AD 33061

Ext. Copy I

103

UNCLASSIFIED

THEORETICAL INVESTIGATION OF OPTIMUM PRESSURES IN AIRCRAFT HYDRAULIC SYSTEMS

Conrad H. Cooke Eugene Gessner Robert L. Smith

THE GLENN L. MARTIN COMPANY

January 1954

WRIGHT AIR DEVELOPMENT CENTER

NOTICES

When Government drawings, specifications, or other data are used for any purpose other than in connection with a definitely related Government procurement operation, the United States Government thereby incurs no responsibility nor any obligation whatsoever; and the fact that the Government may have formulated, furnished, or in any way supplied the said drawings, specifications, or other data, is not to be regarded by implication or otherwise as in any manner licensing the holder or any other person or corporation, or conveying any rights or permission to manufacture, use, or sell any patented invention that may in any way be related thereto.

The information furnished herewith is made available for study upon the understanding that the Government's proprietary interests in and relating thereto shall not be impaired. It is desired that the Judge Advocate (WCJ), Wright Air Development Center, Wright-Patterson Air Force Base, Ohio, be promptly notified of any apparent conflict between the Governments proprietary interests and those of others.

WADC TECHNICAL REPORT 54-189
VOLUME I

THEORETICAL INVESTIGATION OF OPTIMUM PRESSURE IN AIRCRAFT HYDRAULIC SYSTEMS

Conrad H. Cooke - Eugene Gessner - Robert L. Smith
The Glenn L. Martin Company

January 1954

Aircraft Laboratory
Contract No. AF 33(616)-344
RDO No. R-452-495

Wright Air Development Center
Air Research and Development Command
United States Air Force
Wright-Patterson Air Force Base, Ohio

FOREWARD

This report is prepared by The Glenn L. Martin Company for Contract No. AF 33(616)-344, Project No. 53-610A-10. Work was begun in February 1953 and completed in January 1954.

This is the final report and completes the requirements of the contract and contains all of the pertinent data that has been compiled and edited.

In an effort to make the results of the study applicable to the entire industry many companies and organizations were questioned concerning their opinions, suggestions, and cautions for the study. Excellent cooperation was obtained and a list of those contributing is given in Appendix G (Vol. 2).

ABSTRACT

An investigation to determine the optimum pressure of aircraft hydraulic systems has been made. A survey of the status of present and future designs of hydraulic systems was conducted to be used as a basis of the analysis. A detail analysis of weight, space, performance, and heat throughout the system was conducted taking into consideration the factors of cost and reliability. A range of system pressures from 1500 to 10,000 psi was found sufficient to define the optimum.

On the basis of constant recurring cost and final reliability, a theoretical optimum pressure was indicated to be 4000 psi. Considering the above variables this optimum pressure results in an overall saving equivalent to 2.46% total system weight. The effect of different variables upon this result was analyzed and the results plotted. The initial expense in temporary loss in reliability associated with a basic change in system pressure was weighted against the saving quoted above in making the final recommendations.

PUBLICATION REVIEW

The publication of this report does not constitute approval by the Air Force of the findings or the conclusions contained therein. It is published only for the exchange and stimulation of ideas.

FOR THE COMMANDER:

for E H Schwartz

D. D. MCKEE
Colonel, USAF
Chief, Aircraft Laboratory
Directorate of Laboratories

FINAL REPORT

VOLUME I

TABLE OF CONTENTS

FINAL REPORT

VOLUME I

TABLE OF CONTENTS

PAGE

FINAL REPORT

VOLUME I

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS

LIST OF TABLES

FINAL REPORT

VOLUME I

LIST OF TABLES

FINAL REPORT APPENDIX

VOLUME II

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS

FINAL REPORT APPENDIX

VOLUME II

LIST OF ILLUSTRATIONS

FINAL REPORT APPENDIX

VOLUME II

LIST OF ILLUSTRATIONS

LIST OF TABLES

INTRODUCTION

The Glenn L. Martin Company has conducted a "Theoretical Investigation of Optimum Pressures in Aircraft Hydraulic Systems." A complete analysis of hydraulic systems, through a range of pressures of 1500 psi to 10,000 psi, was made.

What is the optimum pressure? The optimum pressure is defined as the system pressure which results in the most ideal combination of design factors, considered in their relative importance, in an aircraft hydraulic system.

What is an aircraft hydraulic system? An aircraft hydraulic system includes all systems used primarily to transmit or control power hydraulically, designed to become airborne in piloted airplanes, pilotless aircraft, and helicopters.

What previous studies have been made? With the use of hydraulic power in aircraft nearly twenty years old, there have been many previous studies on the same subject. One of the earliest studies was that of Harold W. Adams of Douglas Aircraft Company made in 1943. His results are published in his book Aircraft Hydraulics. His analysis is based on 90% efficiency through ten feet of tubing, a safety factor of 6:1, and fittings spaced at an average of 40 inches. The minimum weight of the hydraulic lines occurs at 1700 psi for 52S0 lines and dural fittings, at 2900 psi for 17 ST lines and steel fittings, and 3100 psi for 18-8 lines and steel fittings. An approximate analysis for cylinders indicates a minimum weight above 5000 psi.

Flight magazine in the 17 July 1950 issue presented a history of aircraft hydraulics. At that time the British hydraulics designers had made a study of 2000 psi, 3000 psi, and 4000 psi systems. The results stated in the article (if corrected by reversing the percentages) are 12% weight saving from 2000 to 3000 psi and 6% weight savings from 3000 to 4000 psi.

More recently a weight comparison between 3000 psi and 5000 psi has been made for a typical subcircuit, by Lou Berthelsen of North American Aviation. The scope of the study was limited to one subcircuit and the results indicate little difference in the weights of 3000 psi and 5000 psi subcircuits. (The analysis appears in Appendix H, Volume II)

Why has this new study been conducted? All of the previous studies have limited themselves to studying the problem of optimum pressures with weight and availability as the only criteria. With the aircraft structure becoming more dense for high speed operation, the space occupied by the components of the hydraulic system has become of great importance. Many new materials have become available for use in hydraulic components, lower safety factors have come into use through the refinement of the art, and the importance of miscellaneous components such as valves is sufficiently great to justify inclusion in the analysis. The rise in system operating temperature to 275°F in the immediate future, 400°F in several years, and ultimately higher have made it necessary to develop many components. This suggests concurrent change to optimum pressure.

The intent of this study is to make a comprehensive analysis including the effect of various pressures on all components and on system arrangements in use today or predicted for the near future, taking into consideration weight, cost, reliability, performance, and heat rejection.

1. SUMMARY AND RECOMMENDATIONS

1.a. Purpose

The purpose of this report was to determine the optimum pressure or pressures in aircraft hydraulic systems and to evaluate the practical significance of these pressures.

The aircraft hydraulic systems were to include all systems used primarily to transmit or control power hydraulically,designed to become airborne in piloted airplanes, pilotless aircraft, and helicopters.

The optimum pressure is taken to mean a value of system pressure which would result in the most ideal combination of design factors, including weight, space and other considerations recommended by industry as being important.

1.b. Integrity of Results

The analysis involved more than pure mathematics. As an engineering problem it involved determination of input data from representative statistics where available, from reasonable estimates obtained from leading engineers and, in some cases, from opinion. The integrity of the results of any such analysis depends upon the attitude of the engineers being impartial and devoid of any preconceived ideas or preference as to how the study should be concluded.

The importance of these facts was recognized at the outset. In any well rounded engineering organization one can find agressive individuals with active imaginations who have learned to sell their ideas by painting rosy pictures. Conservative individuals can be found who, after an overdose of bitter design backfires, fear, resist or exaggerate any departure from the conventional. Personnel for the analysis were selected from a group in between these extremes and supervision was continually directed toward obtaining unbiased results.

1.c. Data

A survey of the aircraft industry was made to define the hydraulic systems to be used as a basis for the analysis. To obtain factual data from responsible engineers and obtain a direct insight into the most acute problems involving the other possible operating pressures, personal interviews were arranged with all of the principal aircraft producing companies and information was obtained directly. From this survey was determined the system horsepower (which is plotted against time in figure 1-1), the types of systems and the extent of use of each, the types of components, tubing and fittings, flexible hose connections in use, materials of construction and the safety factors in current use.

A history of system pressure versus time is presented in figure 1-2. This figure illustrates the degree of standardization that was accomplished throughout the industry as a result of the efforts of the services.

Figure 1-1, system horsepower versus time, shows the wide divergence of system horsepower with a tendency toward higher horsepower in the future. Based on this figure the range from zero up to 100 horsepower was explored in the study.

To obtain the development and production status of high pressure components and the adaptability of 3000 psi units to higher pressure, questionnairs were sent to the leading hydraulic equipment manufacturers. Their opinions, suggestions, and cautions have been incorporated in Section 4 on practical significance. The present development work is concerned mostly with pressures up to 5000 psi and many of the 3000 psi units are adaptable to pressures of 4000 psi and 5000 psi.

The available literature concerning previous pressure studies, materials for hydraulic equipment, new developments, and analysis techniques was surveyed and where applicable was used in the study. Historical information concerning aircraft hydraulic systems was also available in books, reports, periodicals (see bibliography) and supplemented the survey of aircraft and hydraulic equipment manufacturers.

1.d GENERAL METHOD OF ANALYSIS

In order to make best use of the analysis time available, the problem was examined to determine the factors which must be considered, the relative importance of each factor, and a method of including each factor in the study. This is completely covered in Section 3, Determination of Optimum Pressure. The factors considered were weight, space, cost, reliability, performance and heat. In correlating the different factors used to determine the optimum pressure it was necessary to establish the relative importance of each. The total system weight is the factor of primary importance, all other factors were evaluated independently and then correlated with total system space. Total system space was considered twenty percent as important as total system weight. Cylinder space was considered to be thirty percent as important as total system weight. The factors of cost and reliability were incorporated into the study; first, by considering recurring, production, and maintenance cost and ultimate reliability held constant throughout the numerical analysis, then by weighing the initial cost and temporary loss in reliability associated with making a change against the benefits otherwise indicated by the numerical analysis. Performance was reflected by the factor of cylinder stiffness which was handled by considering the percent deflection of cylinders one-tenth as important as total system weight. The heat factor was represented by a measure of system stabilized temperature increment which was considered one-tenth as significant as total system weight.

The aircraft companies surveyed were in general agreement with the above order of importance of these factors. Once the relative importance had been established, the detail analysis was undertaken, so that the degree of detail and the accuracy of the results for each factor were in proportion to the relative importance of the factor.

The range of pressures studied was 1500 psi to 10,000 psi. The results obtained indicate that the region of the optimum pressure is well defined by this range of variation under all conditions.

A detail analysis of the components comprising a hydraulic system of any aircraft has been made for both weight and space. The basic method of derivation of the theoretical formulas was as follows:

<u>Step 1</u>

A theoretical formula was derived for each component with the independent variables being horsepower (or work) and pressure. The component itself was broken down into basic elements, and each element was designed to meet various internal pressures at each horsepower or work level. The internal size of the component, of course, changes in accordance with the reduced flow or displacement at higher pressures. By integrating the elements, an equation was obtained which represented the weight or space of the entire component.

Step 2

Data on actual 3000 psi were then collected and the weight and space curves were plotted.

Step 3

The coefficients in the theoretical equations in Step 1 were then determined based on the actual data plotted in Step 2.

Step 4

The characteristic curves showing the effects of variables were then superimposed on the actual curves to assure the validity of the mathematical functions.

Examination of the makeup of the elements of the valves, in general, indicates that the proportions of bulk and pressure stressed materials are approximately the same as for the common check valve. Plots of actual weight and space for each valve were compared to that of the check valve. Since the characteristics are similar, but the actual values of a fixed ratio, at a given horsepower, the weight and space equations for check valves, which were developed in great detail, were multiplied by the statistical ratio obtained above to yield a weight and space equation for each type of valve.

In the analysis of the transmission lines, considerable statistical data was used to determine the utilization of each type of fitting (including support clamps) per unit length of tubing and the fitting was analysed in detail. The effect of pressure on the density of the fluid was included.

In the analysis of actuating cylinders statistical data was used to determine the proportions in current use and the effect of column strength limitations was included in the analysis.

These analyses are completely reported in the various appendixes in Volume II. At the end of each analysis is presented a family of summary curves illustrating the effect of weight and space upon the particular component involved.

In order to integrate the results of the component analysis, it was necessary to establish the relative proportions of these components in various systems. From the results of the survey, six different systems were designed and used as a basis for the summary analysis being considered representative of the range and sizes, proportions, and type of systems for present and future use. A discussion of the selection of each system and the applications it represents is given in Section 2-A. The components and lines were all specified, and rated by work or horsepower. By using the curves developed in Volume II, the weight and space of each element was determined. The summation of the results have been reported completely so that the contribution of each component of the system can be observed. The subtotal and total weight and space curves for each system have been plotted in Section 3. The diagrams of the systems are presented in Section 2.

1.0 RESULTS

The total system weight curves (figure 3-1) are amazingly similar in characteristic in spite of the broad variation in type, size, and proportion of systems considered. When these curves are normalized and averaged so that each system counted equally in the results, the summary curve for weight alone indicated a mathematical low point at 4400 psi with a practical point of no return at 4000 psi, and a saving of 4.4 percent in progressing from 3000 to 4000 psi.

The space curves summarizing total system space (figure 3-3) and the cylinder space (figures B-16A and B-16B, Volume II) are reasonably consistent throughout the range of systems considered. These curves as normalized and then averaged as shown in figure 3-8 indicate a saving of 13.8 percent system space and 10.28 percent cylinder space in progressing from 3000 to 4000 psi.

A saving of 19.5 percent system space and 14.22 percent cylinder space is indicated in progressing from 3000 to 5000 psi.

The most noticeable effect in performance of hydraulic actuators is the change in stiffness associated with the change in system pressure. The potential dynamic response of the servo mechanisms depends directly upon their stiffness. Section 3-B is devoted to this problem. The increase in unit loads of cylinders caused by column limitations is illustrated in figure 3-5 and the present deflection of cylinders at various pressures is illustrated in figure 3-6. The combined effect of these factors (marked actuator deflection in figure 3-8) indicates 44.8 percent increase in flexibility from 3000 to 4000 psi and 86.6 percent increase from 3000 to 5000 psi.

A heat analysis of a typical system was conducted to determine the effect of system pressure on the problem of heat rejection. This analysis included the variation in each source of heat within the system and the effect of variation in the exposed system area as plotted on figure 3-7. The combination of these effects (marked stabilized temperature in figure 3-8) show 33 percent increase in the heat problem in increasing from 3000 to 4000 psi and 72.8 percent in increasing from 3000 psi to 5000 psi.

In order to determine the combined effects of all variables, total system weight ratio was chosen as the common denominator. The other variables wefe multiplied by the appropriate weight factor of 20 percent for system space, 30 percent for cylinder space, 10 percent for cylinder deflection, and 10 percent for stabilized temperature increment and the summation of the results plotted in figure 3-9. This figure represents the final results of the theoretical analysis. It being a penalty curve, the optimum is indicated by the low point. Therefore, the theoretical optimum pressure is 4000 psi and considering all factors a total system weight saving of 2.46 percent is achieved in progressing from 3000 to 4000 psi. A detail breakdown of the effect of each variable is given in Section 3-E, the integration of variables.

To illustrate the sensitivity of the weight factor for total system weight was varied from zero to infinity and the results plotted in figure 3-10. Also the actuator deflection weight factor was varied from zero to twenty percent. The theoretical optimum pressure is fairly stable as the weight factors are varied to encompass a variety of design situations. The reduction in penalty or percent system weight, if the system pressure were adjusted to each situation, is so small (with respect to the penalty at 4000 psi) that consideration of more than one optimum pressure is not justified.

The investigation of substitution of new materials such as titanium for highly stressed parts revealed that weight of approximately 11 percent can be obtained throughout the entire pressure range, by application of titanium to such parts as accumulator barrels, tubing and fittings, and housings. However, the reduced system weight curve plotted on 3-1 indicates no shift in the low point, or optimum pressure.

Section 4, entitled Practical Significance of the Optimum Pressure, is written for the purpose of illustrating the feasibility of making the change to 4000 psi and the nature of the problems to be encountered prior to deciding on the advisability of making the change. This section indicates that if economical, and recognizing the temperature problem facing the industry today, it would be feasible to begin designing to 4000 psi as a new system pressure at the present time on the basis of the probability of components being available with relatively minor changes within one year.

1.f CONCLUSIONS

1. Not considering initial cost, the indicated optimum pressure is 4000 psi.

2. The total effective gain in proceeding from 3000 psi to 4000 psi would be an effective system weight saving of 2.46% (See figure 3-9). A breakdown of the saving is as follows:

   a. Total system weight reduction of 4.16%

   b. Total system space reduction of 13.8%

   c. Cylinder space reduction of 10.28%

   d. Cylinder deflection increase from 33% to 44.8%

   e. Stabilized system temperature (increment above compartment temperature) increase of 33%.

3. The theoretical optimum pressure is fairly independent of the type, size, and proportions of hydraulic systems.

4. A tremendous initial, non-recurring, cost would be required to make a change from 3000 psi to 4000 psi. This would be necessary for the establishment of new standards, the design and development of new components and test facilities, and the procurement and the initial training associated with the replacement of all the existing test facilities (See Section 5).

5. A temporary loss in reliability will occur in the transition period during the debugging of newly developed equipment.

6. In the opinion of the contractor, the small gain of 2.46% effective reduction in system weight (itemized above) does not justify the initial cost and temporary loss in reliability which would occur if the standard system pressure were changed to 4000 psi.

1.g RECOMMENDATIONS

It is recommended that the present standard of 3000 psi beheld as the upper limit of pressure for production of hydraulic systems.

This recommendation was made on the basis of:

1. The conclusions (Section 1.f) which indicate that the potential gain by changing system pressure does not justify the initial cost and temporary loss of reliability.

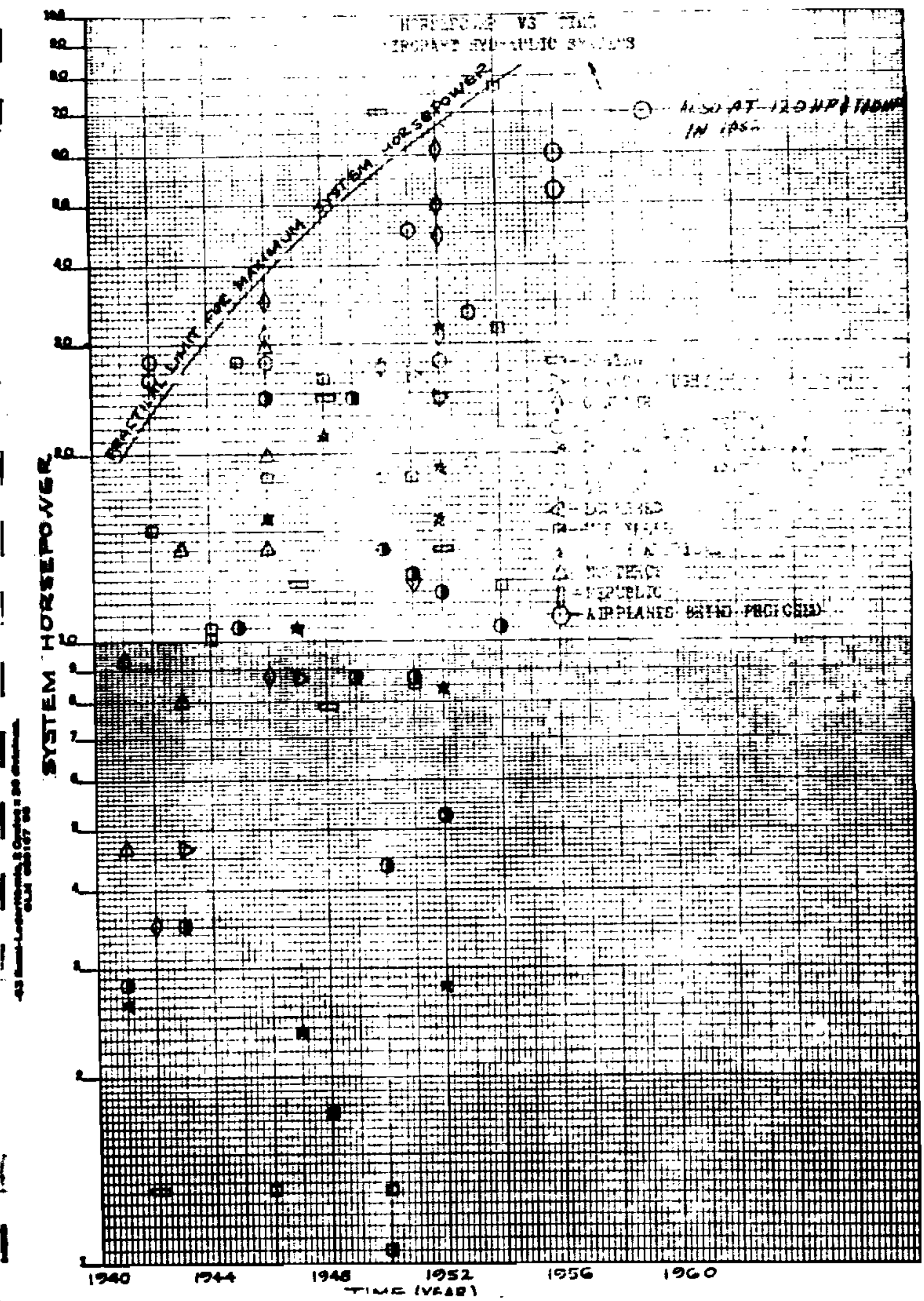
2. A change in system pressure in the near future would have to be accomplished concurrently with development of high temperature systems and would extend the period of development and delay the achievement of either high temperature or high pressure systems being a source of double trouble.

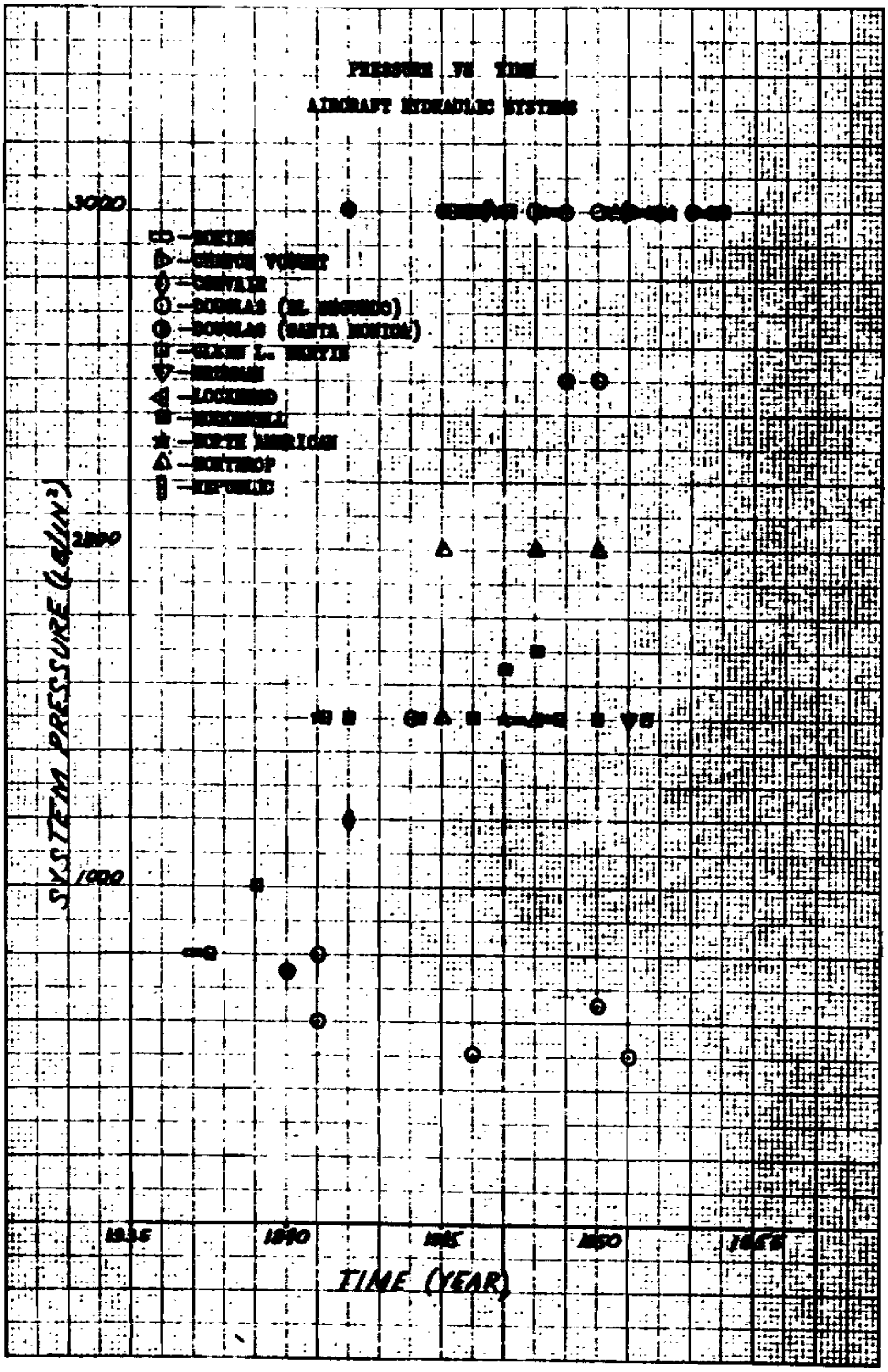
3. There are potential developments within the field of aircraft hydraulics which are necessary to meet future requirements or which indicate more potential improvement with less expenditure than in the case of changing system pressure. Such developments, listed in order of their importance, are as follows:

   a. Development of fluid, seals, flexible lines, components, and system which will be compatable with a temperature range extended upward progressively, keeping in pace with aircraft performance as limited by basic structural and power plant considerations.

   b. Development and maintenance of a reliability program with the ultimate goal of "Engineering for Reliability" by determining economical degree of reliability for each type aircraft, analyzing reliability of each system, specifying reliability requirements for each components with component development where needed, and maintaining control through statistical records.

   c. Use of superior materials in present components as illustrated by Section 2.e (a potential system weight reduction of 11%).


PRACTICAL LIMIT FOR MAXIMUM SYSTEM HORSEPOWER
SYSTEM HORSEPOWER
TIME (YEAR)
1940
1944
1948
1952
1956
1960


PRESSURE VS TIME
AIRCRAFT HYDRAULIC SYSTEMS
LOCKHEED
NORTH AMERICAN
NORTHROP
3000
2000
1000
SYSTEM PRESSURE (LB/IN²)
1935
1940
1950
TIME (YEAR)

2. SYSTEM ANALYSIS

2.A. SYSTEM DESIGN

Having analyzed the individual hydraulic components as reported in Volume II, it is necessary to establish the relative proportions of these components in the system so that these results might be integrated.

To accomplish this, a survey was conducted to determine the sizes, types, and proportions of systems in use or contemplated. The figure 1-1 is a plot of system horsepower versus time. Six different systems were established as representative of the results of the survey.

1.	Small Flight Control	Figure 2-1
2.	Medium Flight Control	2-2
3.	Large Flight Control	2-3
4.	Medium Main Hydraulic System	2-4
5.	Large Utility System	
6.	Power Package	2-5

The small flight control system is representative of the type finding wide application in pilotless aircraft, helicopters, and small subsonic piloted airplanes where the circuit is devoted primarily or entirely to flight control actuators.

The Medium Flight Control System is representative of the type applied to transonic piloted airplanes in the weight class of interceptors or fighter bombers.

The Large Flight Control System is proportioned to a large high speed airplane such as a strategic bomber assuming a typical central hydraulic system used. These three systems span the entire range of physical dimensions, horsepower requirements (predicted by figure 1-1), and degree of complication starting with the elementary small system to the more elaborate detail circuit design of the Large Flight Control System. The variation of plumbing proportions in a given system is treated later.

The Medium Main System represents the requirements, proportions, and degree of complication of the interceptor-fighter bomber night intruder (all transonic) classes of piloted airplanes. The circuit is designed primarily by the airplane utility requirements. Although a subcircuit is included which operates flight controls it is equivalent to an additional utility circuit in all respects except in the heat analysis which it becomes a major contributer.

