\alpha = \frac{64.176(96)^2}{\pi^3 30\,000} = 111.6$$

Therefore $y = \sqrt{\frac{\alpha(B-1)^2}{2B-1}} = \sqrt{\frac{111.6(7.11)^2}{15.22}} = 19.25$ (H-11)

Therefore $D_i = \sqrt{y} = 4.388 \text{ ins (111.5 mm)}$ (H-12)

$$\begin{aligned} D_o^2 &= \sqrt{\alpha + y^2} \\ &= 21.96 \end{aligned} \quad (\text{H-13})$$

Therefore $D_o = 4.686 \text{ ins (119.0 mm)}$ (H-14)

The maximum hoop stress is given by:

$$\sigma = P \frac{D_o^2 + D_i^2}{D_o^2 - D_i^2} = P \left(\frac{z + y}{z - y} \right) \quad (\text{H-15})$$

Giving $P = \left(\frac{21.96 - 19.25}{21.96 + 19.25} \right) 100$
 $= 6.576 \text{ ksi (45.36 MPa)}$ (H-16)

Consequently, the cylinder would support a load of:

$$\begin{aligned} L_p &= \frac{\pi}{4} (D_i^2) P \\ &= \frac{\pi}{4} (19.25) 6.576 \\ &= 99.4 \text{ kips (442 kN)} \end{aligned} \quad (\text{H-17})$$

which is less than the desired value.

It is concluded that internal pressurization is more of a limiting factor than buckling failure.

Mass of the cylinder is:

$$\begin{aligned} M &= \frac{\pi}{4} [0.284(2.70) + 0.035(19.25)] \\ &= 1.13 \text{ lbs/in } (20.18 \text{ kg/m}) \end{aligned} \quad (\text{H-18})$$

Hence, the prop would weigh 108.5 lbs (49.32kg). Dry weight is 57.82 lbs (26.3kg)

CASE B

For Aluminum Alloy: $B = \frac{0.1}{0.035} = 2.857$

$$\alpha = \frac{64.176 (96)^2}{\pi^3 10,000} = 334.8$$

Therefore, $y = \sqrt{\frac{\alpha (B - 1)^2}{2B - 1}}$

or $y = \sqrt{\frac{334.8 (1.857)^2}{4.714}} = 15.65 \quad (\text{H-19})$

$$D_i = \sqrt{y} = 3.956'' (100.5 \text{ mm}) \quad (\text{H-20})$$

$$\begin{aligned} z &= \sqrt{\alpha + y^2} \\ &= \sqrt{334.8 + 244.9} = 24.08 \end{aligned}$$

$$D_o = x = 4.907" (124.6 \text{ mm}) \quad (\text{H-21})$$

The maximum allowable internal pressure is given by:

$$\begin{aligned} p &= \sigma \left(\frac{z - y}{z + y} \right) \\ &= 100 \left(\frac{24.08 - 15.65}{24.08 + 15.65} \right) \\ &= 21.2 \text{ ksi (146.3 MPa)} \end{aligned}$$

$$\begin{aligned} \text{This cylinder would support a load of: } L_p &= \frac{\pi}{4} (D_i^2) p \\ &= \frac{\pi}{4} (15.65)^2 21.2 \\ &= 260.6 \text{ kips (1160 kN)} \quad (\text{H-22}) \end{aligned}$$

which is more than the desired value.

It is concluded that in this case buckling failure and not internal pressure is the limiting factor.

$$\begin{aligned} \text{Mass of the cylinder is: } M &= \frac{\pi}{4} \left[0.1(8.430) + 0.035(15.65) \right] \\ &= 1.092 \text{ lbs/in (18.97 kg/m)} \quad (\text{H-23}) \end{aligned}$$

Hence, the prop would weigh 104.9 lbs (47.7 kg). Dry weight is 63.56 lbs (28.9 kg).

Even though the weight estimates obtained in these two cases may be in some error, the analysis is still useful in illustrating that the mode of failure depends on the relative values of the ultimate tensile strength and the Young's Modulus. Leaving aside the question of minimizing weight

for the moment, we can establish the relationships between dimensions so that the prop is equally strong against internal pressurization and buckling.