The Large Utility System is a typical central hydraulic system devoted entirely to utility requirements of a large high speed piloted airplane such as a large strategic bomber.

The power package is a complete hydraulic system installed in a relatively small area having sufficient power to be applicable to requirements of a medium or large airplane. Since the actuators are motor type it represents the general class of systems which drive rotary equipment.

The Medium Main System was modified by reducing the plumbing lengths to zero to represent a second type of power package with linear actuators.

The plumbing length in the Medium Main System was doubled to represent the effect of long lengths with respect to power requirement such as in the case of a transport airplane.

The analysis revealed that optimum system pressure was not sensitive to system power, system type, or system proportions so that design of additional intermediate systems was unnecessary since the range of variables above encompasses the intended scope of the study

2.B SYSTEM ANALYSIS PROCEDURE

The systems which were considered to be representative of the current usage in the aircraft industry, were established as being a power pack system, a small flight control system, a medium flight control system, a large flight control system, a medium main system, and a large utility system. A complete sketch (see figures 2-1 through 2-5) was made for each type of system except the latter, which is similar to the medium main system, showing each component and its horsepower or work rating, and each length of line and its horsepower rating. To obtain authentic data, free use was made of existing designs of hydraulic systems in use in airplanes and missiles in existance today.

Each component was tabulated (see Table 2-1 through 2-6) and opposite each component, for its horsepower or work rating, the weight and space volume was listed. The valves, cylinders, lines, etc., were grouped in the tabulation so that sub totals of each group could be shown. This grouping permits the reader to observe, by inspection, exactly what the distribution of weight or space is. In order to facilitate accurate interpolation of the curves of weight or space versus pressures for constant horsepower levels other than 2, 10, 50, and 100 horsepower, and for cylinder work levels other than 3000, 20,000, 100,000 and 600,000 inch-pounds, cross plots were made. That is, curves were plotted for weight or space versus horsepower or work at 1500, 3000, 4000, 5000, 7500, and 10,000 pounds per square inch constant pressure. It should be noted that the same type of plot can be made for any pressure level desired, and this is of direct benefit to anyone in search of additional utilization of this report. Finally the columns for each pressure were sub totaled and totaled.

In the case where a number of systems are used in the same airplane for reliability or other reasons, it is the individual system itself which is the basis of the analysis and not the total amount of hydraulic equipment within the airplane.

TABLE 2-1A
14

SYSTEM SIZE SMALL FLIGHT CONTROL (WEIGHT)

NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Pump, Variable Vol.	1	7.61		8.9	8.9	8.9	8.9	8.9	8.9
Reservoir, 100 IN-1500 PSI	1			8.39	7.60	7.39	7.10	6.84	6.71
Eng. Disconnect	2	3.66		.56	.50	.55	.58	.68	1.10
Grnd Test Disconnect	1	11.8		.39	.36	.37	.39	.51	.70
Grnd Test Disconnect	1	5.25		.29	.25	.27	.29	.39	.55
In Line Disconnect	2	1.83		.46	.36	.46	.52	.68	.86
Filter	1	3.66		1.04	1.04	1.04	1.04	1.04	1.04
Filter	2	1.83		1.48	1.48	1.48	1.48	1.48	1.48
Relief Valve	1	3.66		1.17	1.04	1.17	1.24	1.43	2.34
Solenoid Control Valve	2	1.83		2.02	1.57	2.02	2.26	2.96	3.74
Check Valve	2	3.66		.36	.32	.36	.38	.44	.72
Sub Total				7.77	6.92	7.72	8.18	9.61	12.53
Cylinder	1	9430*		1.07	.84	.75	.73	.83	.93
Cylinder	1	8870*		1.03	.81	.72	.70	.79	.89
Sub Total				2.10	1.65	1.47	1.43	1.62	1.82
Pressure Line	1	3.66	28	.28	.14	.14	.14	.20	.34
Pressure Line	1	2.98	14	.13	.08	.08	.08	.10	.15
Pressure Line	1	1.83	310	2.11	1.61	1.61	1.61	2.11	3.20
Sub Total				2.52	1.83	1.83	1.83	2.41	3.69
Supply Line	1	3.66	10	.15	.15	.15	.16	.20	.24
Sub Total				.15	.15	.15	.16	.20	.24
Return Line	1	3.66	26	.25	.17	.16	.15	.15	.17
Return Line	1	1.83	214	1.50	1.11	1.11	1.16	1.26	1.39
Sub Total				1.75	1.28	1.27	1.31	1.41	1.56

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-1A

SYSTEM SIZE SMALL FLIGHT CONTROL (WEIGHT)

NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Hose	1	5.25	20	.60	.35	.35	.35	.45	.90
Hose	1	1.1	16	.28	.21	.21	.21	.28	.34
Hose	1	11.8	24	.96	.66	.66	.66	.72	1.08
Hose	1	3.66	44	1.10	.55	.55	.55	.79	1.34
Hose	1	1.83	48	.82	.62	.62	.62	.82	1.24
Sub Total				3.76	2.39	2.39	2.39	3.06	4.90
Total				35.34	30.72	31.02	31.30	34.05	40.35

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-1B
16

SYSTEM SIZE SMALL FLIGHT CONTROL (SPACE)

NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Pump, Variable Vol.	1	7.61		198.00	198.00	198.00	198.00	198.00	198.00
Reservoir, 100 N3-1500 PSI	1			403.20	365.90	348.30	338.90	325.90	319.37
Eng. Disconnect	2	3.66		5.55	3.15	2.65	2.60	2.60	2.60
Grnd. Test Disconnect	1	11.8		5.11	2.71	2.14	2.02	1.79	1.79
Grnd Test Disconnect	1	5.25		3.15	1.79	1.45	1.38	1.36	1.36
In Line Disconnect	2	1.83		3.60	2.60	2.40	2.35	2.35	2.35
Filter	1	3.66		1.70	1.20	1.00	.95	.90	.75
Filter	2	1.83		2.00	1.60	1.56	1.50	1.44	1.40
Relief Valve	1	3.66		28.85	16.38	13.78	13.53	13.53	13.53
Solenoid Control Valve	2	1.83		92.20	66.60	61.40	60.20	60.20	60.20
Check Valve	2	3.66		2.22	1.26	1.06	1.04	1.04	1.04
Sub Total				144.38	97.29	87.44	85.57	85.21	85.02
Cylinder	1	9430*		2.75	2.40	2.05	1.96	2.00	2.31
Cylinder	1	8870*		2.66	2.30	2.00	1.88	1.90	2.23
Sub Total				5.41	4.70	4.05	3.84	3.90	4.54
Pressure Line	1	3.66	28	3.08	1.40	1.12	1.10	1.06	1.26
Pressure Line	1	2.98	14	1.40	.59	.56	.54	.53	.56
Pressure Line	1	1.83	310	4.48	1.99	1.68	1.64	1.59	1.82
Sub Total				8.96	3.98	3.36	3.28	3.18	3.64
Supply Line	1	3.66	10	3.75	3.20	3.05	2.90	2.90	2.90
Sub Total				3.75	3.20	3.05	2.90	2.90	2.90

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-1B
17

SYSTEM SIZE SMALL FLIGHT CONTROL (SPACE)

NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Return Line	1	3.66	26	2.86	1.30	1.04	.91	.78	.78
Return Line	1	1.83	214	14.98	6.42	6.42	6.42	6.42	6.42
Sub Total				17.84	7.72	7.46	7.33	7.20	7.20
Hose	1	5.25	20	23.80	10.20	8.50	8.50	7.65	7.65
Hose	1	1.1	16	8.16	4.49	4.49	4.49	4.49	4.49
Hose	1	11.8	24	51.00	24.50	18.37	16.32	13.77	16.82
Hose	1	3.66	44	41.10	18.70	15.72	14.97	14.22	16.83
Hose	1	1.83	48	24.50	13.47	13.47	13.47	13.47	13.47
Sub Total				148.56	71.36	60.55	57.75	53.60	59.26
Total				930.10	746.95	712.21	697.57	679.89	679.93

* WORK IN INCH LBS. BY AN ASTERISK

TABLE 2-2A

18

SYSTEM SIZE MEDIUM FLIGHT CONTROL (WEIGHT)

NO.	NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
1	Pump, Variable Volume	1	13.1		13.13	13.13	13.13	13.13	13.13	13.13
2	Reservoir, 231 IN^3 at 3000 psi	1			27.9	18.35	14.43	12.54	10.46	8.37
3	Accumulator, 100 IN^3	1			10.1	9.2	9.15	9.1	10.1	10.4
4	Eng. Disconnect	2	13.1		.954	.861	.891	.963	1.231	1.668
5	Eng. Disconnect	1	.003		.2216	.1892	.2305	.2645	.348	.433
6	Grnd Charge Disconnect	2	18		1.108	1.015	1.05	1.123	1.428	1.917
7	Purge Valve	1	13.1		.31	.28	.29	.313	.40	.542
8	Filter	2	4.3		2.24	2.24	2.24	2.24	2.24	2.24
9	Filter	1	13.2		1.92	1.92	1.92	1.92	1.92	1.92
10	Filter	1	0.5		.39	.39	.39	.39	.39	.39
11	Relief Valve	1	14.9		2.145	1.95	2.014	2.163	2.6	3.522
12	Control Valve	3	4.3		3.25	2.875	3.16	3.47	4.7	6.41
13	Control Valve	1	13.25		1.767	1.595	1.653	1.783	2.28	3.09
14	Shut Off Valve	1	8.6		1.74	1.504	1.607	1.71	2.29	3.16
15	Shut Off Valve	1	13.25		2.075	1.875	1.941	2.095	3.68	3.63
16	Shut Off Valve	1	4.3		1.272	1.125	1.24	1.36	1.843	2.51
17	Check Valve	2	13.25		.62	.56	.58	.626	.80	1.084
18	Check Valve	2	18.0		.72	.66	.682	.730	.928	.846
19	Check Valve	3	4.3		.57	.504	.555	.609	.825	1.125
20	Control Valve	1	.428							

* WORK IN INCH LBS., SHOWN BY AN ASTERISK

TABLE 2-2A
19

SYSTEM SIZE MEDIUM FLIGHT CONTROL (WEIGHT)

NO.	NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
1	Valve Sub Total				21.3006	19.5432	20.4435	21.7595	27.903	34.487
2	Cylinder	2	20,400*		5.68	3.7	3.54	1.78	4.2	4.84
3	Cylinder	1	175,000*		14.4	8.8	8.8	9.4	11.9	14.2
4	Cylinder	1	2820*		.495	.38	.352	.343	.37	.41
5	Sub Total				20.575	12.88	12.692	13.303	16.47	19.45
6	Pressure Line	1	13.1	71	1.208	.824	.796	.824	.924	1.31
7	Pressure Line	1	18	25	.49	.35	.3422	.3555	.395	.535
8	Pressure Line	1	14.9	10	.18	.124	.121	.125	.142	.196
9	Pressure Line	1	8.6	145	2.118	1.305	1.275	1.305	1.538	2.261
10	Pressure Line	1	4.3	290	3.245	1.913	1.885	1.913	2.35	3.598
11	Pressure Line	1	13.7	80	1.375	.936	.92	.951	1.064	1.502
12	Pressure Line	1	13.2	10	.17	.114	.111	.116	.132	.184
13	Pressure Line	1	.5	110	.759	.572	.572	.572	.759	.901
14	Sub Total				9.545	6.138	6.0222	6.1615	7.294	10.487
15	Supply Line	1	18	20	.504	.44	.428	.434	.47	.555
16	Supply Line	1	13.1	30	.675	.591	.564	.585	.64	.768
17	Sub Total				1.179	1.031	.992	1.019	1.11	1.323
18	Return Lines	1	.5	100	.465	.4956	.5159	.5375	.5898	.65
19	Return Lines		13.7	106	1.855	1.144	.996	.900	.859	.944

* WORK IN INCH LBS., SHOWN BY AN ASTERISK

TABLE 2-2A
20

SYSTEM SIZE MEDIUM FLIGHT CONTROL (WEIGHT)

NO.	NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
1	Return Line	1	8.6	120	1.73	1.055	.972	.851	.804	.894
2	Return Line	1	4.3	290	3.046	1.972	1.972	1.77	1.71	1.885
3	Return Line	1	14.9	10	.182	.113	.0962	.088	.085	.092
4	Return Line	1	18	30	.591	.36	.312	.288	.2965	.30
5	Sub Total				7.8681	5.1396	4.8541	4.4345	4.3443	4.765
6	Hose	1	13.1	60	2.55	1.74	1.68	1.74	1.98	2.76
7			.003	30	.48	.39	.39	.39	.48	.68
8			18	38	1.86	1.34	1.31	1.36	1.50	2.035
9			4.3	72	2.015	1.21	1.15	1.21	1.44	2.23
10			13.2	36	1.53	1.046	1.009	1.045	1.17	1.657
11			.5	36	.612	.469	.469	.469	.612	.756
12	Sub Total				9.047	6.194	6.008	6.214	7.182	10.068
13	Total				120.6447	91.6058	87.7018	87.6615	97.9933	112.48
14										

* WORK IN INCH LBS, SHOWN BY AN ASTERISK

TABLE 2-2B

21

SYSTEM SIZE MEDIUM FLIGHT CONTROL (SPACE)

NO.	NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
1	Pump, Variable Volume	1	13.1		288	288	288	288	288	288
2	Reservoir, 231(IN3) at 3000 psi	1			1340	810	680	600	500	420
3	Accumulator 100 (IN3)	1			375	190	140	112	84	57
4	Eng Disconnect	2	13.1		12	6.5	5.24	4.5	4.14	4.1
5	Eng Disconnect	1	.003		1.175	1.175	1.175	1.175	1.175	1.175
6	Grnd Charge Disconnect	2	18		13.8	7.36	5.76	5.06	4.510	4.52
7	Purge Valve	1	13.1		2.4	1.3	1.05	.9	.83	.82
8	Filter	2	4.3		78	50	37	32	25	21
9	Filter	1	13.2		106	64	44	35	24.2	19
10	Filter	1	0.5		9.0	7.5	7.0	6.5	6.0	6.0
11	Relief Valve	1	14.9		67.6	36.4	28.6	26	23.25	23.15
12	Control Valve	2	4.3		153.3	89.6	76.8	70.4	70.4	70.4
13	Control Valve	1	13.25		153.5	83.2	67.2	57.6	53.1	52.5
14	Control Valve	1	.428		30.1	30.1	30.1	30.1	30.1	30.1
15	Shut Off Valve	1	8.6		118.4	62.7	51.2	44.8	43.5	43.5
16	Shut Off Valve	1	13.25		153.5	83.2	67.2	57.6	53.1	52.5
17	Shut Off Valve	1	4.3		76.8	44.8	38.4	36.2	36.2	36.2
18	Check Valve	2	13.25		4.8	2.6	2.1	1.8	1.66	1.64
19	Check Valve	2	18.0		6.0	3.2	2.5	2.2	1.96	1.966

* WORK IN INCH LBS, SHOWN BY AN ASTERISK

TABLE 2-2B
22

SYSTEM SIZE MEDIUM FLIGHT CONTROL SYSTEM (SPACE)

NO.	NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
1	Check Valve	3	4.3		3.6	2.1	1.8	1.65	1.65	1.65
2	Valve Sub Total				990.175	579.735	467.125	413.485	380.775	370.221
3	Cylinder	2	28,400*		76	58	49	47	48	57
4	Cylinder	1	175,000*		422	258	235	230	235	268
5	Cylinder	1	2,820		11.1	9.2	8.9	8.65	8.4	9.3
6	Sub Total				509.1	325.2	292.9	285.65	328.4	334.3
7	Pressure Lines	1	13.1	81	21.85	10.53	7.84	6.8	5.67	7.29
8	Pressure Lines	1	18	25	8.75	4.25	3.125	2.625	2.275	2.625
9	Pressure Lines	1	14.9	10	3.0	1.4	1.05	.93	.82	.95
10	Pressure Lines	1	8.6	145	29.0	13.78	10.15	8.7	7.97	10.15
11	Pressure Lines	1	4.3	290	34.8	15.95	13.05	13.05	11.9	14.22
12	Pressure Lines	1	13.7	80	22.4	10.8	8.0	6.8	6.0	7.2
13	Pressure Lines	1								
14	Pressure Lines	1	.5	110	7.15	3.3	3.3	3.3	3.63	3.63
15	Sub Total				126.95	60.01	46.515	42.205	38.465	46.065
16	Supply Line	1	18	20	17.2	11.2	9.7	8.6	7.2	6.6
17	Supply Line	1	13.1	30	21.9	14.5	12.75	11.1	9.6	8.85
18	Sub Total				39.1	25.7	22.45	19.7	16.8	15.45
19										

* WORK IN INCH LBS., SHOWN BY AN ASTERISK

TABLE 2-2B
23

SYSTEM SIZE MEDIUM FLIGHT CONTROL SYSTEM (SPACE)

NO.	NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
1	Return Lines	1	.5	100	3	3	3	3	3	3
2	Return Lines	1	13.7	106	29.7	12.73	9.23	7.42	5.3	4.77
3	Return Lines	1	8.6	120	24	10.1	7.56	6.0	5.04	4.8
4	Return Lines	1	4.3	290	36.2	17.4	14.5	11.6	10.15	8.7
5	Return Lines	1	14.9	10	3.0	1.3	.95	.75	.65	.5
6	Return Lines	1	18	30	10.2	4.5	3.3	2.55	1.8	1.65
7	Sub Total				106.1	49.03	38.54	31.32	25.94	23.42
8	Hose	1	13.1	96	220.0	106.0	80.0	68.5	59.1	73.4
9	Hose	1	.003	30	7.15	7.15	7.15	7.15	7.15	7.15
10	Hose	1	18	38	113.0	54.9	44.0	33.9	29.4	33.9
11	Hose	1	4.3	72	73.4	33.75	27.55	27.55	25.1	25.1
12	Hose									
13	Hose	1	.5	36	19.85	9.2	9.2	9.2	10.1	10.1
14	Sub Total				433.4	211.00	167.9	146.3	130.85	139.65
15	Total				4207.725	2534.675	2043.43	1938.66	1793.23	1694.106
16										
17										
18										
19										

* WORK IN INCH LBS; SHOWN BY AN ASTERISK

TABLE 2-3A
24

SYSTEM SIZE LARGE FLIGHT CONTROL SYSTEM (WEIGHT)

NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Pump	2	14.2		13.9	13.9	13.9	13.9	13.9	13.9
Reservoir (Tank Type 1045IN^3)	1			72.5	39.6	31.35	26.43	19.85	16.56
Accumulator 150 IN^3	2	450000*		14.2	12.8	12.7	12.8	14.2	14.6
Accumulator 100 IN^3	2	300000*		10.1	9.2	9.15	9.1	10.1	10.4
Check Valves	4	14.2		1.28	1.16	1.208	1.28	1.64	2.24
Check Valves	2	48.5		1.20	1.16	1.214	1.32	1.65	2.2
Check Valves	4	33.5		1.94	1.86	1.95	2.09	2.6	3.62
Check Valves	2	13.6		.66	.57	.59	.634	.81	1.10
Check Valves	2	1.5		.26	.23	.29	.236	.44	.50
Eng. Disconnects	4	14.2		1.965	1.78	1.855	1.965	2.52	3.44
Eng. Disconnects	2	.005		.308	.308	.308	.308	.308	.308
Grnd Test Disconnects	2	28.4		1.19	1.13	1.173	1.268	1.58	2.11
Purge Valves	2	14.2		.64	.58	.604	.64	.82	1.12
Relief Valves	1	28.4		2.89	2.745	2.845	3.07	3.84	5.13
Shut Off Valves	2	33.5		6.5	6.23	6.47	7.0	8.7	11.65
Shut Off Valves	1	13.6		2.21	1.91	1.98	2.125	2.71	3.65
Shut Off Valves	1	1.5		.87	.704	.971	1.127	1.475	1.675
Filters	2	33.5		3.07	3.07	3.07	3.07	3.07	3.07
Filter	1	13.6		1.97	1.97	1.97	1.97	1.97	1.97
Filter	1	1.5		.67	.67	.67	.67	.67	.67
Control Valve,Man.Op.	2	33.5		5.525	5.3	5.5	5.95	7.41	9.91
Control Valve,Man Op.	1	13.6		1.88	1.623	1.68	1.81	2.31	3.135
Control Valve,Man Op.	1	1.5		.741	.599	.827	.958	1.253	1.425
Valves Sub Total				35.769	33.669	35.155	37.491	45.776	58.923
Cylinder	2	222000		17.8	11.2	11.2	12.1	15.0	18.0
Cylinder	1	179000		14.5	8.9	9.0	9.8	12.2	14.5
Cylinder	1	9700		1.13	.86	.87	.85	.85	.96
Cylinder Sub Total				33.43	20.96	21.07	22.75	28.05	33.46

*WORK IN INCH LBS. SHOWN BY AN ASTERISK

Table 2-3A
25

SYSTEM SIZE LARGE FLIGHT CONTROL SYSTEM (WEIGHT)

NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Pressure Lines	1	14.2	230	4.05	2.76	2.715	2.76	3.175	4.37
Pressure Lines	1	28.4	26	.65	.504	.491	.509	.561	.71
Pressure Lines	1	33.5	659	18.33	14.6	14.2	14.73	16.2	20.05
Pressure Lines	1	1.5	100	.52	.52	.52	.52	.52	.52
Pressure Lines	1	13.6	620	10.67	7.31	7.07	7.31	8.18	11.54
Pressure Lines	1	15.1	732	13.2	9.07	8.93	9.22	10.4	14.35
Pressure Lines Sub Total				47.42	34.764	33.926	35.049	39.036	51.54
Supply Lines	1	28.4	378	11.47	9.93	9.75	9.83	10.6	12.21
Supply Lines	1	14.2	140	3.22	2.87	2.73	2.77	3.055	3.7
Supply Lines Sub Total				14.69	12.80	12.48	12.60	13.655	15.91
Return Lines	1	1.5	100	.65	.49	.5	.53	1.59	1.67
Return Lines	1	15.1	528	9.8	6.03	5.21	4.74	4.58	4.95
Return Lines	1	33.5	1120	28.8	17.92	15.45	14.9	14.55	15.57
Return Lines	1	48.5	10	.307	.194	.17	.16	.163	.174
Return Lines Sub Total				39/557	24.634	21.33	20.33	20.883	22.364
Hose	1	14.2	72	3.163	2.16	2.12	2.16	2.48	3.42
Hose	1	33.5	96	6.57	5.23	5.09	5.28	5.8	7.2
Hose	1	13.6	48	2.065	1.417	1.37	1.417	1.584	2.23
Hose	1	1.5	36	.585	.441	.45	.477	.531	.603
Hose Sub Total				12.383	9.248	9.000	9.334	10.395	13.453
Total				293.949	211.575	200.06	199.784	215.845	251.11

*WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-3B
26

SYSTEM SIZE LARGE FLIGHT CONTROL HYDRAULIC SYSTEM (SPACE)

NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Pumps	2	14.2							
Reservoir(Tank-1045IN3	1			604	604	604	604	604	604
Accumulator(150IN3)	2	450000*		2650	1600	1300	1090	820	660
Accumulator(100IN3)	2	300000*		1120	560	420	334	250	168
				750	370	284	224	169	113
Check Valves	4	14.2		10	5.2	4.2	3.8	3.44	3.4
Check Valves	2	48.5		13.5	6.6	5.0	4.5	3.86	3.78
Check Valves	4	33.5		19.4	10.0	7.6	6.8	5.88	5.8
Check Valves	2	13.6		4.9	2.6	2.06	1.86	1.68	1.66
Check Valves	2	1.5		1.4	1.0	.90	.90	.90	.90
Eng. Disconnects	4	14.2		25	13	10.5	9.5	8.6	8.5
Eng. Disconnects	2	.005		2.25	2.25	2.25	2.25	2.25	2.25
Grnd Test Disconnects	2	28.4		20.7	10.12	7.81	6.9	6.06	5.975
Purge Valves	2	14.2		5	2.6	2.1	1.9	1.72	1.7
Relief Valve	1	28.4		117.0	57.1	44.2	39.0	34.3	33.8
Shut Off Valves	2	33.5		437	225	171	153	132.3	130.4
Shut Off Valves	1	13.6		110	58.5	46.3	42.3	37.8	37.3
Shut Off Valves	1	1.5		31.5	22.5	20.25	20.25	20.25	20.25
Filters	2	33.5		524	310	206	162	108	80
Filters	1	13.6		110	66	44	36	25.5	19.8
Filter	1	1.5		17	12.5	10.5	10.0	8.0	7.5
Control Valve Man. Op.	2	33.5		620	320	243	217.5	188	185.5
Control Valve Man. Op.	1	13.6		157	83.3	66.0	59.5	53.8	53.1
Control Valve Man. Op.	1	1.5		44.9	32.0	28.8	28.8	28.8	28.8
Valves Sub Total				2270.35	1240.27	922.47	806.76	671.14	630.415
Cylinder	2	222000		1060	640	592	564	584	664
Cylinder	1	179000		435	265	240	232	240	272
Cylinder	1	9700		28.7	24.7	21.8	20.0	21.0	23.8
Cylinder Sub Total				1523.7	929.7	853.2	816.0	845.0	959.8

*WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-3B
27

LARGE FLIGHT CONTROL HYDRAULIC SYSTEM (SPACE)

NAME AND DESCRIPTION	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Pressure Lines	14.2	290	66.7	31.05	23	20.7	17.7	21.25
Pressure Lines	28.4	26	13.13	6.37	4.68	4.03	2.925	3.59
Pressure Lines	33.5	659	382	184.5	138.4	118.6	98.8	102.1
Pressure Lines	1.5	100	5.0	3.3	3.3	3.3	3.3	3.3
Pressure Lines	13.6	620	173.8	80.6	62.0	52.7	45.9	5.7
Pressure Lines	15.1	732	221.0	104.8	80.6	69.6	58.6	71.0
Pressure Lines Sub Total			861.63	410.62	311.98	268.93	227.225	206.94
Supply Lines	28.4	378	737	272	234.5	222.0	170.0	151.3
Supply Lines	14.2	140	106.3	70.0	60.9	53.9	47.6	43.4
Supply Lines Sub Total			943.3	342.0	295.4	275.9	217.6	194.7

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-3B
28

SYSTEM SIZE LARGE FLIGHT CONTROL SYSTEM (SPACE)

NAME AND DESCRIPTION	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Return Lines	1.5	100	6.0	3.0	3.0	3.0	3.0	3.0
Return Lines	15.1	528	158.5	75.5	50.2	39.7	29.1	26.45
Return Lines	33.5	1120	627	269	190.5	153.5	112.0	95.1
Return Lines	48.5	10	7.7	3.25	2.35	1.85	1.3	1.1
Return Lines Sub Total			779.2	350.75	246.2	198.05	145.4	125.65
Hose	14.2	72	52.1	24.25	18.0	16.2	13.85	16.7
Hose	33.5	96	139.3	67.2	50.4	43.2	36.0	37.2
Hose	13.6	48	33.6	15.6	12.0	10.2	8.89	11.04
Hose	1.5	36	4.5	2.97	2.97	2.97	2.97	2.96
Hose Sub Total			229.5	110.02	83.37	72.57	61.71	67.90
Total			1751.88	6517.36	5320.62	4690.21	4011.075	3730.405

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-4A
29

SYSTEM SIZE MEDIUM MAIN HYDRAULIC SYSTEM (WEIGHT)

NO.	NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
1	Pump	1	13.1		13.1	13.1	13.1	13.1	13.1	13.1
2	Pump	1	3.7		6.2	6.2	6.2	6.2	6.2	6.2
3	Reservoir (430 IN^3)				60.1	33.4	26.7	22.7	17.38	14.714
4	Accumulator (100 IN^3)	1	300000*		10.1	9.2	9.15	9.1	10.1	10.4
5	Eng. Disconnects	2	13.1		.954	.861	.892	.954	1.231	1.66
6	Eng. Disconnects	2	3.7		.554	.477	.529	.585	.80	1.06
7	Eng. Disconnects	2	.003		.3695	.3695	.3695	.3695	.3695	.3695
8	Grnd Test Disconnects	2	13.1		.83	.75	.777	.83	1.07	1.445
9	Purge Valve	1	13.1		.31	.28	.29	.31	.40	.54
10	Purge Valve	1	3.7		.18	.155	.172	.19	.26	.345
11	Filter	1	13.5		1.95	1.95	1.95	1.95	1.95	1.95
12	Filter	2	4.3		2.26	2.26	2.26	2.26	2.26	2.26
13	Filter	1	13.2		1.95	1.95	1.95	1.95	1.95	1.95
14	Relief Valve	1	14.9		2.145	1.95	2.01	2.145	2.76	3.74
15	Relief Valve	1	1.5		.846	.78	.955	1.085	1.43	1.69
16	Control Valve (Solenoid)	1	13.5		2.21	1.91	1.975	2.12	2.71	3.685
17	Control Valve (Solenoid)	1	4.1		1.273	1.325	1.25	1.36	1.84	2.51
18	Control Valve (Solenoid)	1	1.35		.87	.804	.985	1.119	1.474	1.74
19	Control Valve (Manual)	2	4.3		2.165	1.918	2.11	2.315	3.135	4.27
20	Control Valve (Manual)	1	13.2		1.768	1.595	1.654	1.768	2.28	3.08
21	Control Valve (Manual)	1	11.5		1.68	1.51	1.568	1.67	2.11	2.94
22	Brake Control Valve	2	1.3		1.485	1.37	1.47	1.905	2.51	2.965
23	Shut-Off Valve	1	8.6		1.742	1.541	1.61	1.71	2.31	3.15
24	Shut-Off Valve	1	13.2		2.075	1.875	1.943	2.075	2.68	3.62
25	Shut-Off Valve	1	13.5		2.21	1.91	1.975	2.12	2.71	3.685
26	Sequence Valve	1	1.2		.74	.684	.837	.951	1.254	1.48
27	Sequence Valve	3	.5		2.05	2.05	2.05	2.05	2.05	2.05
28	Sequence Valve	2	4.4		2.165	1.925	2.11	2.31	3.135	4.27
29	Check Valve	1.	13.1		.31	.28	.29	.31	.40	.54
30	Check Valve	1	3.7		.18	.155	.172	.19	.26	.345
31	Check Valve	2	.003		.24	.24	.24	.24	.24	.24
32	Check Valve	1	31.0		.465	.445	.462	.50	.625	.85
33	Check Valve	2	4.3		.38	.336	.37	.406	.55	.75
34	Check Valve	1	13.2		.31	.28	.29	.31	.40	.54
35	Check Valve	2	11.5		.59	.53	.55	.586	.74	1.05
36	Check Valve	2	2.6		.32	.28	.32	.36	.48	.62
37	Check Valve	2	13.5		.66	.57	.59	.634	.81	1.10

TABLE 2-4A
30

SYSTEM SIZE MEDIUM MAIN HYDRAULIC SYSTEM (WEIGHT)

NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Check Valve	2	4.1		.38	.336	.37	.406	.55	.75
Check Valve	2	1.35		.26	.24	.294	.334	.44	.52
Priority Valve	1	13.5		1.768	1.595	1.654	1.768	2.28	3.08
Fuse	1	11.5		.295	.265	.275	.293	.37	.515
Fuse	1	1.6		.13	.12	.147	.167	.22	.26
Fuse	1	Min		.13	.2	.147	.167	.22	.26
Fuse	2	13.5		.66	.57	.59	.634	.81	1.10
Fuse	2	4.1		.38	.336	.37	.406	.55	.75
Fuse	2	1.35		.26	.24	.294	.334	.44	.52
Restrictor (1 way)	1	13.5		.33	.285	.295	.317	.405	.55
Restrictor (2 way)	2	2.05		.29	.25	.30	.34	.45	.56
Restrictor (2 way)	1	1.35		.13	.12	.147	.167	.22	.26
Pressure Gage	1								
Valves Sub Total				43.378	39.712	41.995	45.137	56.358	71.834
Cylinder	2	28400*		5.7	3.7	3.6	3.56	4.2	4.86
Cylinder	1	175000*		14.3	8.8	8.8	9.6	12.0	14.3
Cylinder	3	6600*		2.52	1.98	1.77	1.695	1.92	2.13
Cylinder	1	15500*		1.66	1.18	1.08	1.07	1.25	1.42
Cylinder	2	57000*		10.6	6.8	6.4	6.7	8.0	8.86
Cylinder	1	322000*		24.9	15.8	16	17.2	21.7	26.2
Cylinder	2	54500*		10.2	6.1	6.2	6.4	7.76	8.5
Cylinder	12	5950*		9.36	7.44	6.54	6.36	7.08	7.91
Cylinder	2	58800*		10.9	7.0	6.6	6.8	8.2	9.6
Cylinder	2	27900*		5.6	3.64	3.5	3.46	4.1	4.76
Cylinder Sub Total				95.74	62.44	60.49	62.845	76.21	88.54

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-4A
31

SYSTEM SIZE MEDIUM MAIN HYDRAULIC SYSTEM (WEIGHT)

NAME AND DESCRIPTION	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Pressure Lines	13.1	515	8.25	5.87	5.67	5.98	6.7	9.37
Pressure Lines	14.9	25	.45	.31	.305	.315	.355	.4875
Pressure Lines	31.0	15	.393	.309	.30	.313	.345	.426
Pressure Lines	8.6	110	1.605	.99	.99	.99	1.145	1.695
Pressure Lines	4.3	420	4.7	2.77	2.77	2.77	3.36	5.2
Pressure Lines	.68	755	3.925	3.925	3.925	3.925	5.14	6.35
Pressure Lines	2.05	495	2.575	2.575	2.575	2.575	3.37	4.16
Pressure Lines	.23	300	1.56	1.56	1.56	1.56	2.04	2.52
Pressure Lines	11.5	35	.567	.371	.364	.378	.427	.602
Pressure Lines	9.8	25	.380	.235	.235	.235	.280	.410
Pressure Lines	2.6	140	1.12	.756	.756	.756	.98	1.43
Pressure Lines	1.3	955	4.97	4.97	4.97	4.97	6.5	8.025
Pressure Lines Sub Total			30.495	24.641	24.42	24.767	30.642	40.676
Supply Lines	13.1	50	1.11	.98	.935	.97	1.065	1.28
Supply Lines	3.5	10	.145	.145	.15	.162	.197	.242
Supply Lines Sub Total			1.255	1.125	1.085	1.132	1.262	1.522
Return Lines	13.1	430	7.4	4.56	3.98	3.57	3.4	3.74
Return Lines	31.0	30	.747	.462	.399	.381	.366	.399
Return Lines	8.6	130	1.885	1.157	1.052	.923	.87	.761
Return Lines	4.3	415	4.36	2.78	2.78	2.53	2.41	2.7
Return Lines	2.6	140	1.12	.77	.77	.77	.812	.91
Return Lines	1.3	65	.292	.319	.338	.344	.377	.423
Return Lines Sub Total			15.804	10.047	9.319	8.518	8.235	8.933

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-4A
33

SYSTEM SIZE MEDIUM MAIN HYDRAULIC SYSTEM (WEIGHT)

NAME AND DESCRIPTION	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Hose	13.1	180	7.65	5.13	4.95	5.22	5.85	8.20
Hose	3.7	54	1.51	.89	.89	.89	1.08	1.672
Hose	4.3	132	3.7	2.18	2.18	2.18	2.64	4.09
Hose	.5	240	3.12	3.12	3.12	3.12	4.08	5.04
Hose	2.05	144	1.87	1.87	1.87	1.87	2.45	3.02
Hose Sub Total			17.85	13.19	13.01	13.28	16.1	22.022
Total			294.032	213.665	205.462	206.779	235.587	277.941
Weight Less Plumbing			228.62	164.05	157.64			

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

SYSTEM SIZE MEDIUM MAIN HYDRAULIC SYSTEM (SPACE)

NAME AND DESCRIPTION	QTY	HP (WORK)*	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Pump	1	13.1	288	288	288	288	288	288
Pump	1	3.7	136	136	136	136	136	136
Reservoir($430 IN^3$)			2680	1490	1192	1013	776	657
Accumulator($100 IN^3$)	1	300000*	375	185	142	112	84	56
Eng. Disconnects	2	13.1	12	6.25	5.0	4.5	4.12	4.1
Eng. Disconnects	2	3.7	6.5	3.25	2.8	2.6	2.6	2.6
Eng. Disconnects	2	.003	3.75	2.5	2.25	2.25	2.25	2.25
Grnd. Test Disconnects	2	13.1	11.05	5.75	4.6	4.14	3.8	3.77
Purge Valve	1	13.1	2.4	1.25	1.0	.9	.825	.82
Purge Valve	1	3.7	1.3	.65	.56	.54	.54	.54
Filter	2	13.5	218	132	90	72	50	40
Filter	2	4.3	78	50	38	32	26	20
Relief Valve	1	14.9	68.3	36.4	28.6	26.0	22.35	22.35
Relief Valve	1	1.5	19.5	13	12.7	12.7	12.7	12.7
Control Valve Solenoid	1	13.5	153.8	80	64	57.6	52.8	52.5
Control Valve Solenoid	1	4.1	8.32	4.16	35.85	34.6	34.6	34.6
Control Valve Solenoid	1	1.35	48	32	28.8	28.8	28.8	28.8
Control Valve Manual	2	4.3	117	58.5	50.4	48.6	48.6	48.6
Control Valve Manual	1	13.2	108	56.2	45.0	40.5	37.2	36.9
Control Valve Manual	1	11.5	99.0	51.7	42.8	30.2	35.1	35.1
Brake Control Valve	2	1.3	67.5	45.0	40.5	40.5	40.5	40.5
Shut Off Valve	1	8.6	117.8	62.0	51.2	46.1	43.5	43.5
Shut Off Valve	2	13.2	207	160	128	115	105.5	105
Sequence Valve	1	1.2	33.8	22.5	20.25	20.25	20.25	20.25
Sequence Valve	3	.5	101.5	67.5	60.7	60.7	60.7	60.7
Sequence Valve	2	4.4	117	58.5	50.4	48.6	48.6	48.6
Check Valve	4	13.1	9.6	5.0	4.0	3.6	3.3	3.28
Check Valve	1	3.7	1.3	.65	.56	.54	.54	.54
Check Valve	2	.003	1.5	1.0	.90	.90	.90	.90
Check Valve	1	31.0	4.55	2.35	1.8	1.6	1.4	1.37
Check Valve	4	4.3	5.2	2.6	2.24	2.16	2.16	2.16
Check Valve	2	11.5	4.4	2.8	1.9	1.7	1.56	1.56
Check Valve	2	2.6	1.8	1.1	1.0	1.0	.98	.98
Check Valve	2	1.35	1.5	1.0	.90	.90	.90	.90

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-4B
35

SYSTEM SIZE MEDIUM MAIN HYDRAULIC SYSTEM (SPACE)

NAME AND DESCRIPTION	QTY	HP (WORK)*	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Priority Valve	1	13.5	108	56.2	45.0	40.5	37.2	36.9
Fuse	1	11.5	2.2	1.15	.95	.85	.78	.77
Fuse	3	1.6	2.25	1.5	1.35	1.35	1.35	1.35
Fuse	1	Min	1.35	1.35	1.35	1.35	1.35	1.35
Fuse	2	13.5	4.8	2.5	2.0	1.8	1.65	1.64
Fuse	2	4.1	2.6	1.3	1.12	1.08	1.08	1.08
Restrictor (1Way)	1	13.5	2.4	1.25	1.0	.9	.825	.82
Restrictor (2Way)	2	2.05	1.5	1.0	.90	.90	.90	.90
Restrictor (2Way)	1	3.35	1.3	.65	.56	.54	.54	.54
Gage Snubber	1		.75	.5	.45	.45	.45	.45
Valves Sub Total			1856.92	1032.06	872.39	779.2	739.2	721.27
Cylinders	2	28400*	152	116	99	94	93	114
Cylinder	1	175000*	425	260	240	230	235	268
Cylinders	3	6600*	66	55.2	48	45.6	45	52.5
Cylinders	1	15500*	43	35.5	30.5	29	29.5	34.5
Cylinders	2	57000*	296	204	178	170	174	204
Cylinders	1	322000*	760	440	410	390	405	406
Cylinders	2	54500*	280	194	170	164	166	196
Cylinders	12	5950	288	204	176.4	168	168	195.6
Cylinders	2	58800*	296	204	178	170	174	204
Cylinders	2	27900*	152	116	90	94	93	114
Cylinder Sub Total			2698.0	1828.7	1619.9	1565.6	1562.5	1842.6

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-4B
36

SYSTEM SIZE MEDIUM MAIN HYDRAULIC SYSTEM (SPACE)

NAME AND DESCRIPTION	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Pressure Lines	13.1	515	139	67.0	48.9	41.1	37.1	45
Pressure Lines	14.9	25	7.5	3.62	2.75	2.375	2.0	2.75
Pressure Lines	31.0	15	8.1	3.9	3.0	2.475	2.1	2.203
Pressure Lines	8.6	110	22	9.9	7.7	6.6	5.78	7.7
Pressure Lines	4.3	420	50.4	21	16.8	16.8	16.8	20.15
Pressure Lines	1.3	2010	101	60.3	60.3	60.3	60.3	60.3
Pressure Lines	2.05	495	29.7	14.85	14.85	14.85	14.85	14.85
Pressure Lines	2.6	140	11.2	5.6	4.9	4.9	4.9	5.18
Pressure Lines	11.5	35	8.75	3.675	2.975	2.625	2.275	2.87
Pressure Lines	9.8	25	5.5	2.5	1.875	1.625	1.5	1.90
Pressure Lines Sub Total			383.15	192.345	164.05	153.65	147.605	162.903
Supply Lines	13.1	90	36.5	24.0	21.0	18.5	16	14.75
Supply Lines	3.5	10	3.8	3.2	3.0	2.85	2.85	2.85
Supply Lines Sub Total			40.3	27.2	24.0	21.35	18.85	17.60
Return Lines	13.1	430	116	51.6	36.5	30.1	21.5	19.34
Return Lines	31.0	30	15.75	6.9	4.8	3.9	2.85	2.4
Return Lines	8.6	130	26	10.4	7.8	7.15	5.85	5.2
Return Lines	4.3	415	49.8	20.78	16.6	14.53	12.45	12.45
Return Lines	2.6	140	12.6	5.6	4.2	4.2	4.2	4.2
Return Lines	1.3	65	5.85	2.6	1.95	1.95	1.95	1.95
Return Lines Sub Total			225	97.88	71.85	61.83	48.8	45.54
Hose	13.1	180	413	198.8	145.3	122.2	110	133.6
Hose	3.7	186	189.5	79	63.2	63.2	63.2	75.9
Hose	.5	240	61.2	61.2	61.2	61.2	61.2	61.2
Hose	2.05	144	73.5	36.7	36.7	36.7	36.7	36.7
Hose Sub Total			737.2	375.3	306.4	283.3	271.1	307.4
Total			9419.51	5652.485	4816.59	4433.93	4072.055	4234.313

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

SYSTEM SIZE LARGE UTILITY SYSTEM (WEIGHT)

NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Pump	3	14.25		42	42	42	42	42	42
Reservoir (Tank Type)	1	---		740.7	373.7	282.7	226.7	153.7	117.7
Accumulator	3	1.5 (10^5)*		17.7	16.5	16.5	16.2	17.3	18.6
Purge Valve	3	14.25		.97	.87	.92	.98	1.23	1.69
Filter	2	34.1		6.20	6.20	6.20	6.20	6.20	6.20
Filter	3	1.051		1.68	1.68	1.68	1.68	1.68	1.68
Filter	1	28.5		2.84	2.84	2.84	2.84	2.84	2.84
Filter	1	14.25		2.00	2.00	2.00	2.00	2.00	2.00
Relief Valve	1	44		3.10	3.05	3.18	3.45	4.29	5.72
Relief Valve	1	52		4.09	3.96	4.16	4.49	5.65	7.54
Control Valve, 4W, Man. Op	2	34.1		5.59	5.36	5.59	6.04	7.53	9.92
Control Valve, 4W, Sol. Op	1	81.4		5.70	5.43	5.77	6.30	7.91	10.85
Control Valve, 3W, Sol. Op	3	1.05		3.42	2.41	3.01	3.42	4.62	5.62
Control Valve, 4W, Sol. Op	1	147		8.77	8.24	9.10	10.05	12.53	18.10
Control Valve, 4W, Man. Op	1	73.4		4.51	4.33	4.56	4.96	6.27	8.55
Control Valve, Man. Op	2	4.65		2.28	1.99	2.16	2.34	2.62	4.44
Sequence Valve	3	15.8		6.74	6.13	6.43	6.84	8.74	10.77
Sequence Valve	2	33.1		6.43	6.16	6.43	6.97	8.71	11.52
Sequence Valve	1	7.3		1.61	1.47	1.54	1.61	2.21	3.02
Check Valve	3	68.2		2.25	2.18	2.19	2.49	3.14	4.26
Check Valve	2	34.1		.98	.94	.98	1.06	1.32	1.76
Check Valve	2	81.4		1.70	1.62	1.72	1.88	2.36	3.24
Check Valve	2	3.7		.36	.32	.36	.39	.54	.70
Check Valve	6	1.051		1.02	.72	.90	1.02	1.38	1.08
Check Valve	2	42.8		1.12	1.08	1.13	1.22	1.52	2.04
Check Valve	2	147		2.62	2.46	2.74	3.00	3.74	5.40
Check Valve	2	78.4		1.5[illegible]	1.52	1.60	1.74	2.20	3.00
Check Valve	2	9.3		.54	.48	.50	.53	.72	.96
Shuttle Valve	2	1.85		.45	.39	.45	.51	.69	.87
Shuttle Valve	2	40.7		1.62	1.56	1.64	1.77	2.22	2.97
Shuttle Valve	2	73.5		2.37	2.28	2.40	2.61	3.30	4.50
Fuse	1	4.65		.20	.18	.19	.21	.23	.39
Valve Sub Total				82.82	77.85	82.45	88.60	108.39	142.23

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

SYSTEM SIZE LARGE UTILITY SYSTEM (WEIGHT)

NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Cylinder	2	1.835(10^5)*		30.2	17.6	17.8	19.4	24.6	30.4
Cylinder	2	6.8(10^5)*		96	66	67	72	90	110
Cylinder	2	2.69(10^5)*		42	26.4	26.6	28.8	36	44
Cylinder	2	2.14(10^6)*		286	212	214	228	290	356
Cylinder	3	2.09(10^5)*		49.5	30.9	30.9	33.9	42.6	50.7
Cylinder	2	3.13(10^6)*		404.0	312.0	314.0	334.0	440.0	540.0
Cylinder	1	4.73(10^5)*		34.2	23.2	23.3	25.0	32.0	38.0
Cylinder Sub Total				944.9	688.1	693.6	741.1	955.2	1169.1
Pressure Lines	1	14.3	427	7.65	5.13	5.13	5.13	5.98	7.65
Pressure Lines	1	28.6	56	1.40	1.08	1.06	1.10	1.40	1.53
Pressure Lines	1	41.0	220	6.86	5.44	5.41	5.63	6.20	7.52
Pressure Lines	1	55.3	227	8.76	7.09	6.90	7.18	7.99	9.37
Pressure Lines	1	27.6	394	9.54	7.41	7.25	7.49	8.28	10.57
Pressure Lines	1	18.7	197	3.94	2.86	2.80	2.86	3.19	4.26
Pressure Lines	1	3.85	26	.26	.16	.16	.16	.20	.30
Pressure Lines	1	1.92	1408	9.44	5.91	5.91	5.91	9.57	11.82
Pressure Lines	1	42.9	369	11.80	9.59	9.37	9.74	10.78	13.05
Pressure Lines	1	47.3	340	9.52	9.25	9.25	9.58	10.68	12.92
Pressure Lines	1	23.7	502	10.75	8.54	8.34	8.64	9.55	12.36
Pressure Lines	1	11.5	138	2.21	1.44	1.44	1.44	1.66	2.40
Pressure Lines	1	17.2	270	5.91	3.72	3.62	3.75	4.16	5.67
Pressure Lines	1	1.05	1154	7.85	4.85	4.85	4.85	7.86	9.70
Pressure Lines	1	9.3	310	4.65	2.91	2.91	2.91	3.41	5.02
Pressure Lines	1	4.65	2560	25.60	15.37	15.37	15.37	19.98	29.70
Pressure Lines	1	73.4	85	4.01	3.22	3.16	3.28	3.69	4.41
Pressure Lines	1	15.8	575	10.52	7.25	7.25	7.25	8.40	11.50
Pressure Lines	1	7.3	90	1.24	.74	.74	.74	.88	1.33
Pressure Lines	1	66.2	30	1.31	1.06	1.04	1.08	1.21	1.45
Pressure Lines	1	33.1	1260	34.26	27.22	26.70	27.70	30.24	37.80
Sub Total				178.93	130.51	128.66	131.79	155.31	200.33

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-5A

SYSTEM SIZE LARGE UTILITY SYSTEM (WEIGHT)

NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Supply Lines	1	52	381	15.24	12.95	12.78	13.30	14.29	16.00
Supply Lines	1	17.5	486	12.13	10.48	10.20	10.34	11.42	13.69
Supply Lines	1	35	6	.20	.23	.23	.17	.19	.21
Supply Lines	1	1.4	447	5.68	6.04	6.71	7.27	8.84	6.04
Supply Lines	1	2.8	97	1.31	1.34	1.45	1.58	1.92	1.55
Supply Sub Total				34.56	31.04	31.37	32.62	36.66	37.49
Return Lines	1	3.85	5	.05	.03	.03	.03	.03	.03
Return Lines	1	40.6	430	12.03	7.52	6.58	6.41	6.27	6.71
Return Lines	1	23.6	117	2.58	1.58	1.36	1.29	1.25	1.35
Return Lines	1	47.2	353	10.66	6.71	5.89	5.79	5.68	6.03
Return Lines	1	1.05	1154	5.77	5.77	5.09	6.12	6.81	7.51
Return Lines	1	9.3	310	4.65	2.94	2.38	2.26	2.17	2.61
Return Lines	1	4.65	80	.84	.54	.53	.49	.47	.52
Return Lines	1	73.4	35	1.34	.87	.79	.77	.75	.81
Return Lines	1	15.8	345	6.38	3.93	3.24	3.11	3.00	3.38
Return Lines	1	7.3	40	.54	.34	.31	.27	.25	.28
Return Lines	1	60.2	25	.91	.58	.53	.51	.50	.54
Return Lines	1	33.1	800	20.40	12.80	10.95	10.55	10.32	10.95
Return Sub Total				66.05	43.61	38.48	35.71	37.50	40.72
Total				2104.66	1403.31	1315.76	1314.72	1506.06	1768.17

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-58

40

SYSTEM SIZE LARGE UTILITY SYSTEM (SPACE)

NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Pump	3	14.25		912	912	912	912	912	912
Reservoir (Tank Type)	1	----		4200	2540	2050	1750	1300	1040
Accumulator	3	150000*		285	135	105	84	54	42
	3	14.25		7.65	4.05	3.3	2.7	2.58	2.58
Purge Valve	2	34.1		534	310	210	164	108	81
Filter	3	1.051		3	2.4	2.25	2.10	2.00	1.95
Filter	1	28.5		221	132	88	69	46.5	35
Filter	1	14.25		115	69.5	47	38	26	20
Filter	1	44		158.5	79.2	62.4	54.6	46.5	45.8
Relief Valve	1	52		184.5	91.0	72.8	62.4	53	52.3
Relief Valve	2	34.1		627.0	320	250	224	191	178
Control Valve,4W,Man Op.	1	81.4		450	225	175.5	155	126	125
Control Valve,4W,Sol Op.	3	1.05		94.5	67.5	60.7	60.7	60.7	60.7
Control Valve,3W,Sol Op.	1	147		828	373.5	292	247.5	177	176
Control Valve,4W,Sol Op.	1	73.4		620	294	230.5	205	166	164.5
Control Valve,4W,Man Op.	2	4.65		160	89.6	76.8	70.4	70.4	70.4
Control Valve,Man Op.	3	15.8		72.9	37.8	31.05	27.0	24.6	24.3
Sequence Valve	2	33.1		86.5	45	34.2	30.6	26.3	25.9
Sequence Valve	1	7.3		17.9	9.45	7.14	7.2	6.39	6.29
Sequence Valve	3	68.2		27.0	13.8	10.8	9.45	7.35	7.26
Check Valve	3	34.1		14.7	7.5	5.85	5.25	4.47	4.41
Check Valve	2	81.4		20.0	10.0	7.8	6.9	5.6	5.56
Check Valve	2	3.7		2.2	1.1	1.0	1.0	1.04	1.04
Check Valve	6	1.051		4.2	3.0	2.7	2.7	2.7	2.7
Check Valve	2	42.8		12.2	6.1	4.8	4.2	3.58	3.52
Check Valve	2	147		36.8	16.6	13.0	11.0	7.88	7.84
Check Valve	2	73.4		19.4	9.2	7.2	6.4	5.18	5.14
Check Valve	2	9.3		3.8	2.1	1.7	1.6	1.42	1.4
Shuttle Valve	2	1.85		12.6	9.0	8.1	8.1	8.1	8.1
Shuttle Valve	2	40.7		110.0	54.9	43.2	37.0	32.2	31.7
Shuttle Valve	2	73.5		174.5	82.9	64.9	57.6	46.6	46.3
Fuse	1	4.65		1.25	.7	.6	.55	.55	.55
Valve Sub Total				5278.2	2366.9	1814.29	1562.75	1259.64	1249.12

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-5B
41

SYSTEM SIZE LARGE UTILITY SYSTEM (SPACE)

NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Cylinder	2	$1.835(10^5)$*		900	544	500	480	500	540
Cylinder	2	$6.8(10^5)$*		3060	1780	1640	1540	1620	1840
Cylinder	2	$2.69(10^5)$*		1280	760	700	670	696	770
Cylinder	2	$2.14(10^6)$*		8800	5400	5100	4400	4800	6000
Cylinder	3	$2.09(10^5)$*		1000	605	560	540	568	625
Cylinder	2	$3.13(10^6)$*							
Cylinder	1	$4.73(10^5)$*		1080	630	580	560	580	655
Cylinder Sub Total				16120	9719	9080	8190	8764	10430
Pressure Lines	1	14.3	427	123.8	59.8	44.9	38.4	35.0	40.1
Pressure Lines	1	28.6	56	28.6	13.72	10.08	8.96	7.34	7.64
Pressure Lines	1	41.0	220	152.0	72.5	55.0	46.2	39.2	39.4
Pressure Lines	1	55.3	227	195.0	92	70.4	59.0	49.9	49.3
Pressure Lines	1	27.6	394	201.0	96.6	71.0	63.1	51.6	55.1
Pressure Lines	1	18.7	197	71.0	33.5	25.6	21.65	18.3	20.85
Pressure Lines	1	3.85	26	2.86	1.43	1.17	1.04	.806	.961
Pressure Lines	1	1.92	1408	84.5	49.3	42.3	42.3	38.0	38.0
Pressure Lines	1	42.9	369	254	123	92.1	77.5	65.6	66
Pressure Lines	1	47.3	340	265	126	95.2	81.6	68	68.9
Pressure Lines	1	23.7	502	218.5	105.7	80.5	67.9	56.3	62.4
Pressure Lines	1	11.5	138	34.5	15.87	11.8	11.02	8.96	11.48
Pressure Lines	1	17.2	270	91.9	43.2	32.4	28.4	24.3	28.4
Pressure Lines	1	1.05	1154	69.0	40.2	34.5	32.0	31.0	31.0
Pressure Lines	1	9.3	310	63.5	31.0	23.25	20.18	17.85	22.6
Pressure Lines	1	4.65	2560	307	153.5	128.0	115.1	107.5	128.0
Pressure Lines	1	73.4	85	100.2	45.9	35.7	30.6	25.05	24.3
Pressure Lines	1	15.8	575	184	86.3	63.3	57.5	47.7	56.9
Pressure Lines	1	7.5	90	16.2	7.2	5.85	5.4	4.59	5.85
Pressure Lines	1	66.2	30	32.1	14.85	11.4	9.45	8.1	7.85
Pressure Lines	1	33.1	1260	719	252.5	264.5	22.05	18.64	18.78
Sub Total				3213.66	1463.97	1198.95	839.35	723.9	784.011

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-5B
42

SYSTEM SIZE LARGE UTILITY SYSTEM (SPACE)