I.e., $F = F_p$ or

$$\frac{\pi^3 E (D_o^4 - D_i^4)}{64 L^2} = \frac{\pi}{4} D_i^2 \sigma \left(\frac{D_o^2 - D_i^2}{D_o^2 + D_i^2} \right) \quad (H-24)$$

Let $D_o^2 = z$ and $D_i^2 = y$, so that

$$\frac{\pi^2 E}{16 L^2} (z^3 - y^2) = y \sigma \left(\frac{z - y}{z + y} \right) \text{ or } a = \sqrt{\gamma y - y} \quad (H-25)$$

Where:

$$\gamma = \frac{16 \sigma L^2}{\pi^2 E} \quad (H-26)$$

Note that $4y < \gamma$ since z has to be greater than y .

For Steel:

$$\begin{aligned} \gamma &= \frac{16 \times 100 \times 96^2}{\pi^2 30,000} \\ &= 49.8 \end{aligned}$$

$$\text{I.e., } D_i^2 < \frac{49.8}{4}$$

or

$$D_i < 3.528" \text{ (89.6 mm)}$$

Since this is larger than the value established in Case A., i.e., 4.388 inches (111.5 mm), we infer that using steel, the prop cannot be made equally strong against buckling and internal pressurization.

For Aluminum Alloy:

$$\gamma = \frac{16 \times 72 \times 96^2}{\pi^2 10,000} = 107.6$$

I.e.,
$$D_i^2 < \frac{107.6}{4}$$

or
$$D_i < 5.187" \text{ (131.7 mm)}$$

Let $D_i = 3.956" \text{ (100.5 mm)}$, i.e., the value obtained in Case B.

Then
$$z = \sqrt{107.6(3.956)^2 - (3.956)^2}$$

$$= 25.39$$

I.e.,
$$D_o = \sqrt{25.39} = 5.038" \text{ (128 mm)}$$

The load supported by a prop of this dimensions would be:

$$L_b = L_h = \frac{\pi}{4} D_i^2 \sigma \left(\frac{D_o^2 - D_i^2}{D_o^2 + D_i^2} \right)$$

$$= \frac{\pi}{4} (15.65) 72 \left(\frac{25.39 - 15.65}{25.39 + 15.65} \right)$$

$$= 210 \text{ kips (935 kN)}$$

with an internal pressure of 17.09 ksi (117.9 MPa).

Such a prop would weigh:

$$W = \frac{\pi}{4} \times 0.1(9.74) - \frac{\pi}{4} \times 0.035(15.65)$$

$$= 1.195 \text{ lbs/in (21.34 kg/m)}$$

The prop would weigh 114.7 lbs (51.2 kg). Dry weight would be 73.4 lbs (33 kg).

We conclude from the above analysis that the minimum weight aluminum alloy prop is close to optimal in making the best use of both the tensile strength and the material stiffness.

Since the steel prop, Case A, has a low capacity due to internal

pressurization, we can attempt use on stronger material.

If the steel prop of $D_i = 4.388''$ is to support a load of 176 kips (783.5 kN), the pressure is given by:

$$P = \frac{176}{\frac{\pi}{4} D_i^2} = \frac{176}{\frac{\pi}{4} \times 19.25}$$

or

$$P = 11.64 \text{ ksi (79.06 MPa)}$$

If $D_o = 4.686''$ (119.0 mm), the maximum hoop stress is given by:

$$\begin{aligned} \sigma &= P \frac{D_o^2 + D_i^2}{D_o^2 - D_i^2} \\ &= 11.64 \frac{21.96 + 19.25}{21.95 - 19.25} \\ &= 177 \text{ ksi (1220.8 MPa)} \end{aligned}$$

This value of tensile strength is not beyond the capability of high strength alloy steels. Hence, by using such material, a steel prop of approximately the same capacity and weight of the aluminum alloy prop can be fabricated.

The above conclusions are subject to the validity of the assumptions made at the beginning of the analysis. The truly optimal dimensions and the minimum weight are expected to be "near" the values obtained above.