NAME AND DESCRIPTION	QTY	HF (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
Supply Lines	1	52	381	594	400	343	308.5	244	209.5
Supply Lines	1	17.5	486	414	267.5	233.5	206.5	175.0	160.5
Supply Lines	1	35	6	73.5	4.92	4.2	3.78	3.06	2.67
Supply Lines	1	1.4	447	127	127	127	127	127	127
Supply Lines	1	2.8	97	32.0	29.1	27.6	27.6	27.6	27.6
Supply Sub Total				1240.5	828.52	735.3	673.38	476.66	527.27
Return Lines	1	3.85	5	.65	.3	.225	.2	.175	.15
Return Lines	1	40.6	430	284.0	120.5	86.0	68.8	49.4	43.0
Return Lines	1	23.6	117	49.1	21.05	15.2	11.7	9.36	7.6
Return Lines	1	47.2	353	264.5	113.0	79.5	63.5	45.9	38.8
Return Lines	1	1.05	1154	462	347	347	347	347	347
Return Lines	1	9.3	310	65.1	27.9	21.7	17.05	13.95	12.4
Return Lines	1	4.65	80	10.4	4.8	3.6	3.2	2.8	2.4
Return Lines	1	73.4	35	36.9	16.1	11.9	9.1	6.3	5.6
Return Lines	1	15.8	345	107	46.5	34.5	27.6	20.7	19.0
Return Lines	1	7.3	40	7.2	3.2	2.4	2.0	1.6	1.4
Return Lines	1	66.2	25	25.5	10.5	7.62	6.0	4.12	3.55
Return Lines	1	33.1	800	448	192	136	108	80.0	65.6
Return Sub Total				1257.15	902.85	745.645	664.15	580.305	546.5
Total				32506.51	18868.24	1661618	14675.63	14086.5	155309

* WORK IN INCH LBS. SHOWN BY AN ASTERISK

TABLE 2-6A

43

SYSTEM SIZE POWER PACK HYDRAULIC SYSTEM (WEIGHT)

NO.	NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
1	Pump	1	14.0		13.8	13.8	13.8	13.8	13.8	13.8
2	Pump	1	.8		4.2	4.2	4.2	4.2	4.2	4.2
3	Reservoir	1			44.2	25.2	20.75	17.95	14.2	12.32
4	Accumulator (200 IN³)	1	600,000*		18.4	16.6	13.8	16.6	18.4	18.8
5	Motor	1	48.6		15.1	15.1	15.1	15.1	15.1	15.1
6	Grnd Disconnect	1	14.0		.492	.446	.461	.474	.639	.861
7	Grnd Disconnect	1	48.6		.93	.697	.934	1.015	1.269	1.7
8	Filter	1	14.0		2.0	2.0	2.0	2.0	2.0	2.0
9	Relief Valve	1	14.0		2.08	1.883	1.95	2.0	2.695	3.64
10	Relief Valve	1	1.75		.844	.78	.941	1.072	1.4	1.56
11	Control Valve	1	48.6		3.45	3.32	3.46	3.76	4.7	6.31
12	Check Valve	1	47.8		.595	.575	.60	.65	.815	1.055
13	Check Valve	1	.805		.11	.10	.14	.155	.18	.22
14	Check Valve	1	14.0		.32	.29	.30	.308	.415	.56
15	Pressure Control Valve	1	14.0							
16	Pressure Switch	1			.20	.20	.20	.20	.20	.20
17	Valves Sub Total				11.021	10.491	10.986	11.634	14.313	18.106

*WORK IN INCH LBS SHOWN BY AN ASTERISK

TABLE 2-6A
44

SYSTEM SIZE POWER PACK HYDRAULIC SYSTEM (WEIGHT)

NO.	NAME AND DESCRIPTION	QTY	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
1	Pressure Lines		14.0	110	1.925	1.3	1.275	1.32	1.495	2.09
2	Pressure Lines		34.6	10	.279	.222	.216	.226	.216	.306
3	Pressure Lines		48.6	24	.84	.681	.662	.691	.763	.924
4	Pressure Lines									
5	Pressure Lines									
6	Pressure Lines		.8	126	.655	.655	.655	.655	.857	1.058
7	Pressure Lines Sub Total				3.699	2.858	2.808	2.892	3.361	4.378
8	Supply Lines		.8	10	.127	.135	.15	.162	.197	.212
9	Return Lines		48.6	144	4.42	2.81	2.46	2.405	2.345	2.505
10	Return Lines		.005	10	.045	.048	.05	.054	.058	.065
11	Return Lines Sub Total				4.465	2.858	2.51	2.459	2.403	2.570
12	Hose		14.0	10	.437	.2945	.29	.30	.34	.475
13	Hose		.005	8	.104	.104	.104	.104	.136	.168
14	Hose		48.6	12	1.05	.851	.828	.864	.954	1.155
15	Hose Sub Total				1.591	1.2495	1.222	1.268	1.430	1.798
16	Total				16.603	92.491	85.326	86.065	87.404	91.314
17										
18										
19										

*WORK IN INCH LBS SHOWN BY AN ASTERISK

TABLE 2-68

45

SYSTEM SIZE POWER PACK HYDRAULIC SYSTEM (SPACE)

NO.	NAME AND DESCRIPTION	QTY	HP (WORK)*	LGH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
1	Pump	1	14.0		302	302	302	302	302	302
2	Pump	1	.8		90	90	90	90	90	90
3	Reservoir	1			1800	1070	880	560	450	443
4	Accumulator (200 IN^3)	1	600000*		745	375	280	223	168	112
5	Motor	1	48.6		330	330	330	330	330	330
6	Grnd Disconnect	1	14.0		5.75	3.035	2.415	2.185	2.07	1.955
7	Grnd Disconnect	1	48.6		15.4	7.59	5.98	5.18	4.42	4.35
8	Filter	1	14.0		114	50	35	28	20	16
9	Relief Valve	1	14.0		67.5	35.65	27.35	25.65	24.3	22.95
10	Relief Valve	1	1.75		18.9	12.15	12.15	12.15	12.7	12.7
11	Control Valve	1	48.6		302	148.5	117	101.2	86.4	85
12	Check Valve	1	47.8		6.55	3.25	2.5	2.25	1.90	1.86
13	Check Valve	1	.805		.45	.45	.45	.45	.45	.45
14	Check Valve	1	14.0		2.5	1.32	1.05	.95	.90	.85
15										
16	Pressure Switch	1			.90	.90	.90	.90	.90	.90
17	Valves Sub Total	1			533.95	262.845	204.795	178.915	154.05	147015
18										
19										

TABLE 2-6B

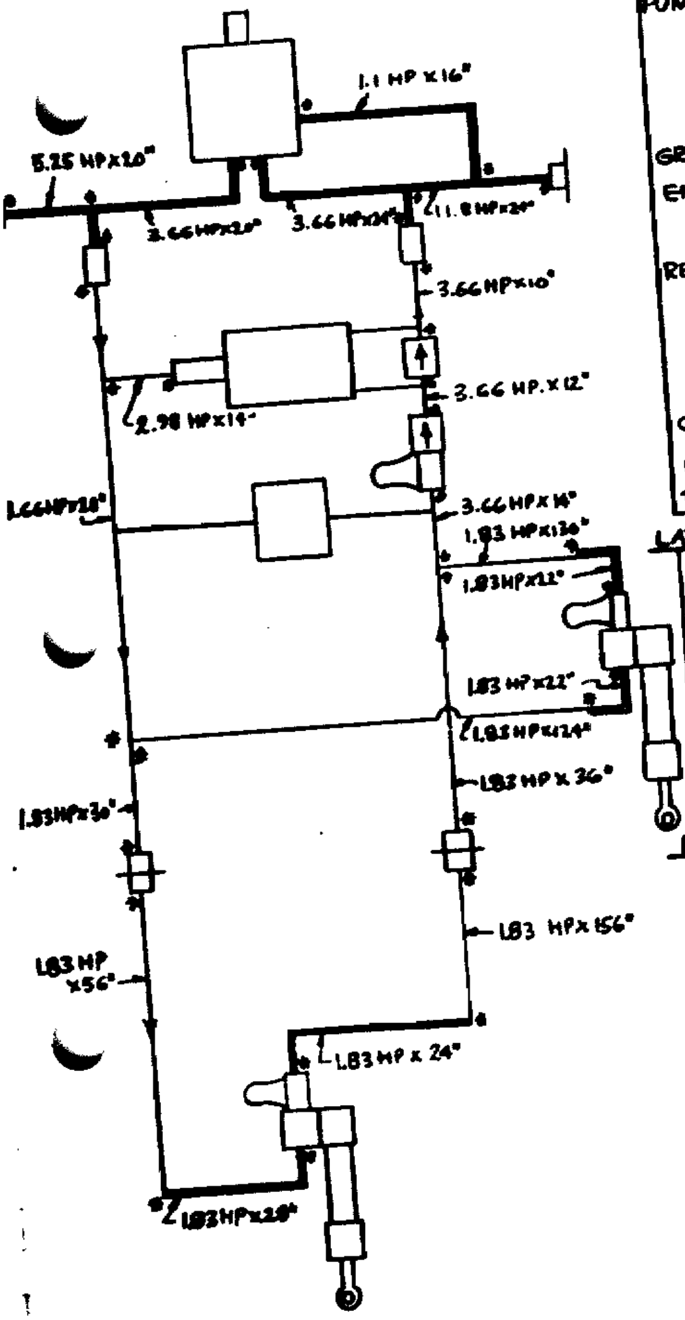
46

SYSTEM SIZE POWER PACK HYDRAULIC SYSTEM (SPACE)

NO.	NAME AND DESCRIPTION	HP (WORK)*	LGTH (IN)	1500 PSI	3000 PSI	4000 PSI	5000 PSI	7500 PSI	10000 PSI
1	Pressure Lines	14.0	110	31.9	14.85	11.0	9.9	8.47	10.15
2	Pressure Lines	34.6	10	5.9	2.85	2.15	1.8	1.53	1.58
3	Pressure Lines	48.6	24	19.2	9.12	6.96	5.88	4.95	4.87
4	Pressure Lines	.8	126	29.0	29.0	29.0	29.0	29.0	29.0
5	Pressure Lines Sub Total			86.0	54.82	49.11	46.58	43.95	45.6
6	Supply Lines Sub Total	.8	10	2.85	2.85	2.85	2.85	2.85	2.85
7	Return Lines	48.6	144	111.0	46.8	33.8	25.8	18.7	15.85
8	Return Lines	.005	10	.3	.3	.3	.3	.3	.3
9									
10	Return Lines Sub Total			111.3	47.1	34.1	26.1	19.0	16.15
11	Hose	14.0	10	24.6	11.45	8.5	7.64	6.54	7.81
12	Hose	.005	8	2.31	2.31	2.31	2.31	2.31	2.31
13	Hose	48.6	12	81.6	38.75	29.6	25.0	21.0	20.7
14	Hose Sub Total			108.51	52.51	42.41	34.95	29.85	30.82
15	Total			4387.76	2587.125	2215.265	1794.395	1589.7	1519.435
16									
17									
18									
19									

* WORK IN INCH LBS., SHOWN BY AN ASTERISK

SMALL FIGHTER CONTROL SYSTEM:



POWER SUPPLY

PUMP: VARIABLE VOLUME:
HYDRAULIC HP AT CRUISE 7.61 HP

GROUND TEST DISCONNECTS (2)
ENGINE DISCONNECTS (2)

RESERVOIR CAP. 100 IN^3
SYST. PRESS'R 1500 PSI
RES. PRESS'R $10^{\pm 3}$ PSI

CHECK VALVES (2) 3.66 HP
FILTER
RELIEF VALVE 3.66 HP

LATERAL CONTROL

FILTER
CONTROL VALVE 1.83 HP
SOLEN. OPER. PROPORTION'G
SPOILER CYLINDER 9430 in#

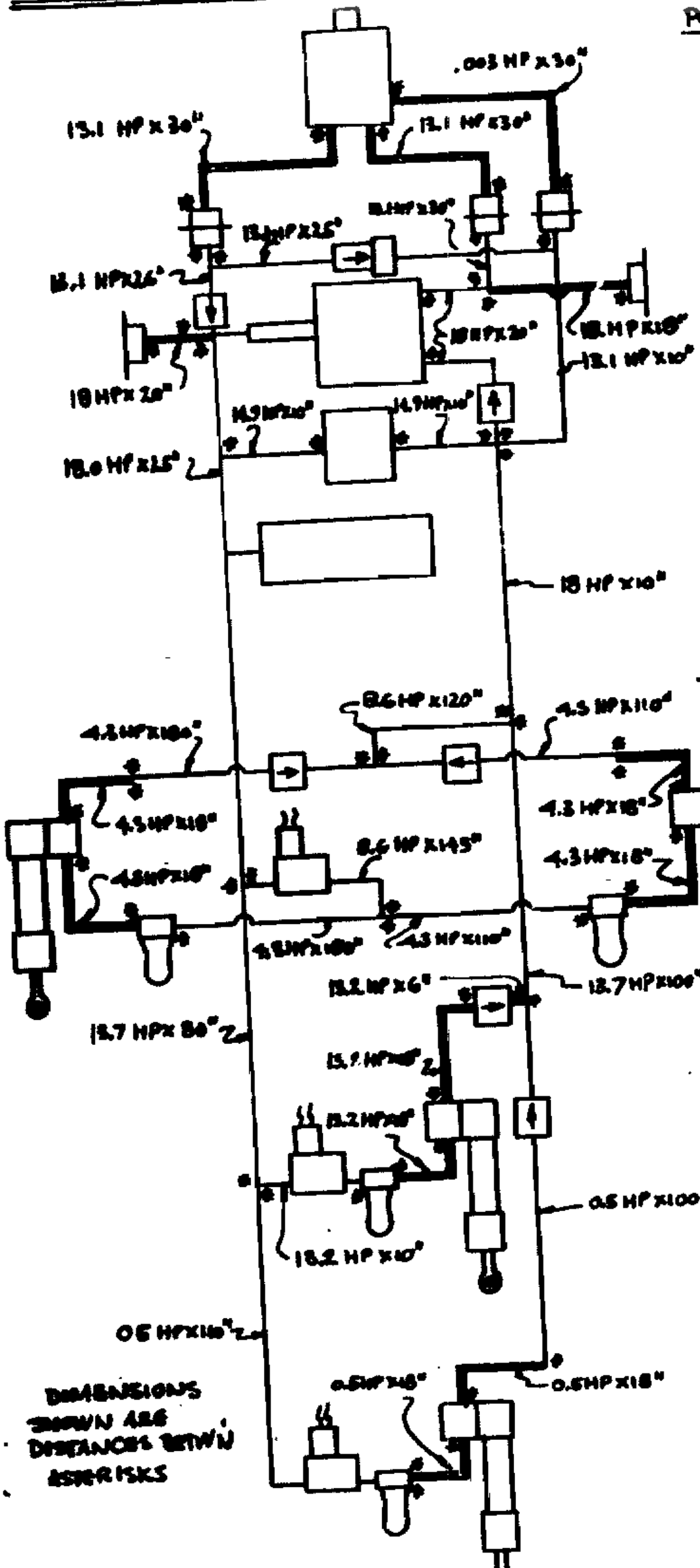
LONGITUDINAL CONTROL

STABILIZER AND FUSELAGE
DISCONNECTS (2)

FILTER
CONTROL VALVE 1.83 HP
SOLEN. OPER. PROPORTIONING
STABILIZER CYLINDER 8870 in#

DISTANCES SHOWN ARE BETWEEN ASTERISKS

MEDIUM FLIGHT CONTROL HYDRAULIC SYSTEM



POWER SUPPLY

PUMP, VAR. VOLUME	
HYDRAULIC HP	13.1 HP
ENGINE DISCONNECT (2)	
PURGE VALVE	13.1 HP
RESERVOIR	231 IN^3
SYSTEM PRESSURE 3000 PSI	
GROUND TEST STA (2)	
CHECK VALVE (2)	13.1 HP
CHECK VALVE (2)	18. HP
RELIEF VALVE	14.9 HP
ACCUMULATOR	300,000 IN^4

SHUT OFF VALVE	8.6 HP
SOLENOID OPERATED	
FILTER (2)	
CONTROL VALVE (2)	4.3 HP.
MANUALLY OPERATED 4WAY	
CYLINDER (2)	28000 IN$^{\#}$
CHECK VALVES (2)	4.3 HP

LONGITUDINAL CONTROL

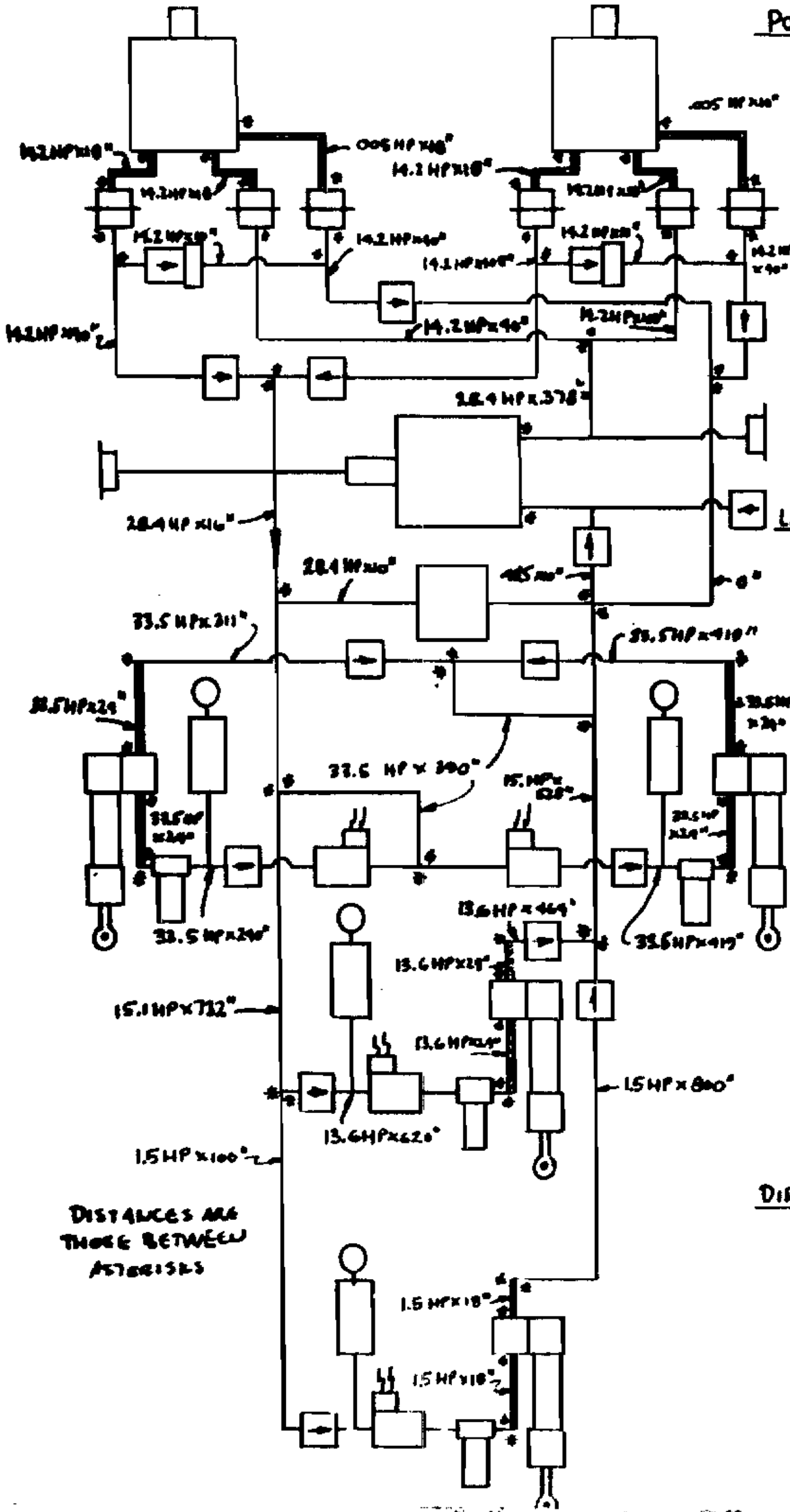
SHUTOFF VALVE	
SOLENOID OPERATED	
FILTER	
CONTROL VALVE	13.25 HP
MANUALLY OPERATED 4WAY	
CYLINDER	175000 IN$^{\#}$
CHECK VALVE	13.25 HP

DIRECTIONAL CONTROL

SHUTOFF VALVE	.428 HP
SOLENOID OPERATED	
FILTER	
CONTROL VALVE	.418 HP
MANUALLY OPERATED 4WAY	
CYLINDER	2820 IN$^{\#}$
CHECK VALVE	.418 HP

LARGE FLIGHT CONTROL SYSTEM

FIGURE 2-3



POWER SUPPLY	
PUMPS, VARIABLE VOL. (2)	
HYDRAULIC H.P.	14.2HP
ENGINE DISCONNECT(6)	
PURGE VALVE (2)	14.2HP
CHECK VALVE (2)	14.2HP
CHECK VALVE (2)	14.2HP
DISCONNECT (2)	
RESERVOIR	
TANK TYPE	1045IN³
CHECK VALVE	
CHECK VALVE	
RELIEF VALVE	28.4HP
LATERAL CONTROL	
SHUTOFF VALVES SOLEN. OPER.	33.5HP
CHECK VALVE (2)	33.5HP
CHECK VALVE (2)	33.5HP
ACCUMULATORS(2)	450,000**
FILTERS (2)	33.5HP
CONTR. VALVES MAN. OPER (2)	33.5
CYLINDER (2)	222000**
LONGITUDINAL CONTROL	
SHUTOFF VALVE SOLEN. OPER.	13.6HP
CHECK VALVE	13.6HP
CHECK VALVE	13.6 HP
ACCUMULATOR	300000**
FILTER	13.6HP
CONTR. VALVE, MAN. OPER	13.6HP
CYLINDER	174000**
DIRECTIONAL CONTROL	
SHUTOFF VALVE SOLEN. OPER.	1.5HP
CHECK VALVE	1.5HP
CHECK VALVE	1.5HP
ACCUMULATOR	300,000**
FILTER	1.5HP
CONTR. VALVE, MAN. OPER.	1.5HP
CYLINDER	9700**

MEDIUM MAIN SYSTEM

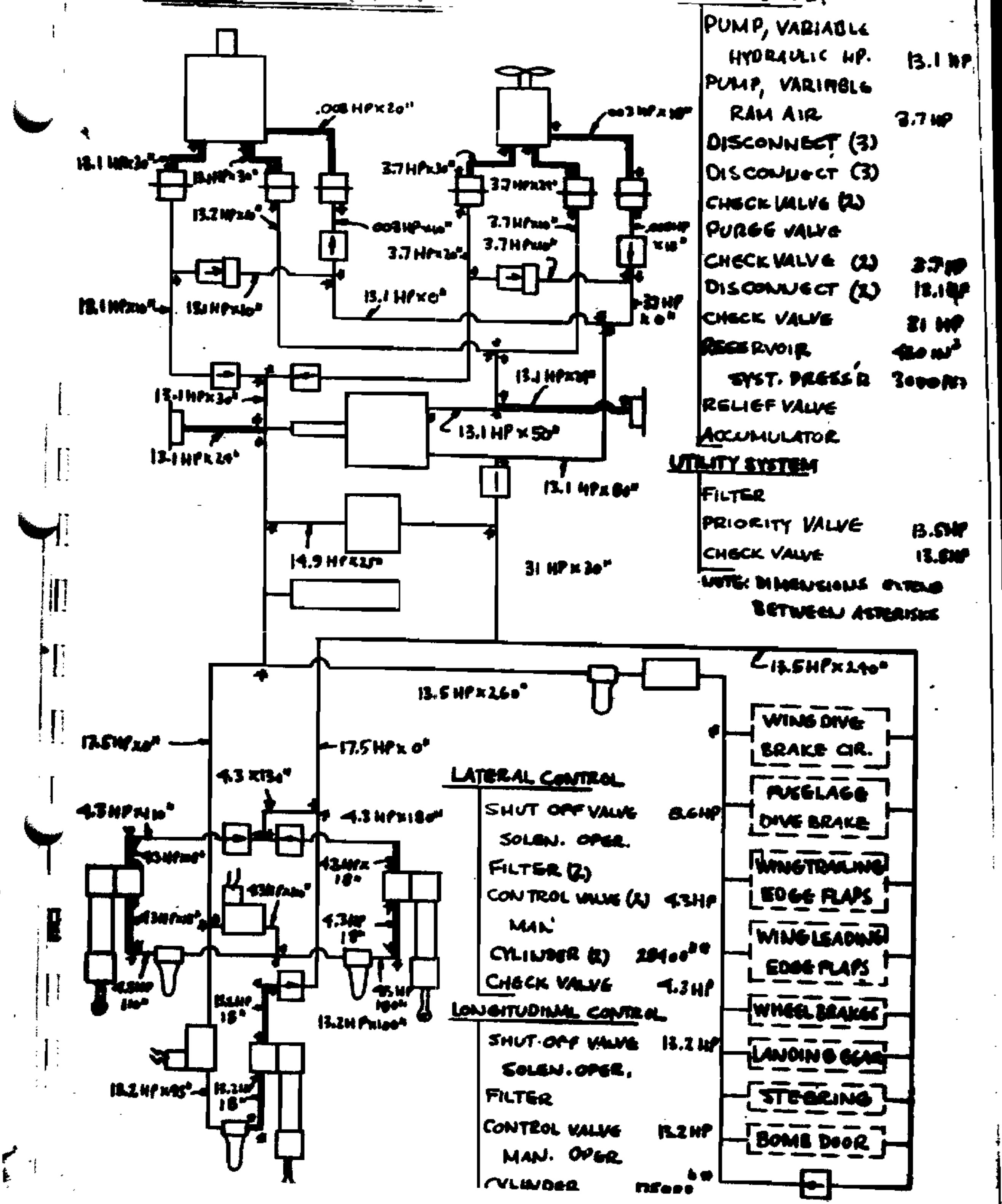
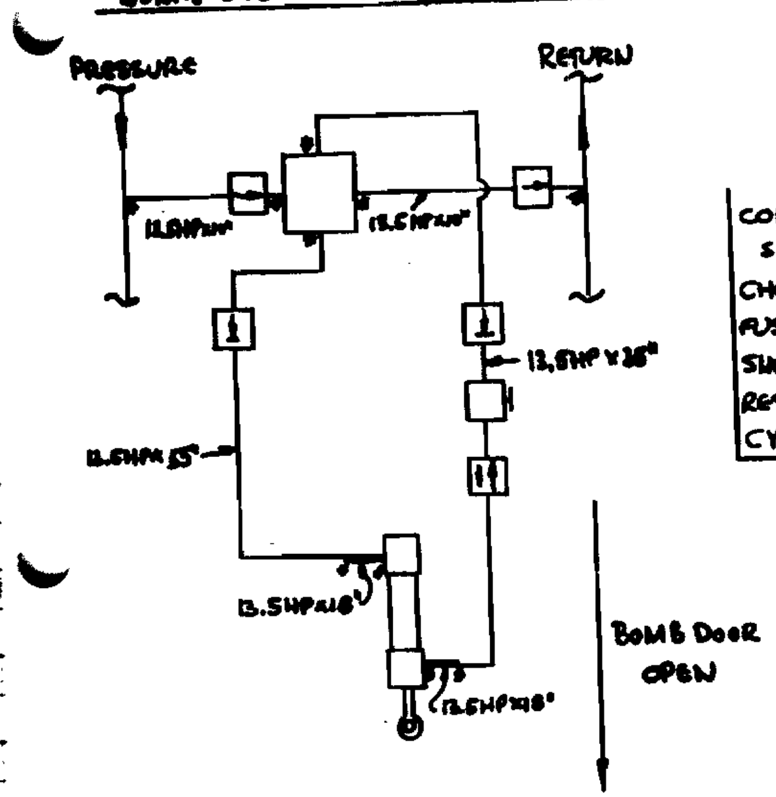


FIGURE 2-43

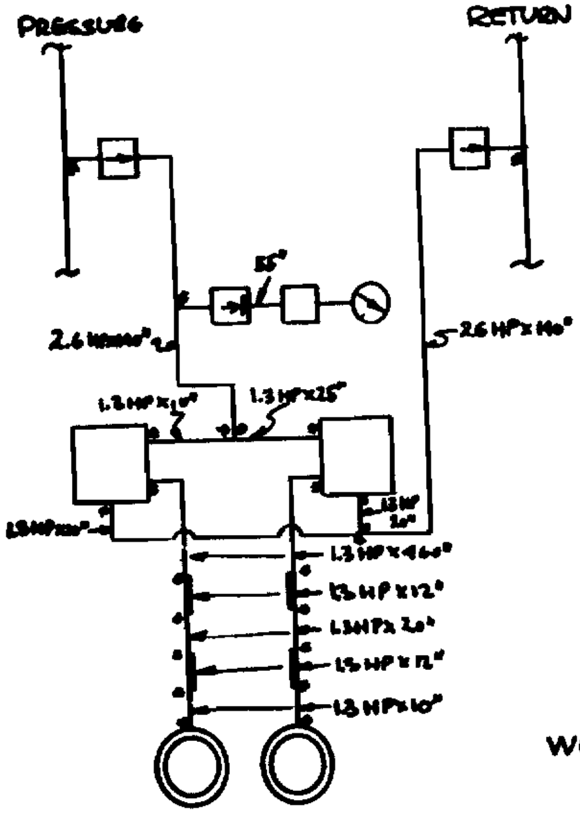
51

MEDIUM MAIN HYDRAULIC SYSTEM
BOMB DOOR UTILITY CIRCUIT



CONTROL VALVE SOLEN. OPER.	13.5HP
CHECK VALVE (2)	13.5HP
FUSE (2)	13.5HP
SHUTOFF VALVE	13.5HP
RESTRICTOR	13.5HP
CYLINDER	322000

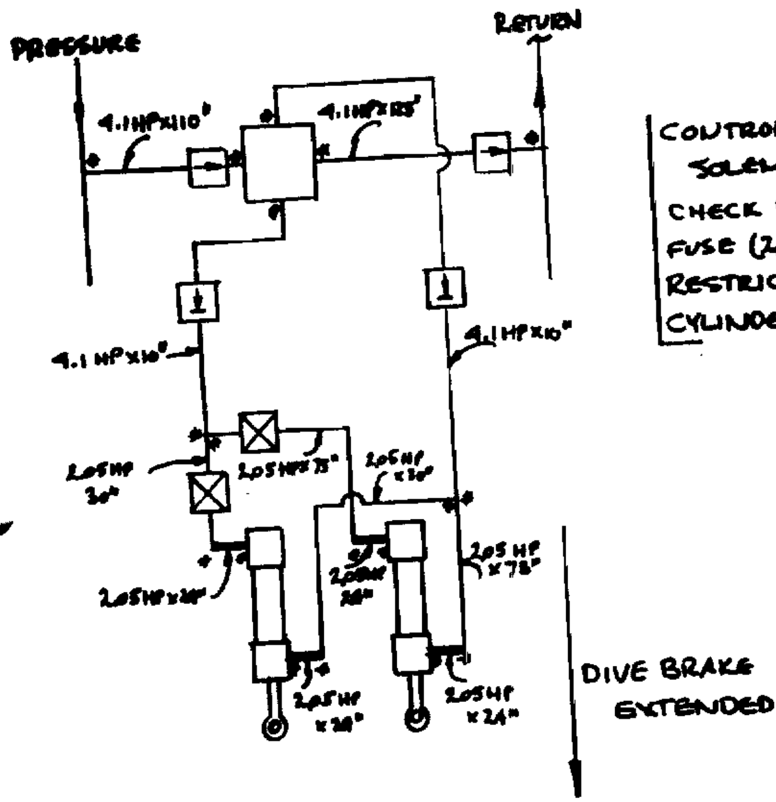
MEDIUM MAIN HYDRAULIC SYSTEM
WHEEL BRAKES UTILITY CIRCUIT



CHECK VALVE (2) 2.6HP
FUSE
GAGE SNUBBER
PRESSURE GAGE
BRAKE CONTROL VALVES (2) 1.3HP
MAN. OPER.

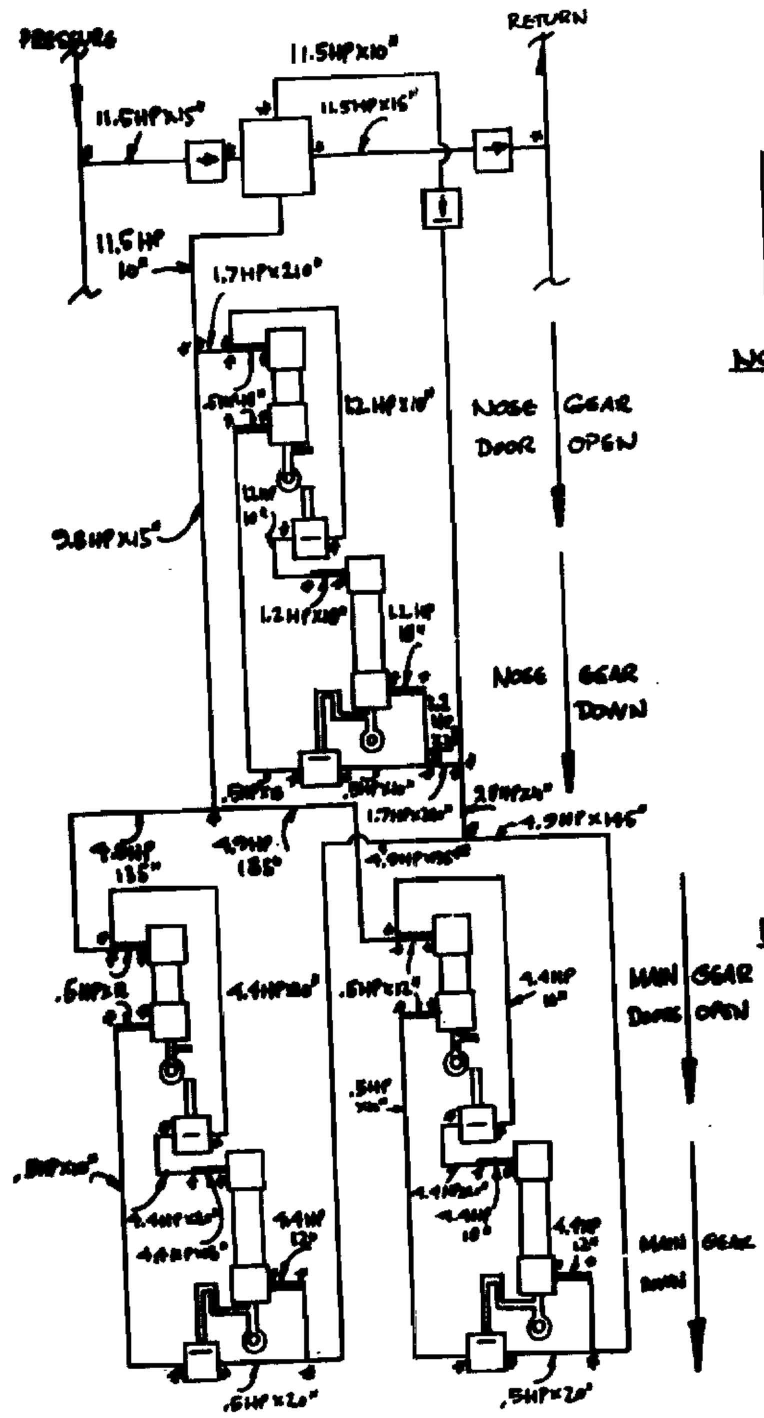
WHEEL BRAKES

MEDIUM MAIN HYDRAULIC SYSTEM
FUSELAGE DIVE BRAKE UTILITY CIRCUIT



CONTROL VALVE SOLEN. OPER.	4.1HP
CHECK VALVE (2)	4.1HP
FUSE (2)	4.1 HP
RESTRICTOR (2)	2.05HP
CYLINDER (2)	81500 " #

MEDIUM MAIN HYDRAULIC SYSTEM
LANDING GEAR UTILITY CIRCUIT



CONTROL VALVE	11.5HP
MANUALLY OPER	
CHECK VALVE(2)	11.5HP
FUSE	11.5HP

NOSE GEAR

CYLINDER DOOR	6600 LB
SEQUENCE VALVE (DOOR OPER.)	1.2HP
SEQUENCE VALVE (GEAR OPER.)	.5HP
CYLINDER	15800 LB

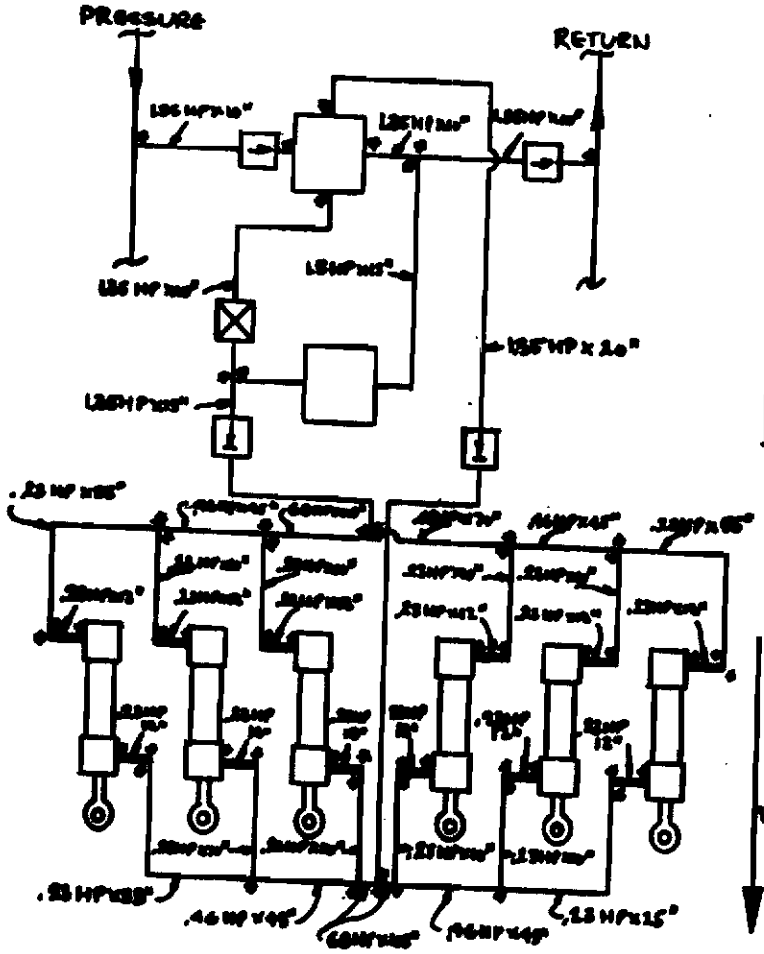
MAIN GEAR

CYLINDERS (2)	6600 LB
SEQUENCE VALVE(2) (DOOR OPER.)	4.4HP
SEQUENCE VALVE (2) (GEAR OPER.)	.5 HP
CYLINDER (2)	57000 LB

MEDIUM MAIN HYDRAULIC SYSTEM

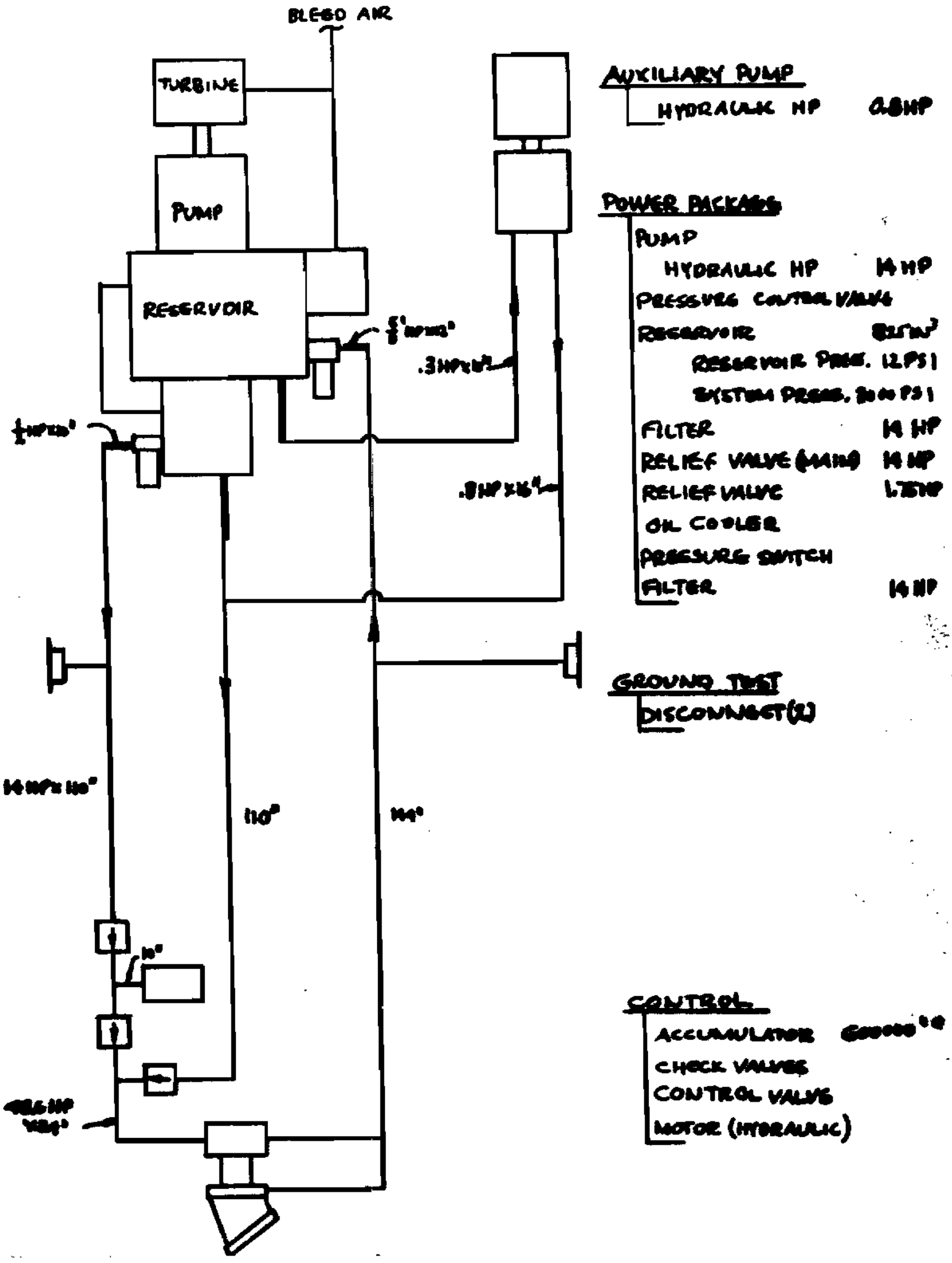
LEADING EDGE WING FLAP CIRCUIT (TRAILING EDGE IS SIMILAR)

FIGURE E-47
55



CONTROL VALVE	
SOLEN. OPER 4 WAY	1.35 HP
CHECK VALVE (2)	1.35 HP
RESERVOIR	1.35 HP
RELIEF VALVE	1.5 HP
FUSES (2)	1.35 HP
CYLINDERS (6)	5950 " "

POWER PACK HYDRAULIC SYSTEM



2.D HEAT REJECTION

Heat is generated within an aircraft hydraulic system by mechanical friction, line and valve pressure drop, and internal leakage.

Horsepower which is converted to heat equals pump friction horsepower loss, plus line and valve pressure drop during system actuation expressed as horsepower, plus pump internal leakage horsepower, plus valve internal leakage horsepower, plus some additional leakage horsepower for the remaining components.

On the basis of the discussion in Appendix E on the effect of higher pressure in pump design, the mechanical friction of the moving parts within the pump would not be seriously effected by various system pressures. The pump mechanical friction constant is directly proportional to pump horsepower and independent of pressure. Consequently, the pump mechanical friction horsepower loss can be stated:

$$HP_{PF} = \text{Constant} \times HP \tag{2D-1}$$

Two popular makes of aircraft hydraulic pumps of 3 GPM capacity (nominal rating) were examined to find the horsepower consumed when driven at 3750 RPM and fully feathered at 3000 PSI. It was observed that the average horsepower lost to overcome mechanical friction was 0.47 horsepower, and the horsepower lost through internal leakage was .91 horsepower. The rated horsepower was seen to be 12.6 horsepower. (In the full flow conditions at 2600 PSI which condition has been used in this study for obtaining a horsepower rating for a pump.) So the constant in Equation 2D-1 is found to be:

$$\text{Constant} = \frac{\text{Horsepower lost by friction}}{\text{Total driving horsepower}}$$

$$= \frac{0.47}{12.6} = .0373$$

So then Equation 2D-1 appears as:

$$HP_{PF} = .0373\ HP \tag{2D-2}$$

Where:

HP_{PF} = Horsepower lost through mechanical pump friction

HP = Rated horsepower described above

Using the medium main hydraulic system, Figure 2-4, as a basis of the heat analysis, with a normal rated pump horsepower of 12.6;

$$HP_{PF} = .0373(12.6) = .47$$

For the medium main system

In order to arrive at an equation for internal leakage past spools in valves or pistons in pumps, test data from cylindrical lap fitted parts was utilized. From this data:

$$Q_L = \text{Constant } d\, c^3\, P \qquad (2D\text{-}3)$$

Where:

Q_L = Leakage flow quantity

d = Spool or piston diameter

c = Radial clearance space distance

P = System pressure

Equation 2D-3 is based on laminar flow through the clearance space. With the assumption that the mechanical design of a pump would remain relatively constant throughout the pressure range, the piston area would be inversely proportional to pressure and directly proportional to rated horsepower, so:

$$A = \frac{\pi}{4} d^2 = \frac{K^1}{P} \times HP$$

$$d = \frac{K}{P^{.5}} \times (HP)^{.5}$$

The diametral clearance can be varied according to the cost of manufacturing operations and other factors. However, for given materials and at a constant cost, the diametral clearance is independent of system pressure (P) and piston diameter (d) within the range of primary concern.

$$Q_L = \text{Constant} \times \frac{1}{P^{.5}} (HP)^{.5} \times P$$

$$HP_{PL} = \text{Constant} \times P^{1.5} (HP)^{.5}$$

To evaluate the above constant:

$$.91 = \text{Constant} \times (3000)^{1.5} (12.6)^{.5}$$

$$\text{Constant} = 1.62 \times 10^{-6}$$

So:

$$HP_{PL} = 1.62 \times 10^{-6} P^{1.5} (HP)^{.5} \qquad (2D\text{-}4)$$

where HP_{PL} = Internal pump leakage

Flow through the valve port can be considered to be similar to flow through an orifice and as a result an equation of the type expressing orifice flow can be used or:

$$\Delta P = \text{Constant} \times \rho \frac{Q^2}{A^2} \qquad (2D\text{-}5)$$

Where:

ΔP = Pressure drop through the valve

ρ = Fluid density

Q = Flow through valve

A = Cross sectional area of spool

For a given horsepower capacity of the valve port:

$$P \times Q = \frac{HP_v}{1714}$$

Where P = System pressure

so:

$$Q = \frac{HP_v}{1714\ P}$$

Assume $\Delta P = KP$

Whereupon, Equation 2D-5 becomes:

$$KP = \text{Constant} \times \rho \frac{\left(\frac{HP_v}{1714\ P}\right)^2}{A^2}$$

Solving for area squared:

$$A^2 = \text{Overall constant} \times \frac{\overline{HP_v}^2}{P^3} = \frac{\pi^2}{16} d^4$$

where: d = Diameter

And so:

$$d^4 = \text{Constant}' \times P^{-3} (HP_v)^2$$

$$d = \text{Constant } P^{-.75} (HP_v)^{.5} \qquad (2D\text{-}6)$$

Combining Equations 2D-6 and 2D-3

$$Q_L = \text{Constant} \times P^{-.75} \times P \times (HP_V)^{.5}$$

$$Q_L = \text{Constant} \times P^{.25} \times (HP_V)^{.5}$$

Horsepower is defined as:

$$HP = K' P Q$$

So leakage expressed in terms of horsepower becomes:

$$HP_{VL} = K' P Q_L = K \times P \times P^{.25} \times (HP_V)^{.5}$$

$$HP_{VL} = K P^{1.25} (HP_V)^{.5}$$

For internal valve leakage, it was found from six actual representative control valves that the average valve leakage amounted to .02 times full flow for 3000 PSI with the average full flow of 3.38 GPM. The leakage horsepower is:

$$HP_{VL} = \frac{(.020)(3.38)(3000)}{1714} = .131 \text{ HP}$$

The valve horsepower is:

$$HP_V = \frac{(3.38)(3000)}{1714} = 5.91 \text{ HP}$$

And so for K for valve leakage:

$$K = \frac{HP_{VL}}{(HP_V)^{.5} \times P^{1.25}} = \frac{.131}{(5.91)^{.5}(3000)^{1.25}} = 2.43 \times 10^{-6}$$

$$HP_{VL} = 2.43 \times 10^{-6} P^{1.25} (HP_V)^{.5} \qquad (2D\text{-}7)$$

If 50 percent of valve leakage is allowed for other miscellaneous leakages, the total is

$$HP_{VL} = 1.5 (2.43 \times 10^{-6}) P^{1.25} (HP_V)^{.5} + 1.62 \times 10^{-6} P^{1.5} (HP_P)^{.5}$$

Finally:

$$HP_L = 3.64 \times 10^{-6} P^{1.25} (HP_V)^{.5} + 1.62 \times 10^{-6} P^{1.5} (HP_P)^{.5} \qquad (2D\text{-}8)$$

Since the heat rejection depends on the individual proportion of the components within the system as indicated by the existance of variables HP_V and HP_P in Equation 2D-8, it was decided to take a typical system as a basis of the heat rejection analysis. The fighter bomber medium size main system Figure 2-5 was selected for this purpose.

The total HP_P = 13.1 + 3.7 = 16.8 HP

Total HP_V = 2(4.3) + 13.2 + 3.5 = 25.3 HP

Therefore, Equation 2D-8 becomes:

$$HP_L = 1.83 \times 10^{-5} P^{1.25} + .664 \times 10^{-5} P^{1.5} \qquad (2D\text{-}9)$$

For the medium size main system

The line and valve losses during actuation were obtained by analysis of the fighter bomber medium main system Figure 2-5. As in Appendix A, the system efficiency was assumed to be held constant at each pressure level and thus line pressure drop was assumed to be 20 percent of system pressure:

line pressure drop = .2 P

By definition of rated line HP_L:

$$\text{line flow} = \frac{HP_L}{P} \times 1714$$

Therefore, for line loss horsepower, HP_{LL}:

$$HP_{LL} = \underline{\text{Line drop} \times \text{line flow}}/1714$$

$$HP_{LL} = .2\,P \times \frac{HP_L \times 1714}{P \times 1714} = .2\,HP_L$$

Thus the horsepower lost in lines is independent of system pressure. However, its magnitude will be determined to complete the total heat analysis. Continuous losses occur in the flight control circuits of this system where the required horsepower is:

For longitudinal control	13.2 HP
For lateral control	+ 4.3
	17.5 HP

Assuming continuous rough air utilization of flight control as being equal to 15 percent of maximum horsepower output:

.15 x 17.5 HP = 2.625 HP

For average conditions, the line loss can be set at 20 percent, and valve loss at 50 percent, and so:

(.20 + .50) x 2.625 = 1.8375 HP

and this is the horsepower converted to heat by the flight control section of the main system.

The work of the various utility functions are listed as:

Wing Dive Brake	173,500 in.lb.
Fuselage Dive Brake	109,000
Wing Trailing Edge Flaps	36,200
Wing Leading Edge Flaps	35,700
Landing Gear	151,000
Bomb Door	+ 322,000
	827,400 in.lb.

827,400 in.lb. equals 0.418 horsepower assuming this work is completed in five minutes. Assuming 30 percent valve and line loss, 0.125 horsepower is converted to heat by the utility functions.

Summarizing th heat rejection from pump friction, from pump and valve leakage, and from line and valve loss during actuation

$$HP_H = .47 + 1.83 \times 10^{-5} P^{1.25} + .664 \times 10^{-5} P^{1.5} + .125 + 1.8375$$

$$HP_H = 2.4325 + 1.83 \times 10^{-5} P^{1.25} + .664\ 10^{-5} P^{1.5} \qquad (2D\text{-}10)$$

This equation is plotted in Figure 3-5.

The surface area of systems at higher pressures might be expected to reduce as a function indicated by the tubing outside diameter versus system pressure curve, Figure A7. The relationship between these variables can be stated as:

$$D = \text{Constant} \times P^{-.4}$$

Since the surface area is proportional to diameter, it can be further stated:

$$A_S = \text{Constant} \times P^{-.4} \qquad (2D\text{-}11)$$

If it is defined that Δt be equal to the temperature increment between the stabilized fluid temperature above compartment temperature, it can be assumed that the coefficient of heat transfer is constant within the range of variation of this study. So:

$$\Delta t = \text{Constant} \times \frac{HP_H}{A_S}$$

Or:

$$\Delta t = \text{Constant}\ (.595 + 1.83 \times 10^{-5} P^{1.25} + .664 \times 10^{-5} P^{1.5}) P^{.4}$$

Finally:

$$\Delta t = \text{Constant}\ (2.4325\ p^{.4} + 1.83 \times 10^{-5}\ p^{1.65} + .664 \times 10^{-5}\ p^{1.9}) \tag{2D-12}$$

Without special heat transfer previsions, fluid temperature might be expected to stabilize at 65° F above the compartment air temperatures for a 3000 psi system. Introducing this information into Equation 2D-12, the constant becomes:

$$\text{Constant} = .676$$

Accordingly, Equation 2D-12 is:

$$\Delta t = 1.647\ p^{.4} + 1.237(10^{-5})\ p^{1.65} + 4.49\ (10^{-6})\ p^{1.9}$$

This equation appears plotted in figure 3-5.

TABLE 2-[illegible]

$$HP_H = 2.4325 + 1.83\,(10^{-5})\,P^{1.25} + 6.64\,(10^{-6})P^{1.5}$$

P	HP_H
1000	2.7455
2000	3.2675
3000	3.9095
4000	4.7185
5000	5.5015
6000	6.4225
7000	7.5295
8000	8.5325
9000	9.6155
10000	10.9025

TABLE 2-8

$$\Delta t = 1.647P^{.4} + 1.579\,(10^{-5})\,P^{1.65} + 5.73\,(10^{-6})\,P^{1.9}$$

P	Δ t
1000	29.40
2000	46.20
3000	65.00
4000	86.84
5000	112.18
6000	141.00
7000	176.20
8000	212.10
9000	249.30
10000	293.50

2-E SELECTION OF MATERIALS

A study of the various materials available for hydraulic systems as well as materials under development were considered for temperatures up to 1200°F. Materials included are aluminum alloy; heat treated steels; corrosion resistant steel and titanium.

In selecting materials used at elevated temperatures, consideration must be given to their coefficients of thermal expansion. Dissimilar metals should be avoided, especially when large temperature differentials are encountered.

Ultimate tensile strength versus temperature and strength-weight ratio versus temperature curves are plotted in figure 2-6,7. Due to insufficient substantiating data, considerable data presented here-in is estimated and hence, should be used as a guide and with caution.

With reference to figure 2-6, the progress reports issued under this contract, various publications, industrial and governmental, and The Glenn L. Martin Company Engineering personnel, the following conclusions are obtained.

> The aluminum alloys lose strength rapidly at temperatures above 350°F. 75 ST is not recommended because of poor fatigue properties and rapid reduction in strength with increased temperatures. Within the above temperature range, recommended steels are the heat treated steels and 18-8(304) corrosion resistant steel. The latter steel is selected due to its availability, relative low cost, lack of critical elements required in its composition, and high temperature resistance. Titanium alloys are not recommended for immediate use because of high costs, poor availability, lends itself poorly to production techniques, poor fatigue characteristics, rapid strength reduction with increased temperatures; and availability of suitable substitutes

At increased temperatures, only the heat treated steels and corrosion resistant steels are recommended. Titanium is again not recommended because of reasons noted in the above paragraph. Heat treated steels may be used up to temperatures where the alloy loses its heat treatment: 700°F for 180,000 H.T., 800°F for 150,000 H.T., and 950°F for 125,000 H.T. (SAE 4130 Steel). Above 950°F, 18-8(304) corrosion resistant steel is recommended.

Titanium alloys may find use in a temperature range between 350 and 950°F if suitable alloys are developed with subsequent reduction in cost, increase in physical properties and better adherence to production techniques.

Based on the rapid progress made in the past, there is an excellent possibility that suitable alloys will be developed and will find application within aircraft hydraulic systems.

An analysis has been made to determine the most applicable places within the system for application of titanium to determine the saving in weight at various system pressures, and thus determine any effect that substitute materials may have on the optimum pressure (See figure 2-7)

The materials selected as a basis, for the analysis of elements, are itemized below. In cases where two different materials are indicated for a part, the material designated by "a" was used in the basic analysis, and the material designated by "b" was used for more limited calculations in the region of major concern.

1. Tubing

   a. MIL-T-6845, stainless steel, F_{tu} = 105,000 psi was used for all tubing except supply lines. A practical limit of 3/16 inch outside diameter minimum and .020 inch wall thickness was established. The limit, however, had little influence where the optimum pressure is indicated.

   b. RC 130 B, titanium, F_{tu} = 150,000 psi was used as an alternate material for tubing. Assuming a hydraulic system working temperature of 400°F, the ultimate tensile stress at that temperature dropped to 90,000 psi which is very close to the tensile stress of one eighth hard 18-8 stainless steel (See figure 2-6, 88,000 psi at 400°F for 18-8 steel) This substitution means considerably greater difference in fabrication of the hardness condition at room temperature.

2. Supply Tubing

   a. 52 SO aluminum F_{tu} = 28,000 psi

3. Cylinder Piston Rod and Piston

   a. 4130 steel F_{tu} = 125,000 psi. Piston and rod assemblies are used as an integral part and are rarely heat treated above this value because the design is governed by cylinder column stiffness which is dependent only on Young's modulus. (E = 29,000,000 psi)

   b. RC 130 B, titanium, F_{tu} = 150,000 psi. In substituting titanium, the piston rod wall thickness was increased inversely with the ratio of Young's modulus.

4. Cylinder Rod End

   a. Steel, heat treated, F_{tu} = 180,000 psi

   b. TI-150A, titanium, F_{tu} = 150,000 psi at room temperature

5. Cylinder Heads and Barrel

a. 24S-T4, aluminum alloy, F_{tu} = 50,000 psi (transverse)

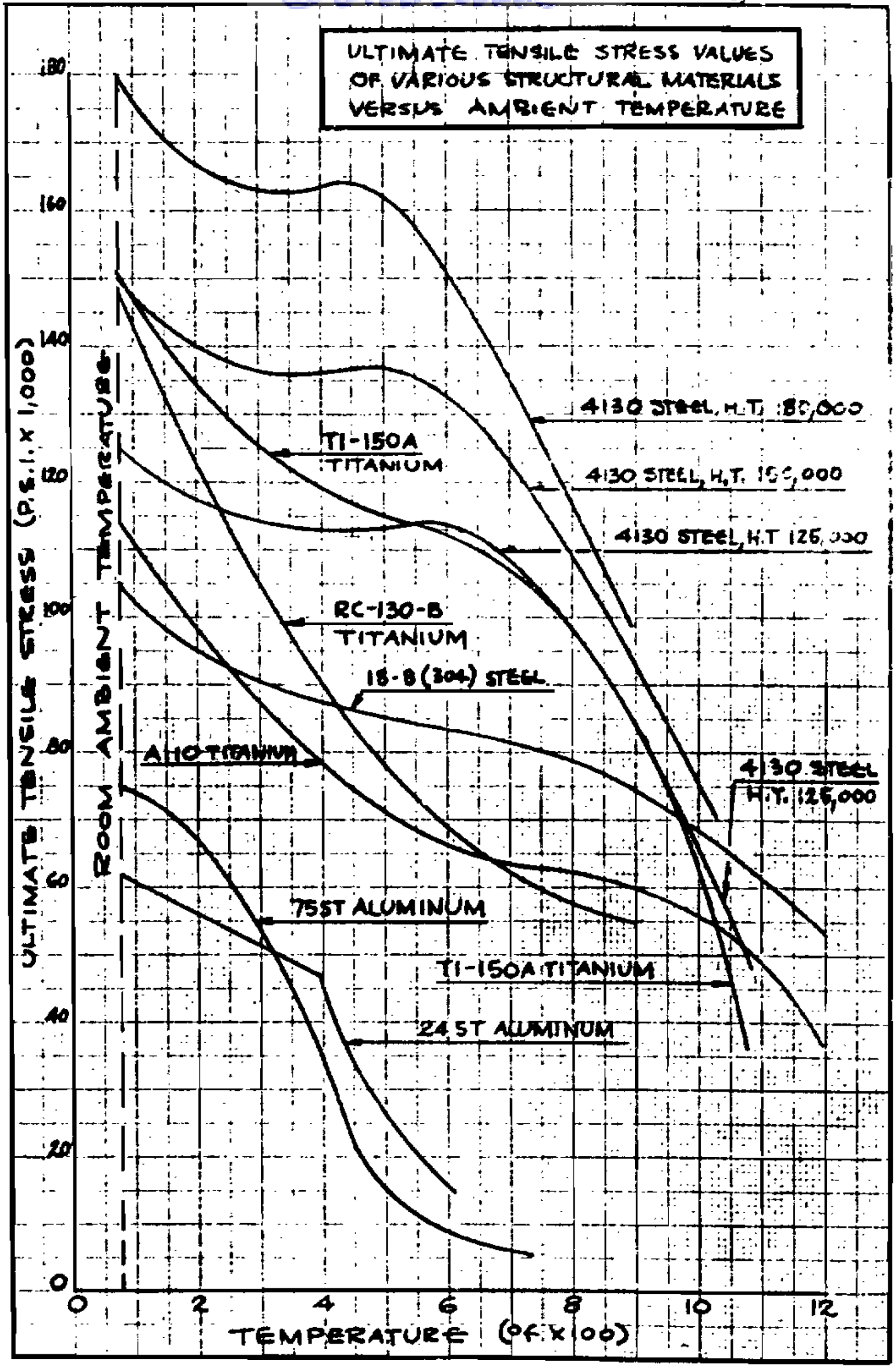
6. Fittings

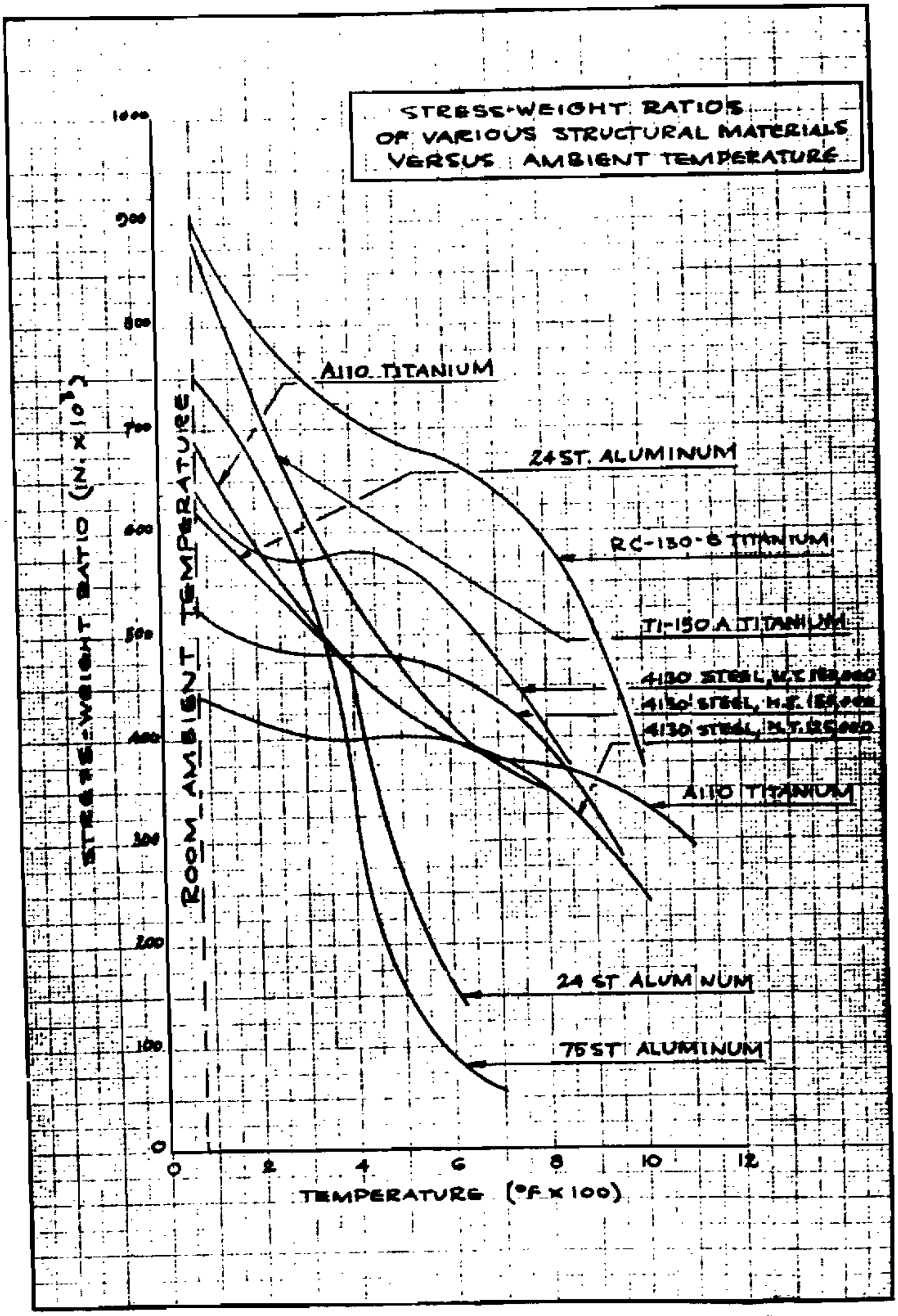
a. MIL-S-6758, condition F, 4130 steel, heat treated to 144,000 psi at a safety factor of 4:1, was selected for all fittings in accord with MIL-F-5509A (The numerical values of 180,000 psi with a safety factor of 5:1, which result in the same stress of 36,000 psi as stated above, was fed into the equation in the fitting analysis.)

b. RC 130 B, titanium, F_{tu} = 150,000 psi at room temperature was analyzed as a substitute material using the properties at 400°F which are F_{tu} = 119,000 psi.

7. Valves

Since the equations for valves were derived from expressions previously obtained for elements of fittings and cylinder barrels, the valve materials were considered to be a combination of 24S-T4 aluminum alloy and some 4130 steel.


ULTIMATE TENSILE STRESS VALUES OF VARIOUS STRUCTURAL MATERIALS VERSUS AMBIENT TEMPERATURE
ULTIMATE TENSILE STRESS (P.S.I. x 1,000)
ROOM AMBIENT TEMPERATURE
TEMPERATURE (°F x 100)
4130 STEEL, H.T.
4130 STEEL, H.T.
4130 STEEL, H.T 125,
TI-150A TITANIUM
RC-130-B TITANIUM
18-8 (304) STEEL
A 110 TITANIUM
4130 STEEL H.T. 125,000
75ST ALUMINUM
TI-150A TITANIUM
24 ST ALUMINUM
0
20
40
60
80
100
120
140
160
180
0
2
4
6
8
10
12


STRESS-WEIGHT RATIOS
OF VARIOUS STRUCTURAL MATERIALS
VERSUS AMBIENT TEMPERATURE
A110 TITANIUM
24ST. ALUMINUM
RC-130-B TITANIUM
TI-150 A TITANIUM
4130 STEEL
4130 STEEL
4130 STEEL
A110 TITANIUM
24 ST ALUMINUM
75 ST ALUMINUM
ROOM AMBIENT TEMPERATURE
STRESS-WEIGHT RATIO (IN. X 10³)
1000
900
800
700
600
500
400
300
200
100
0
0
2
4
6
8
10
12
TEMPERATURE (°F X 100)

2-F POTENTIAL APPLICATION OF TITANIUM

Steel parts of cylinders were examined to determine the advisability of using titanium.

Unibal Bearing

Titanium does not have the potential for presenting the surface equivalent to case hardened steel, ball bearing steel, or highly heat treated stainless steel and could not be considered a suitable substitute for the elements of the unibal bearing.

Cylinder Rod Ends

Titanium could be used for cylinder rod ends assuming a fatigue property comparable to 180,000 psi. Heat treat steel can be ultimately obtained. The weight saving for the use of titanium then would be equivalent to the ratio of strength density ratios. (Strength is taken as ultimate tensile stress).

The strength density ratio for steel is:

$$SDR_{ST} = \frac{180,000}{.283} = 636,000$$

The strength density ratio for titanium is:

$$SDR_{T} = \frac{119,000}{.167} = 712,000$$

The ratio of strength density ratios is:

$$\frac{SDR_T - SDR_{ST}}{SDR_{ST}} = \frac{712,000 - 636,000}{636,000}$$

OR: 11.95% weight reduction

Piston

The piston is usually integral with the rod for the most economical design of simple aircraft linear actuators and thus would be made out of the same material. The piston rod, which is the most critical element of the step column must be equivalent in stiffness if made of titanium. Thus the wall thickness must be increased by the ratio of Young's modulus of the two materials. Therefore, if both the piston and the rod were made of titanium with a suitable correction in rod wall thickness, the weight change for the typical cylinder (20,000 Lb In) would be as follows:

ITEM	STEEL WEIGHT	TITANIUM WEIGHT	DIFFERENCE
Piston	.0363 Lb.	$.0363 \left(\frac{.167}{.283}\right) = .0214$ Lb	-.0149 Lb.
Rod	.4675	$.4675 \left(\frac{.167}{.283}\right) \left(\frac{29}{17.9}\right) = .447$	-.0205
Total			-.0354 Lb.

This represents 7.03% weight reduction.

Summary of Cylinder Weight Savings

The remainder of the parts made of aluminum alloy, which in general consists of a relatively high percentage of partially stressed bulk, will not show any weight savings by using titanium because of its higher density (approximately 67% increase).

The following table summarizes the effect of the substitutions of titanium in place of steel for pistons, rods, and cylinder rod ends.

DESCRIPTION	3000 PSI		4000 PSI		5000 PSI	
	Steel	Titanium	Steel	Titanium	Steel	Titanium
Cylinder Rod End	.0207	.0184	.0274	.0241	.0376	.0330
Piston And Rod	.5038	.4680	.3945	.3670	.2884	.2680
TOTAL	.5245	.4864	.4218	.3911	.3260	.3010

The remaining aluminum parts weight 0.9204, 0.9099, and 0.9841 for 3000, 4000, and 5000 psi, respectively.

Accordingly the total cylinder weight can be shown as:

DESCRIPTION	3000 PSI		4000 PSI		5000 PSI	
	Steel	Titanium	Steel	Titanium	Steel	Titanium
Cylinder	1.445	1.407	1.332	1.301	1.310	1.285
Percent cylinder weight saved by use of titanium	2.6		2.2		1.9	

Fittings

Stressed material within fittings will be reduced in proportion to the strength weight ratios mentioned in the cylinder section.

Stressed material is shown to include that required for containing hydraulic pressure and that required to withstand torqueing preload (See Appendix A).

The remaining material, bulk, will be reduced in proportion to density ratio of steel and titanium.

The following table illustrates the effect on fitting weight when titanium is introduced in place of steel, using the typical value of 50 horsepower for a 1/2 inch fitting.

DESCRIPTION	3000 PSI		4000 PSI		5000 PSI	
	Steel	Titanium	Steel	Titanium	Steel	Titanium
Stressed Material	.00300	.00264	.00328	.00289	.00382	.00337
Bulk	.00244	.00144	.00244	.00144	.00244	.00144
Total	.00544	.00408	.00572	.00433	.00626	.00481

The weight saving thus is 25%, 24.4%, and 24.3% for 3000, 4000, and 5000, respectively.

Accumulaters

Titanium would appear to be profitable for use in accumulator barrels since this is the one highly stressed element without considerable partially stressed bulk. In the previous analysis, steel heat treated to 180,000 psi with a safety factor of 4.3 was used. Assuming the use of titanium (number RC 130 B) with a ultimate tensile stress of 150,000 psi at room temperature, the barrel would be reduced in weight by the change in strength density ratio values at 400°F or 11.95%.

Tubing

Titanium substituted for steel in fabrication of hydraulic tubing would show a weight saving propertional to a direct density ratio, and this would be so for all pressure ranges. Consequently tubing would be 41% lighter (i.e. .283 minus .167 and divided by .283)

Valves

Similar to fittings, valves are composed of stressed material and bulk, stressed material, as stated earlier, will be reduced by the ratio of strength density ratio and bulk will be reduced by direct ratio of densities.

The table below illustrates the effects of introducing titanium into valves; in this case a typical check valve, i.e. a 1/2 inch valve at 50 horsepower.

DESCRIPTION	3000 PSI		4000 PSI		5000 PSI	
	Steel	Titanium	Steel	Titanium	Steel	Titanium
Stressed Material	.3843	.3390	.4341	.3820	.4994	.4390
Bulk	.2090	.1233	.1865	.1100	.1750	.1032
Total	.5933	.4623	.6206	.4920	.6744	.5422
Percent Weight Savings	2.21		2.06		1.96	

The medium main hydraulic system weight with titanium introduced is shown in Table 2:8 and the total values are plotted in figure 3.1

The use of a lighter material such as titanium, when it is further developed, shows no significant change in optimum pressure, but it does affect considerable weight saving. In the example shown the percent of weight saved is approximately eleven percent.

TABLE 2-9

WEIGHT SUMMATION OF MEDIUM MAIN HYDRAULIC SYSTEM USING TITANIUM

ITEM	3000	4000	5000
Pumps (uneffected)	19.30	19.30	19.30
Reservoirs (uneffected)	33.40	26.70	22.70
Accumulators	8.10	8.05	8.01
Valves	31.00	33.30	36.30
Cylinders	60.80	59.10	61.70
Pressure Lines	14.55	14.42	14.62
Supply Lines	1.13	1.09	1.13
Return Lines	10.05	9.32	8.52
Hose	13.19	13.01	13.28
Total	191.52	184.29	185.56
Percent of Weight Saving	10.4	11.5	11.4

3. DETERMINATION OF OPTIMUM PRESSURE

As a general basis for determination of optimum pressure for aircraft hydraulic systems, the phases of design, manufacture, useful service life, and maintenance were examined to determine which factors must be considered. There follows a discussion of each of the following six factors where its characteristics are summarized and its degree of importance is determined.

1. Weight
2. Space
3. Cost
4. Reliability
5. Performance
6. Heat

The remaining variables are then compared, weighted, and combined to yield the final result in this section.

3.A WEIGHT

Each pound of weight in the hydraulic system requires the existance of 3 to 15 pounds of indirect weight in the airplane (on the basis of constant airplane performance). Thus, in a modern military airplane where the combination of high aerodynamic forces and reliability requirements demand hydraulic systems having average direct weights of 1% gross weight, the hydraulic systems actually are responsible for existance of 3% to 15% of the gross weight of the airplane. It is this major effect of the hydraulic system on the overall efficiency of the airplane which makes weight the most important of all factors in determining the optimum hydraulic system pressure.

The opinions stated by the leading hydraulic engineers, who were consulted during the industry survey, confirm the primary importance of the weight consideration. Pervious optimum pressure analysis such as Reference 3 and Appendix A were conducted on the basis of weight as the only primary consideration.

A large portion of this analysis has been devoted to an accurate determination of the weight break-down and total weight of each system throughout the range of pressures. Figure 3.1 presents a summary of the results, plotting total system weight versus pressure for the normal configuration of each of the six different basic systems considered.

It is observed that each curve contains a low and relatively flat region between 3000 and 6000 psi. The previous increase from 1500 psi to the present standard 3000 psi indicates a weight saving of 33.3% (on the basis of the same margins of safety and same refinement of design at each pressure).

Changing system pressure from 3000 to 4000 psi results in an overall average saving of 4.4% (of the 3000 psi system weight) showing a substantial saving in each system except the small flight control system.

The step from 4000 to 5000 psi results in no significant change; the overall average figure being a .3% weight increase which is negligible considering the probable accuracy of the data. Although a mathematical low point occurs at 4400 psi, a practical point of no return would be located at 4000 psi.

Above 5000 psi the total weight of each system increases progressively as shown by concave upward curvature on the log scale and more pronounced upward curvature on the straight scale of Figure 3.1. If extended to the right far enough the system weight curves would approach a vertical asymtote at the structural limit of the weakest material in the system where system weights would all be infinite.

A typical weight breakdown of the medium main hydraulic system is plotted against pressure in Figure 3.2 to illustrate the relative proportions and the effect of each component upon the optimum pressure. The pump and accumulator weights have no effect upon the optimum pressure. The valves and cylinders have similar characteristics and tend to produce a slightly lower optimum pressure which effect is compensated by reservoir weight. Fortunately valves, cylinders, and reservoirs were thoroughly analysed in detail, the reservoir analysis being in accordance with MIL-R-5520A. The four uppermost curves represent total system weight with no plumbing at all, with 50%, with 100%, or with 200% of the normal proportion of lines which are found in current airplanes. The same optimum pressure occurs in all cases because the total of pressure, return, and supply line weight characteristic is similar to the total system weight characteristic.

Thus, considering weight, Figure 3.2 illustrates that the optimum pressure is independent of the length of plumbing and relatively independent of the proportions of the other components. The variety of systems plotted in Figure 3.1 illustrates that the optimum pressure is independent of system horsepower throughout the range in use today or predicted for the near future; is independent of the type of system such as flight controls consisting mainly of a power circuit, utility systems consisting of more lines, valves, and actuators; package type systems, and systems containing only motor type actuators.

3.8 SPACE

The trend toward higher density aircraft for efficient aerodynamic performance at high subsonic and supersonic speeds has brought about an increasing number of space problems throughout the aircraft. This trend will undoubtedly continue and the need for compacting aircraft equipment and systems will always exist.

Space problems have occurred to various degrees in the design of aircraft hydraulic systems. The industry survey revealed that the most acute space problem occurs at the installation of hydraulic cylinders for actuation of surface controls within thin wing sections. The problem starts with the mechanism in providing sufficient moment arms to keep deflections, lost motion, and bearing loads within reason. The actuator must then be applied to this mechanism (or moment arm) closely coupled to the control surface, keeping the outside diameter within contours, keeping the length within the length of the compartment available and leaving enough space around the actuator for associated equipment such as the control valve, control and follow-up mechanism, hydraulic plumbing, and any special servo-mechanism accessories.

Cylinder space was analyzed in considerable detail taking into account the effect of column stability limitations. Figure B-16 and B-16B show the space made useless by the existance of cylinders plotted against pressure. The various work levels represent the complete range of sizes obtained from statistics in Appendix B. Although the smaller sizes are influenced by practical minimum wall thickness, the curves are reasonably consistent in shape. These curves were first normalized based on unity at 3000 psi and then averaged to produce the curve marked Average Cylinder Space on figure 3-8. To determine the influence of cylinder space with respect to other factors, the space occupied by cylinders is considered 37% as important as total system weight.

During the industry survey it was stated that a reduction in space in routing the plumbing and installation of components was also generally desirable. However, this factor was considered of second order importance with respect to total system weight and less important than cylinder space. A space analysis was conducted on each component and the results integrated into a summary for each synthetic system considered in the study. Figure 3-3 is a plot of total system space versus pressure for each system. Since these curves are reasonably consistent, each system curve was normalized based on unity at 3000 psi and then the systems were averaged to produce the curve marked Average Total System Space, see figure 3-8. The curve indicates substantial improvement in space as pressure is increased up to 4000 psi beyond which the curve is fairly flat with a theoretical low point at 8000 psi. The influence of total system space on determination of optimum pressure was considered to be 20% as important as total system weight.

Combining the 30% factor for direct cylinder space with the 20% factor for total system space (of which 35% is cylinder space) the total influence of all space considerations is 50% as important as total system weight. In reality this 50% is made up of 37% factor for the total influence of cylinders and 13% factor for the total influence of other components and plumbing.

3.C COST

On the basis of a fixed budget economy the factor of cost is of first order importance. The entire study was conducted on the basis of recurring costs being held relatively constant and conversely the cost factor does not enter into the determination of optimum pressure. For example, many ingenious design improvements are possible throughout the hydraulic system (although not necessarily advisable at this time) such as concentric laminated or wire wrapped tubing involving combinations of materials, complete packaging of miscellaneous components, or complicated actuator internal designs to delay column instability, each of which involves additional cost and complication to gain weight, space or performance at any system pressure level. Although such ideas are used in isolated cases to meet extreme requirements, the more typical designs of components and systems are usually dictated by overall economy. These designs were used in the analysis holding cost relatively constant so that the influence of various system pressures is reflected entirely in the variation of weight, space, stiffness, and heat factors.

Of course, on the short term basis, initial costs for component development, and replacement of test and service facilities are inevitable during the transition period following any basic change of system temperature or pressure. This factor is examined in section 4, Practical Significance of the Optimum Pressure, and should be considered from the standpoint of advisability of making a change.

3.D RELIABILITY

When it can be evaluated, reliability is a primary factor to be considered in making any basic decision. Throughout the study, the concept of reliability was incorporated by considering it necessary to maintain an equivalent degree of reliability in component and system designs analyzed throughout the pressure range. On this basis the factor of reliability is relatively constant in determination of optimum pressure.

However, on the short term basis, it is inevitable that there will be a temporary loss in reliability during the transition period following any basic change such as system temperature or pressure level. Reliable information such as probability factors for present day components and systems is not in existance at this time and there is no valid method of predicting this change in reliability. However the trend should be considered from the standpoint of advisability of making a change.

3.E PERFORMANCE

Because of the low inertia and compact design features offered, hydraulically powered servo-mechanisms have found extensive use in aircraft. These are used exclusively for piloted airplane surface controls and nose wheel steering, and in majority of missile surface controls. Hydraulic servos are finding increasing numbers of applications in turrets, antennas and power stabilized devices. In the majority of these cases, the angular travel or the mechanical arrangement permits the use of a simple hydraulic cylinder.

The trends in requirements for supplimentary or complete aerodynamic stabilization through actuation of surface controls and requirements for tracking at higher speeds have all been in the direction of requiring increased dynamic response of the mechanism. The potential dynamic response of a basically stable servo depends, among other things, directly upon the stiffness of the load carrying members, such as cylinder, linkage, and back up structure. Flutter stability of a surface not completely mass balanced is a function primarily of stiffness of the same members. Reduction of load deflection has been accomplished to date by one or more of the following methods:

1. Applying the actuator more directly to the surface or load (through less intermediate linkage).

2. Increasing stiffness of back-up structure and mechanism.

3. Reduction of unit loads by improving mechanical advantage of each element of the mechanism.

4. Increasing stiffness of the actuator.

Methods 3 and 4 are both affected by the level of system pressure. In the cylinder analysis it was determined that many cylinders approach the critical column curve at 3000 psi, see figure B-5 and B-6 (Appendix B). At higher system pressures these cylinders must operate at shorter strokes and higher loads as plotted on figure 3-5. As an alternative special design changes could be made such as use of through piston rods, internal area balancing surfaces, or trunnion type mounting bearings, each of which results in another form of penalty such as added weight, space, bearing deflection, or cost. It is assumed that 30% of servo actuators are so affected.

Cylinder stiffness depends directly upon volumetric displacement which varies inversely with working pressure range. Another approach is that piston deflection in percent of total stroke is increased as the internal column of fluid is compressed by higher pressures. Using MIL-0-5606 fluid, for example, with a bulk modulus of 225,000 psi at 160°F, the percent piston deflection through various ranges of working pressure is plotted in figure 3-6.

Since the mechanical and structural elements are in series their respective deflections are additive. This means that there is a limit to the overall improvement by working on any one element; ie., as soon as the weakest element is stiffened, some other element becomes critical.

In the industry one case was reported where a detail deflection analysis was performed and confirmed by test indicating that actuator deflection was 58% of back up structure deflection. Preliminary analyses at other companies indicated values of 6.7%, 7.2% and 36% yielding an average of 27%.

In order to correlate the stiffness data with other factors, the percent stiffness curve was nondimensionalized with unity at 3000 psi and the increments were reduced to 27%. The load level curve was then nondimensionalized and multiplied directly by the 27% cylinder deflection curve. The resulting curve was considered to be 10% as important as total system weight (this figure is the percent of total system weight contributed by servo cylinders).

3.F HEAT

The problem of temperature control within hydraulic systems has become increasingly important in recent years for a number of basic reasons.

Compartment temperatures may be expected to stabilize after 4 to 5 minutes of high speed flight at a temperature of 85 to 90% of the theoretical ram rise due to skin friction alone and higher temperatures will occur where internal heat is being generated. As a result of the trend toward operational airplanes in the high subsonic and supersonic speed range, a temperature barrier now exists and will continue to exist in the foreseeable future whereby airplane performance is limited by the temperature at which the basic airframe structural properties are seriously affected. Since the construction materials in the hydraulic system are there primarily to contain the internal pressure, their weight efficiency is governed by the same properties as in the case of airframe structure. This means that in many future aircraft the internal fluid temperature in general should be no higher than average compartment temperature. In other aircraft, where there is some margin with respect to the airframe temperature barrier, the hydraulic system temperature may be higher than compartment temperatures but the amount will be definitely limited.

Another trend influencing the heat problem has been the increase in system power requirements for a given size airplane. The heat generated has increased accordingly.

The rapid increase in the use of high powered hydraulic servo-mechanisms has been an important addition to the sources of heat generated because such devices are continuously in use and because servo valves generate more heat by pressure drop and by leakage than do selector valves. In the heat analysis medium main hydraulic system, 39.5% was generated at the pump and 40.5% was generated by the flight control valves. The combined influence of these trends has brought about the following action:

> The Services have undertaken and made considerable progress in the development of higher temperature fluids, seals, and components with the ultimate goal of operating at a temperature near the structural temperature barrier. When this is accomplished, the importance of temperature control will be as stated above. During the present transition period temperature control is more critical than any other factor.
>
> System designers have become increasingly more temperature concious to the point where mechanical friction and internal leakage are primary factors in pump selection, servo valves are designed and analyzed on the basis of low heat generated, unnecessary leakage is eliminated, installations provide good convection cooling, and supplimentary heat transfer units are added where necessary.

A typical average size system was analyzed in Section 2-D to determine the influence of various system pressures on the heat problem. The result plotted on figure 3-7 reveals that a considerable increase in heat generated by the hydraulic system (HP_H) occurs at higher pressures. This is caused by reduced leakage efficiency of pumps and control valves. The reduction of external surface area of lines and components causes the system stabilized temperature to rise very rapidly in the higher pressure region. This curve was normalized on the basis of unity at 3000 psi and plotted on figure 3-8.

The relative importance the heat rejection problem will vary considerably with each individual system from cases where no special steps must be taken to the case where a number of existing components are compromised and a heat transfer system must be added. For direct comparison with other factors it was estimated that the total influence of the heat problem is 10% as important as total system weight and the stabilized temperature curve normalized was taken as more representative of the effect of pressure than the heat generated curve.

TABLE 3-1

WEIGHTED SUMMARY OF FINAL ANALYSIS
NORMALIZED AT 3000 psi = 1.000

DESCRIPTION	SYSTEM PRESSURE PSI 1500	3000	4000	5000	7500	10000
Total System Weight	1.333	1.000	.956	.959	1.050	1.213
Total System Space	1.632	1.000	.862	.805	.719	.717
Cylinder Space	1.4002	1.000	.8972	.8578	.8053	.9003
Stabilized Temperature	.578	1.000	1.330	1.728	2.950	4.510
Actuator Deflection	.285	1.000	1.448	1.866	2.980	4.470
Weighted Total System Weight	1.333	1.000	.956	.959	1.050	1.213
Weighted Total System Space	1.1264	1.000	.9724	.9558	.9438	.9434
Weighted Cylinder Space	1.1201	1.000	.9692	.9573	.9596	.9700
Weighted Stabilized Temperature	.9578	1.000	1.033	1.0728	1.195	1.351
Weighted Actuator Deflection	.9285	1.000	1.0448	1.0866	1.1980	1.3470
Sum of Weighted Values	1.4658	1.000	.9754	1.0315	1.3404	1.8214
Sum of Weighted Values Less Weighted Total System Weight	1.1328	1.000	1.0194	1.0725	1.[illegible]904	1.6114

3.8 COMPARISON OF VARIABLES

In review, each of the six basic considerations, weight, space, cost, reliability, performance, and heat have been treated above. The entire analysis has been conducted on the basis of constant cost and reliability so that the influence of various system pressures is reflected entirely in the variation of weight, space, performance, and heat.

Each of these four variables was explored throughout the range encountered in aircraft hydraulic systems. The detail extent of the analyses and the resulting accuracies were in each case proportioned to the relative importance and to the weight factors ultimately to be used. An average total summary curve for each variable was first normalized with unity at 3000 psi as tabulated in the first 5 lines in Table 3-1. This data plotted in figure 3-8 shows the characteristic effect of system pressure one each variable. The vertical scale reads the ratio of the variable at any system pressure with respect to the value of the variable at 3000 psi. These are the true individual characteristics before taking any steps toward the weighted importance of each variable.

Interpreting figure 3-8, at 1500 psi the average total system weight is 1.333 times the weight at 3000 psi. At 1500 psi total system space is 1.632, average cylinder space is 1.4, actuator deflection (considered the inverse of performance) is .285, and system stabilized temperature increment is .578 times the value at 3000 psi. Instead of looking backward as a study based on the present standard of 3000 psi tends to do, if we proceed from 1500 psi as a base to 3000 psi, the system weight ratio is .75 showing a 25% saving, system space ratio is .612 showing 38.8% saving, cylinder space ratio is .713 showing 28.7% saving, deflection ratio is 3.5 or 250% increase, stabilized temperature ratio is 1.73 showing 73% increase.

Progressing from 3000 psi to 4000 psi reduces system weight by a ratio of .956 or a reduction of 4.4%, reduces system and cylinder space by 13.8% and 10.28% respectively, adds 44.8% to actuator deflection, and adds 33% to system stabilized temperature.

Progressing from 3000 psi to 5000 psi reduces system weight 4.1%, reduces system and cylinder space 19.5% and 14.22%, increases deflection 86.6%, and increases temperature increment 72.8%.

3.H INTEGRATION OF VARIABLES

In order to determine the combined effect of all variables, total system weight ratio (which is unity at 3000 psi) was chosen as the common denominator, or the basic unit for representing each other variable. The other variables were correlated with weight ratio by establishing a factor of importance in each write-up. System space was considered 20%, cylinder space 30%, performance (as defined by cylinder deflection) 10%, and stabilized temperature increment 10% as important as total system weight.

The increments from unity in each curve on figure 3-8 were reduced to the percentage quoted above and the weighted characteristic of each variable tabulated on lines 6 to 10 in Table 3-1. The weighted increments from unity were added at each pressure to determine the total weighted effect of all variables and the resulting curve was plotted on figure 3-9.

Figure 3-9, is entitled composite penalty ratio because it is composed of the combined effect of all variables, because each variable is expressed in the form of the penalty it imposes upon the aircraft, and because the numerical values are normalized with unity at 3000 psi and thus represent the ratio of penalty at a given pressure with respect to the penalty at 3000 psi.

Figure 3-9 represents the final results of the theoretical analysis. It being a penalty curve, the optimum is indicated by the low point. Therefore the theoretically optimum pressure is 4000 psi.

Proceeding from 1500 psi to 3000 psi indicates a reduction of 32% of the total penalty at 1500 psi. (46.58% of the value at 3000 psi). Progressing from 3000 psi to 4000 indicates an overall saving of 2.46%. From 3000 psi to 5000 psi indicates an added penalty of 3.15%. At 7500 psi and 10000 psi the added penalties are 29.84% and 61.14%.

A more detail examination of the changes from 3000 to 4000 psi reveals the following.

Variable Considered	% Direct Change	% Effective Change in System Weight
Total System Weight	- 4.4 x 1. =	- 4.4
Total System Space	-13.8 x .2 =	- 2.76
Cylinder Space	-10.28 x .3 =	- 3.08
Cylinder Deflection	+44.8 x .1 =	+ 4.48
Stabilized Temperature	+33.0 x .1 =	+ 3.3
Total Effective Weight		-2.46

A Breakdown of the Change from 3000 psi to 5000 psi

Variable Considered	% Direct Change				% Effective Change In System Weight
Total System Weight	- 4.1	x	1	=	- 4.1
Total System Space	-19.5	x	.2	=	- 3.9
Cylinder Space	-14.22	x	.3	=	- 4.26
Cylinder Deflection	+86.6	x	.1	=	+ 8.66
Stabilized Temperature	+72.8	x	.1	=	+ 7.28
Total Effective Weight					+ 3.68

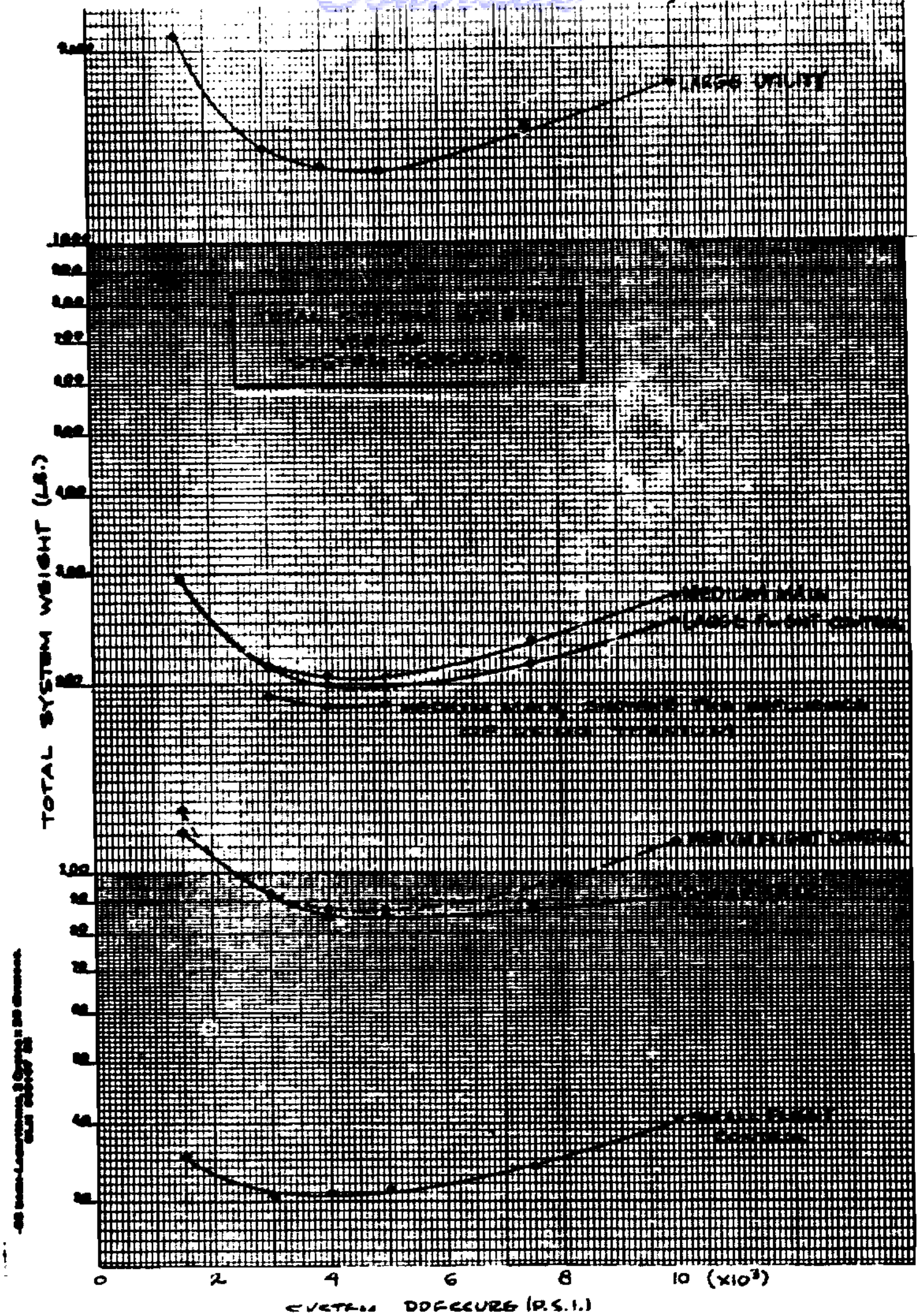
Since the principle variables have different characteristics as illustrated by figure 3-8, the choice of correlation factors has considerable effect on the exact location of the theoretical optimum pressure. However the magnitude of difference in penalty is likely to be small for special cases where one or more of the factors are changed. For instance, consider the sensitivity of the principle variable, total system weight. If weight were infinitely important, the optimum system pressure would be determined from the shape of the Average Total System Weight curve on figure 3-8 which indicates an optimum of 4400 psi but the magnitude of weight gain from 4000 psi to 4400 is less than 1/2%. On the other hand, if the factor of importance for system weight were reduced to zero, the composite penalty ratio of the remainder of the variables would be as plotted on figure 3-10. The optimum pressure has shifted to 3000 psi for this special case but the reduction in penalty from 4000 to 3000 psi is only 1.8%.

The sensitivity of a variable depends upon its relative magnitude as well as upon its characteristic. The remainder of the variables when weighted by the factor of importance are of relative small magnitude in each case. The analysis has been fully reported and plotted in such a manner that any combination of weight factors for the variable can quickly be carried through to determine the net effect. For example if cylinder deflection were considered of no importance (0%) the composite penalty ratio curve (figure 3-9) would be shifted downward and the right having a low point at 4400 psi where a saving of only 1/2% would occur with respect to the penalty at 4000 psi. On the other hand, if the 10% factor of importance previously used for deflection were doubled to 20% as important as total system weight, the modified composite penalty ratio curve would have a low point at 3400 psi where a saving of 2.7% occurs with respect to the penalty at 4000 psi. The effect of varying stabilized temperature increment and space through a similar range of zero to twice the factor of importance used in the analyses would reveal that these variables are slightly less sensitive than actuator deflection as quoted above.

Thus it is seen that the previously stated theoretical optimum pressure of 4000 psi is fairly stable as the weight factors associated with the variables are manipulated to encompass a variety of design situations. The reduction in penalty or in percent system weight, if the system pressure were adjusted for each situation, is so small (with respect to the penalty at 4000 psi) that consideration of more than one optimum pressure is not justified.

The advisability of undergoing the initial expense associated with changing from 3000 psi or adding another standard system pressure such as the theoretical optimum 4000 psi has not been considered in this section and this subject is covered in the Section 1 Conclusions and Recommendations and in Section 5 Cost.

Fig. 3.1


LARGE UTILITY
MEDIUM MAIN
LARGE FLIGHT CONTROL
MEDIUM FLIGHT CONTROL
TOTAL SYSTEM WEIGHT (LB.)
0
2
4
6
8
10 (x10³)
SYSTEM PRESSURE (P.S.I.)

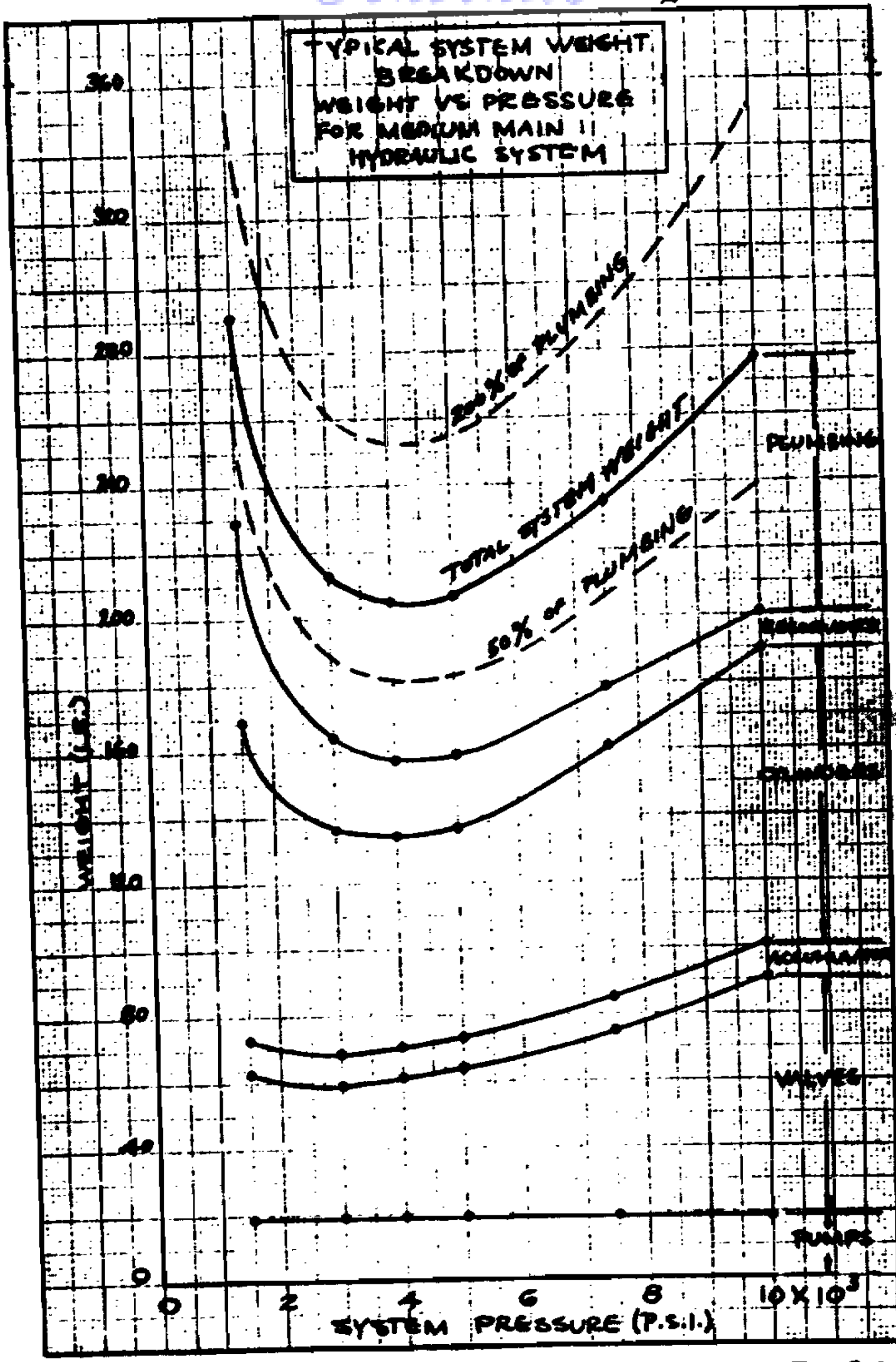

TYPICAL SYSTEM WEIGHT BREAKDOWN
WEIGHT VS PRESSURE FOR MEDIUM MAIN HYDRAULIC SYSTEM
360
320
280
240
200
160
120
80
40
0
WEIGHT (LB.)
200% of PLUMBING
TOTAL SYSTEM WEIGHT
50% of PLUMBING
PLUMBING
VALVES
PUMPS
0
2
4
6
8
10 X 10^3
SYSTEM PRESSURE (P.S.I.)

FIG 3.?

Fig. 3.3

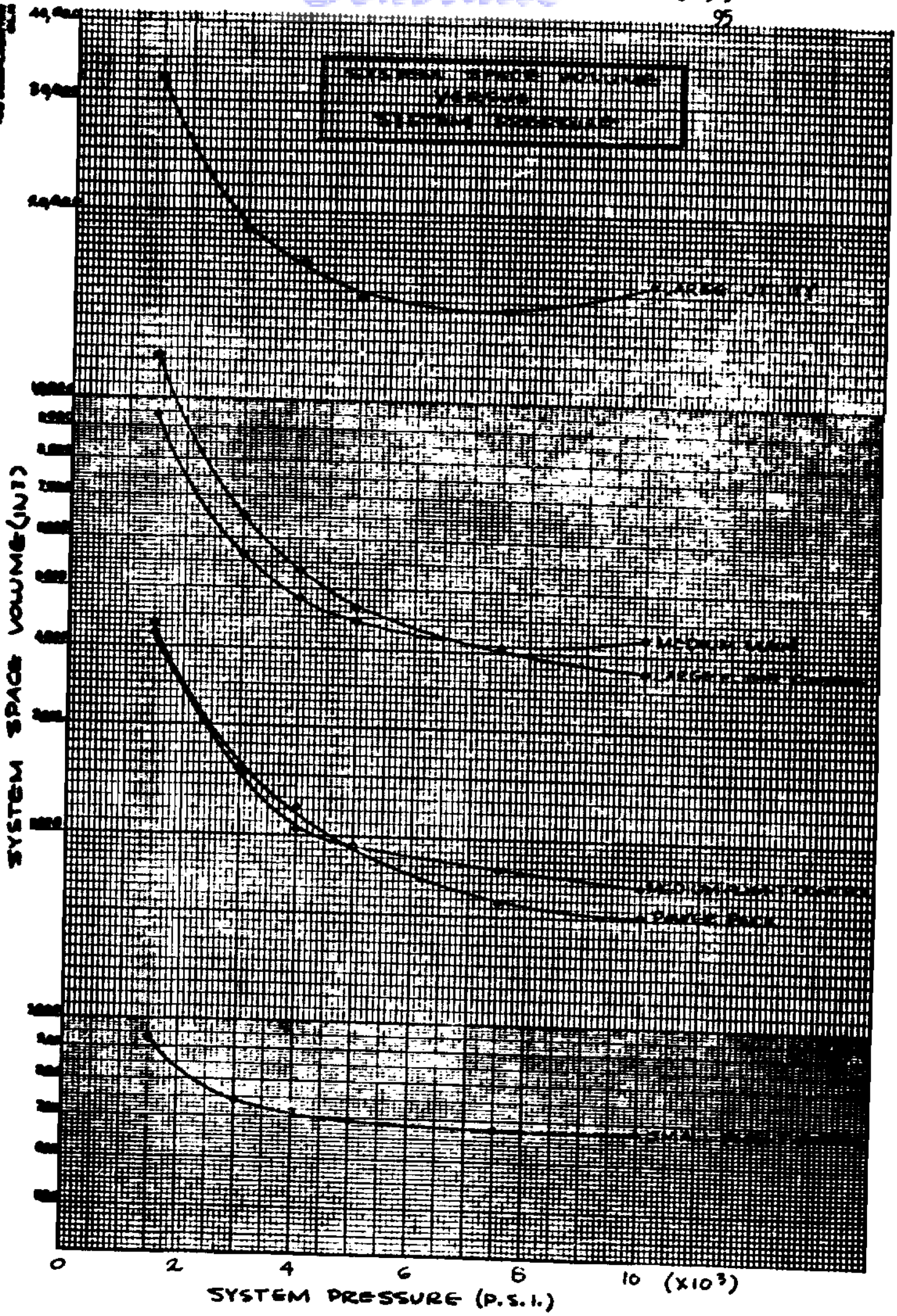
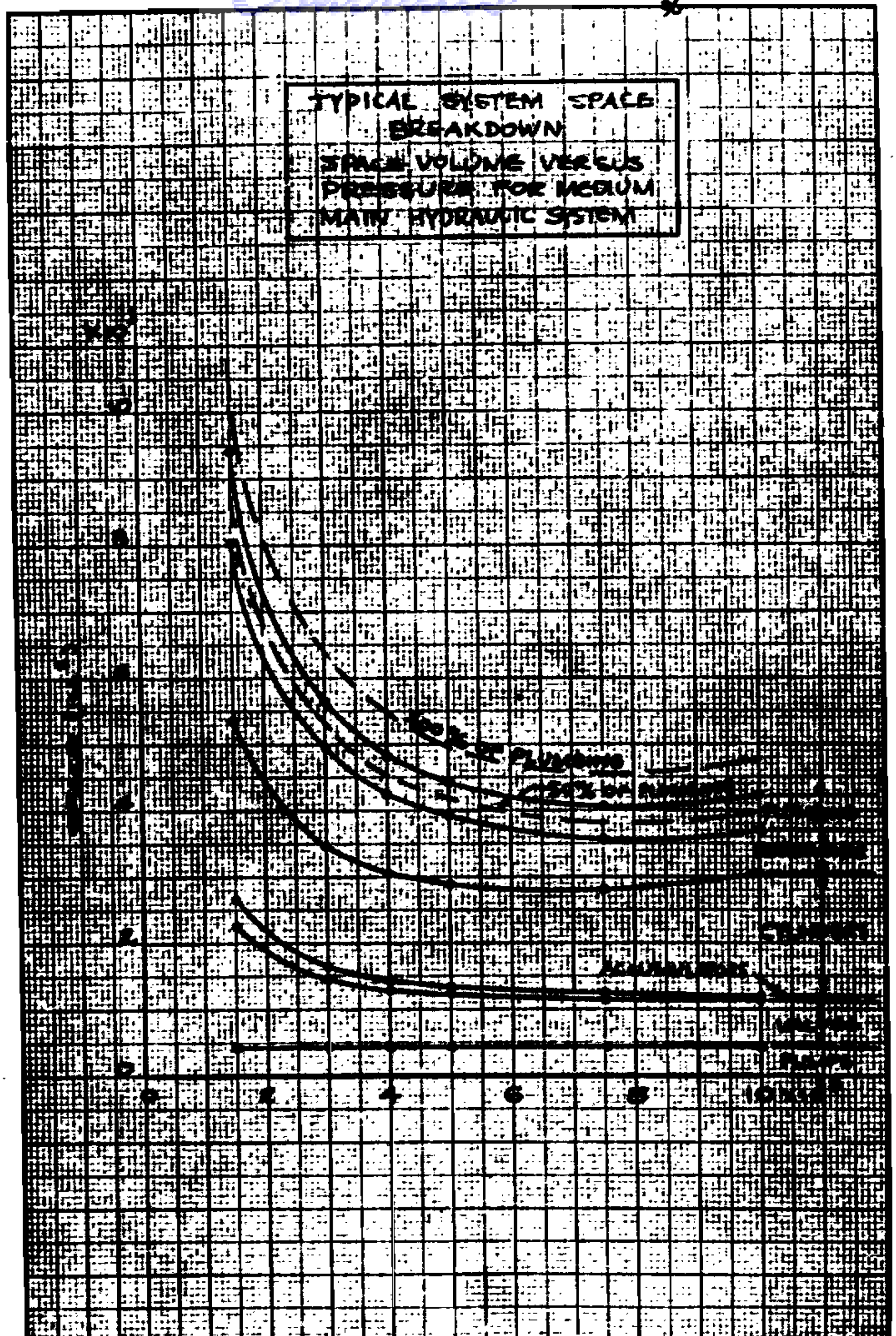
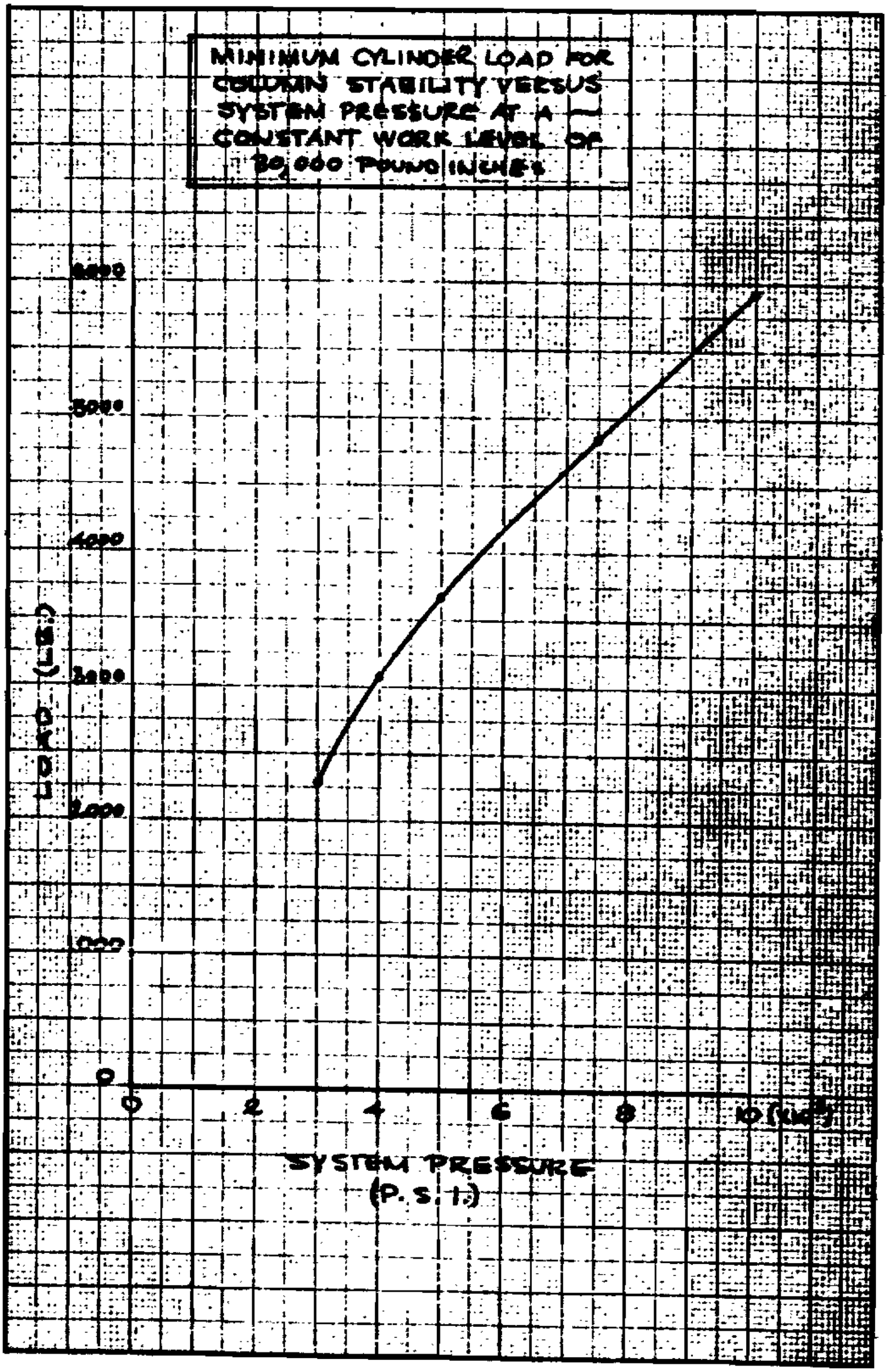
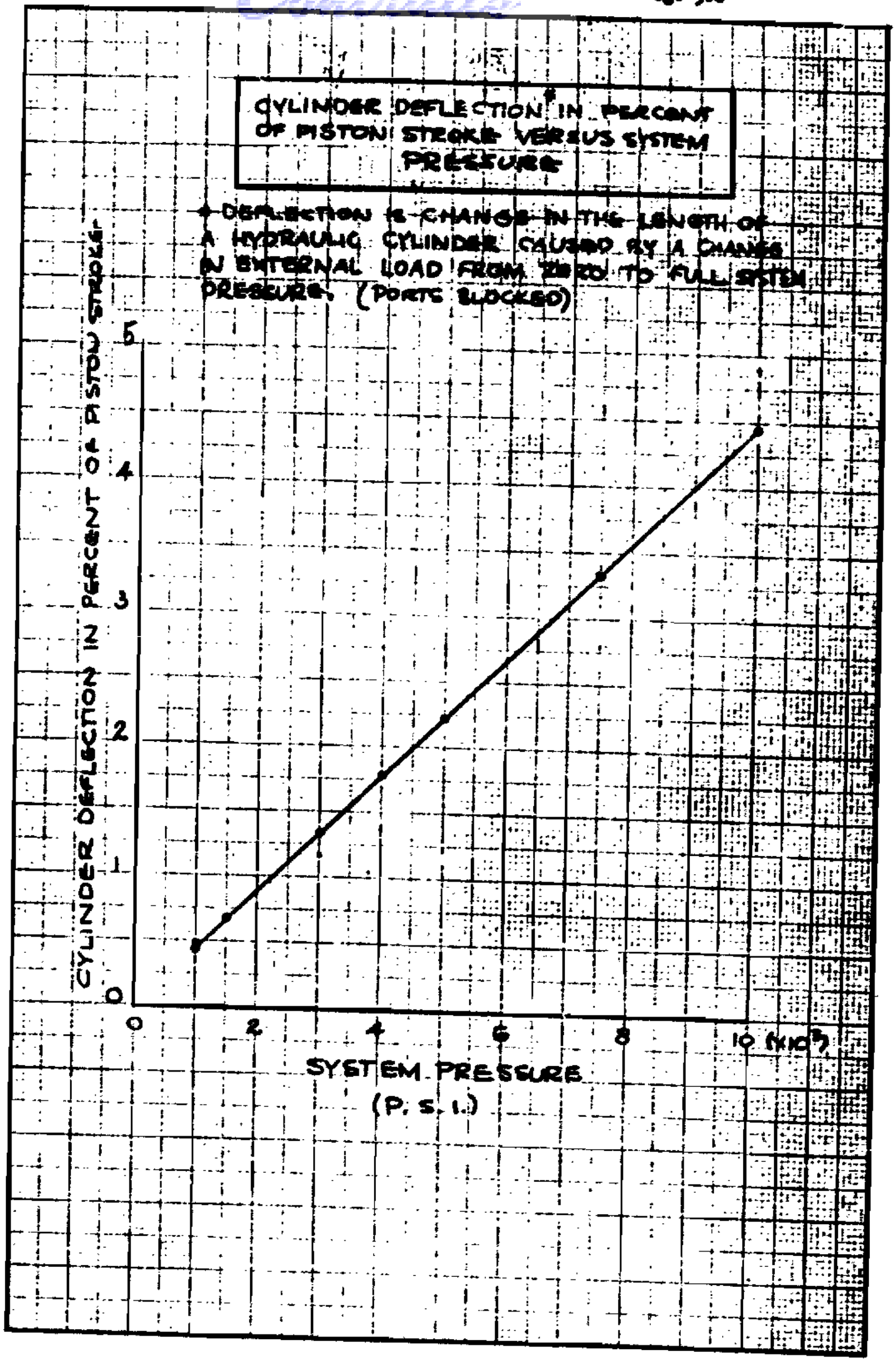
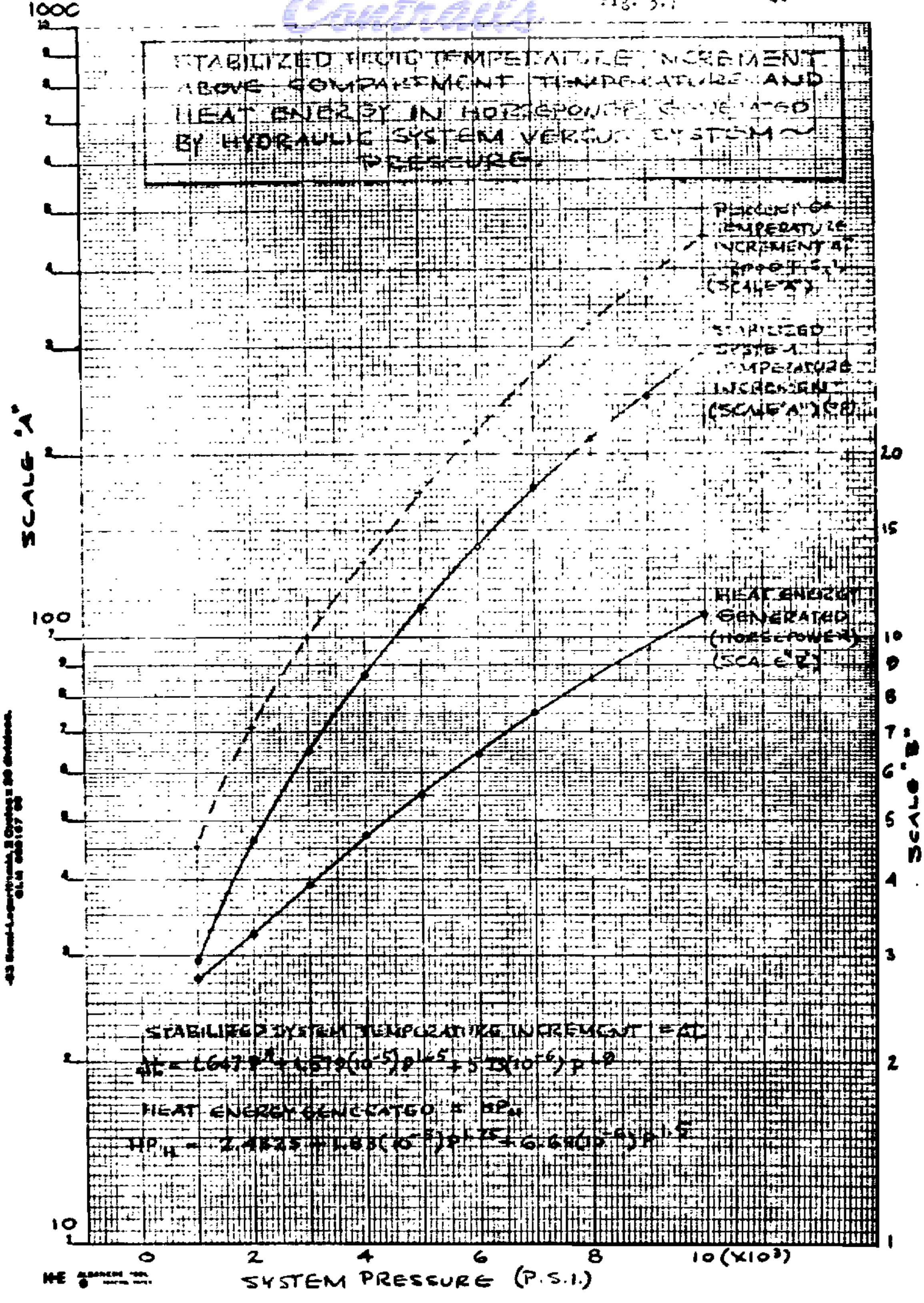


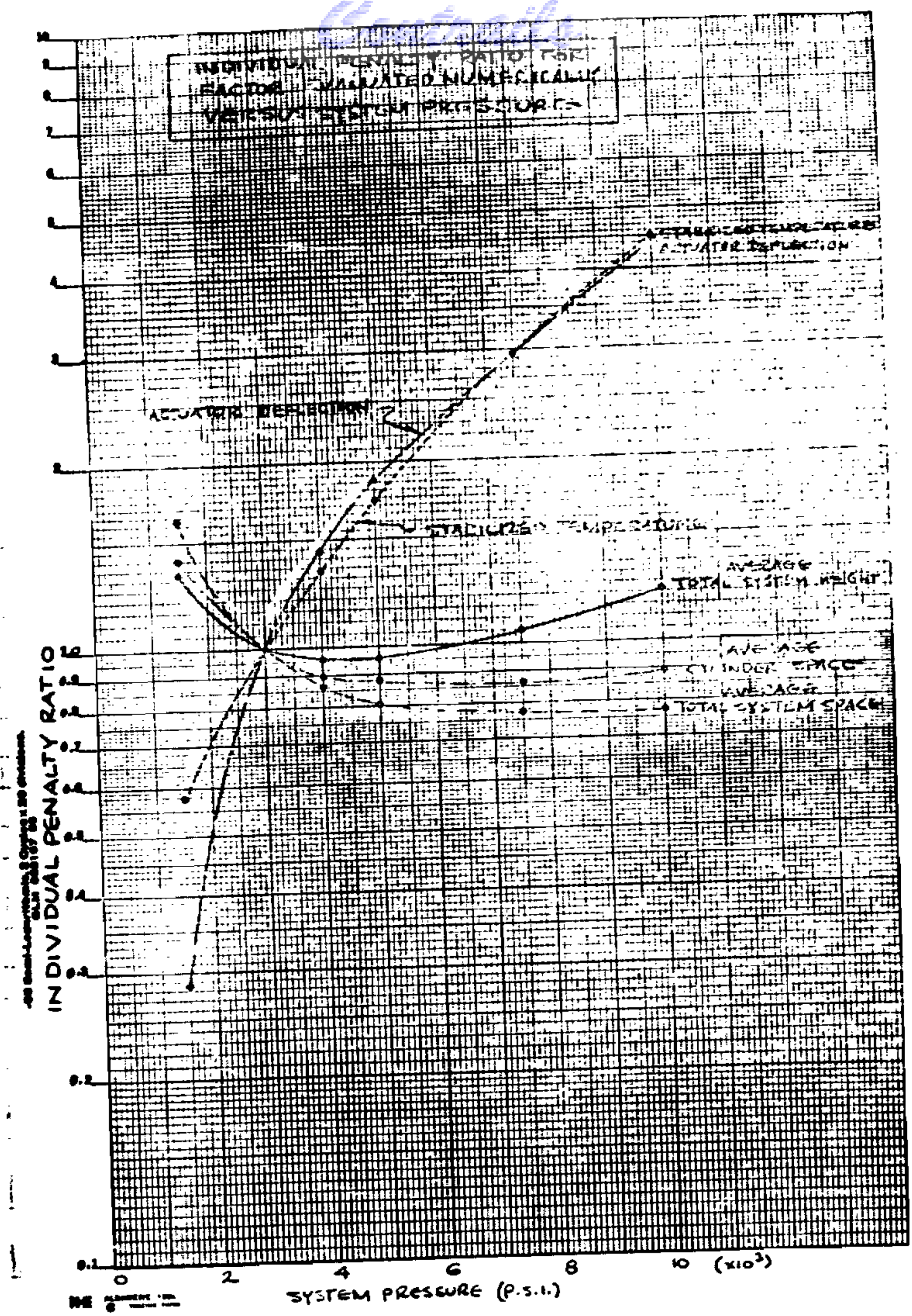
Fig. 34

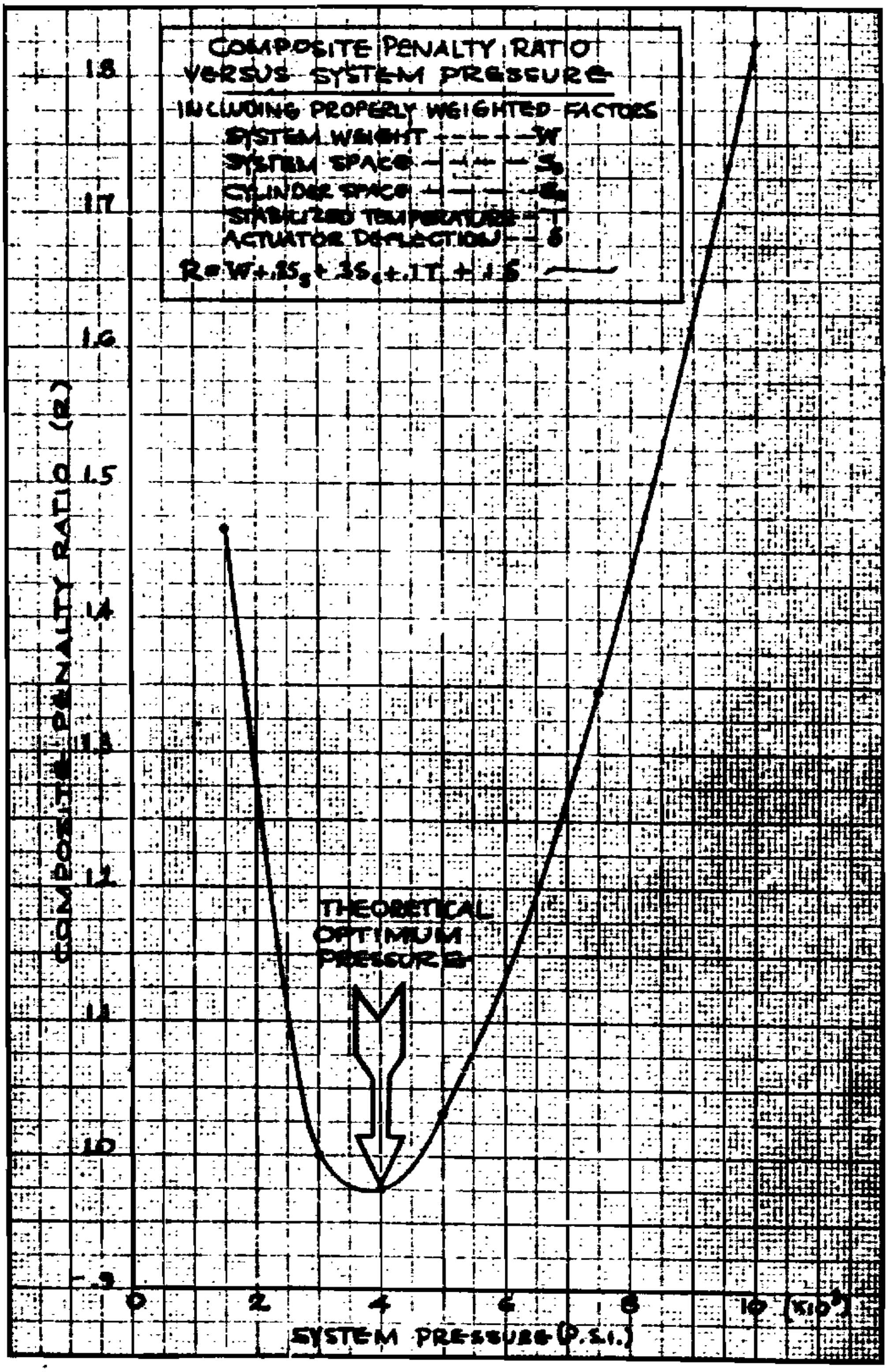

TYPICAL SYSTEM SPACE BREAKDOWN
SPACE VOLUME VERSUS PRESSURE FOR MEDIUM MAIN HYDRAULIC SYSTEM
0
2
4
6
8
10
PLUMBING

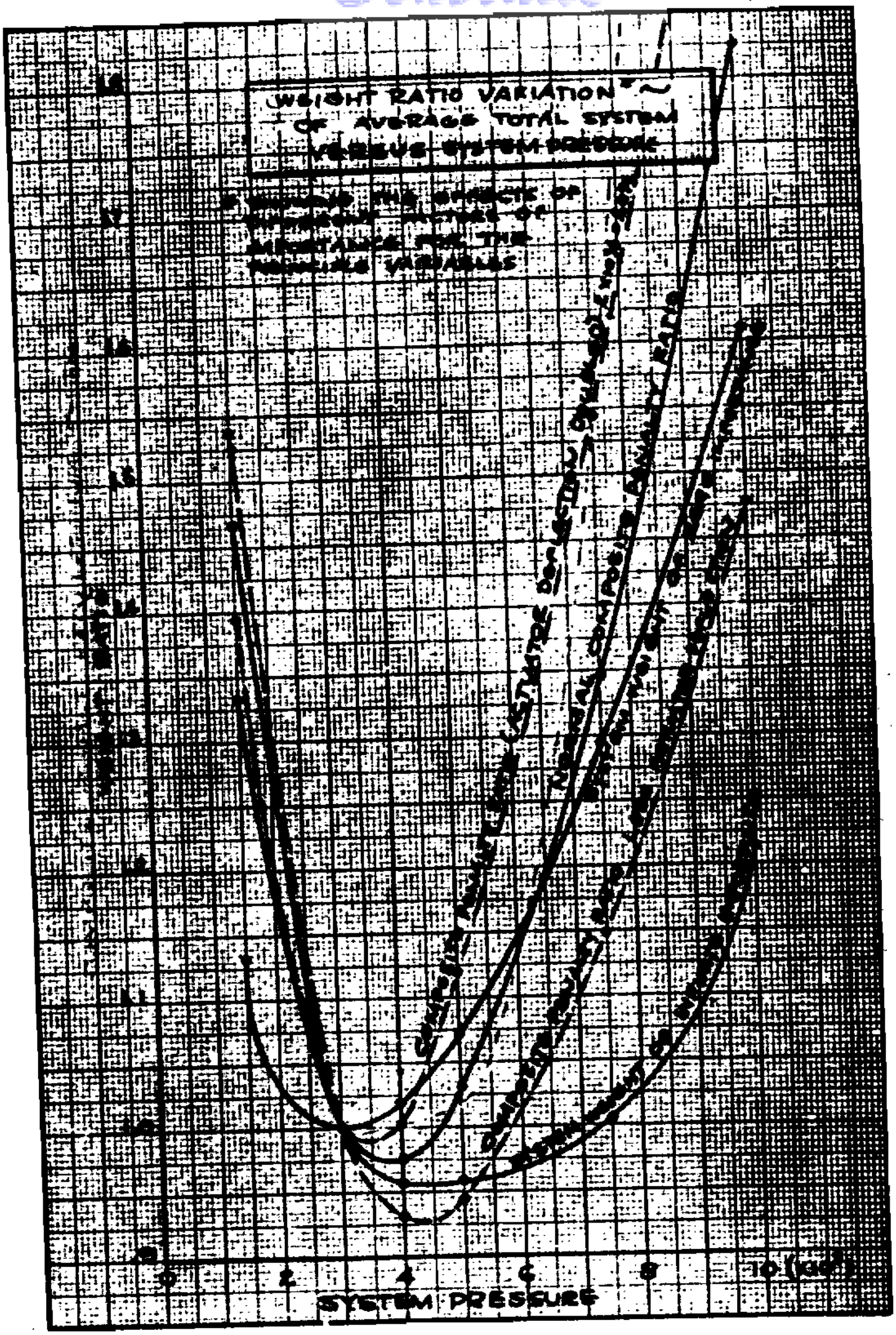

MINIMUM CYLINDER LOAD FOR COLUMN STABILITY VERSUS SYSTEM PRESSURE AT A CONSTANT WORK LEVEL OF 30,000 POUND INCHES
LOAD (LBS.)
6000
5000
4000
3000
2000
1000
0
0
2
4
6
8
10 (10³)
SYSTEM PRESSURE
(P.S.I.)


CYLINDER DEFLECTION* IN PERCENT OF PISTON STROKE VERSUS SYSTEM PRESSURE
* DEFLECTION IS CHANGE IN THE LENGTH OF A HYDRAULIC CYLINDER CAUSED BY A CHANGE IN EXTERNAL LOAD FROM ZERO TO FULL SYSTEM PRESSURE. (PORTS BLOCKED)
CYLINDER DEFLECTION IN PERCENT OF PISTON STROKE
0
1
2
3
4
5
0
2
4
6
8
10 (x10³)
SYSTEM PRESSURE
(P. S. I.)


STABILIZED FLUID TEMPERATURE INCREMENT ABOVE COMPARTMENT TEMPERATURE AND HEAT ENERGY IN HORSEPOWER GENERATED BY HYDRAULIC SYSTEM VERSUS SYSTEM ~ PRESSURE
SCALE "A"
SCALE "B"
1000
100
10
20
15
10
9
8
7
6
5
4
3
2
1
STABILIZED SYSTEM TEMPERATURE INCREMENT (SCALE "A")
HEAT ENERGY GENERATED (HORSEPOWER) (SCALE "B")
STABILIZED SYSTEM TEMPERATURE INCREMENT = ΔT
HEAT ENERGY GENERATED = HP_H
0
2
4
6
8
10 (X10³)
SYSTEM PRESSURE (P.S.I.)


ACTUATOR DEFLECTION
AVERAGE
TOTAL SYSTEM WEIGHT
AVERAGE
CYLINDER SPACE
AVERAGE
TOTAL SYSTEM SPACE
INDIVIDUAL PENALTY RATIO
0.1
0.2
1.0
0
2
4
6
8
10
(x10³)
SYSTEM PRESSURE (P.S.I.)


COMPOSITE PENALTY RATIO
VERSUS SYSTEM PRESSURE
INCLUDING PROPERLY WEIGHTED FACTORS
SYSTEM WEIGHT ------ W
SYSTEM SPACE ------ S_s
CYLINDER SPACE ------ S_c
STABILIZED TEMPERATURE ------ T
ACTUATOR DEFLECTION ------ δ
$R = W + .85_s + .35_c + .1T + .1\delta$
THEORETICAL OPTIMUM PRESSURE
COMPOSITE PENALTY RATIO (R)
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
.9
0
2
4
6
8
10 (×10³)
SYSTEM PRESSURE (P.S.I.)


WEIGHT RATIO VARIATION ~
OF AVERAGE TOTAL SYSTEM
0
2
4
6
8
SYSTEM PRESSURE

4. PRACTICAL SIGNIFICANCE OF OPTIMUM PRESSURE

A review of the historical data presented in the Introduction and Section 1 reveals that the hydraulic system pressures used in the airplanes built during the past fifteen years have steadily increased from 800 psi, in vogue in 1939, to 3000 psi used in all airplaces designed in 1952 and 1953. During this period the horsepower requirements of the systems have greatly increased demanding imoreved design efficiency and indicating the need for high pressure (figure 1-1). This condition, and because the 3000 psi system is a recognized standard system has encouraged its use by designers almost universally.

Actually at this time the systems designers are faced with the well known high temperature problem, which is expected to become even more critical, later. The same high performance aircraft which are burdening the hydraulic system with greater power demands are requiring that the system operate, at least intermittently in high envirommental temperatures, generally ranging from 300° F to 600° F. A great amount of investigation, development, and test work is currently being performed on materials and equipment in order to cope withthis problem. Inasmuch as new products will have to be developed and new specifications and standards formulated, it would appear logical to consider the optimum pressure (4000 psi) as one of the conditions to be met, and thereby obtain considerable savings to the government and the aircraft industry in time and money.

While cognizant of the high temperature situation as described above, it appears that the problems associated with pressure increase are such that it would be feasible to go to the optimum pressure system in the near future with relatively minor changes in existing materials and equipment. Some of the effects of using 4000 psi on the various equipment and materials comprising hydraulic systems are noted below.

4a. Pumps

Pumps are considered to be the most important items of equipment in the hydraulic system. Information recently obtained from the principal aircraft hydraulic pump manufacturers indicates that little or no difficulty will be experienced in obtaining pumps for operation at 4000 psi pressure. (To avoid indicating preference the companies are listed alphabetically.)

Denison Engineering Corp. -- Now have both fixed and variable displacement pumps being tested at 5000 psi.

Hamilton Standard -- Can adapt the existing production pumps to 5000 psi pressure operation in less than one year.

New York Air Brake Company -- Pumps currently used in 3000 psi systems will be suitable for operation in 4000 psi systems either as is, or with slight modifications.

Pesco -- The present 3000 psi pumps will operate satisfactorily at 5000 psi with slight modifications. A 5000 psi pump is now being tested.

Vickers -- Expect no trouble with 5000 psi in hydraulic system pumps. Now using 4500 to 5000 psi pressure in constant speed drives. The life expectancy of existing pumps at the optimum pressure (4000 psi) is approximately 500 hours.

4.b. Fluid

Increase in working pressure affects hydraulic fluid only regarding decrease in shear stability and increase in density and hence, viscosity. The pressure increase from 3000 psi to the optimum pressure (4000 psi) is still well within the limitations of the existing fluids to resist shear stability breakdown, and it is believed that no appreciable change in the life of the fluids will be experienced. In the average hydraulic system the fluid is usually changed because of contamination, or other reasons, long before any shear stability weakness is apparent.

The increase in viscosity due to increase in pressure causes increase in pressure drop. The percent of pressure drop per foot increase, when going from 3000 psi working pressure to the optimum pressure (4000 psi) for the same flow rate and tube size, varies from approximately 5% at 80° F to approximately 26% at -65° F. These figures are for 1/2 O.D. tube x .065 wall at 6 GPM flow. However, this theoretical pressure drop increase is more than compensated for by virtue of lower flow requirements. To accomplish the same job of work at constant efficiency only 3/4 of the flow is required when using 4000 psi as when using 3000 psi pressure, and 4/3 as much pressure drop per foot can be allowed. Refer to figure A-6.

4.c. Actuators

Two types of actuators are commonly used in hydraulic systems, cylinders and hydraulic motors. Since the same type of problems are present in motors as in pumps the practical significance of 4000 psi would be the same as listed for pumps by the corresponding manufacturer.

By far, the greater number of actuators employed in hydraulic systems are cylinders. As a means of power transmission, it is difficult to imagine a simpler or more efficient device. The art of cylinder design and manufacture for 3000 psi working pressure, has at this time reached the state of perfection, that very little service trouble is experienced with these "work-horses" of the hydraulic system. In most cases, except for periodic replacement of seals and back-up rings, no other service is required for years. If a change from 3000 psi to the optimum pressure (4000 psi) were made it is believed that no major design or manufacturing practices need be changed, except that perhaps slightly better control of the gland extrusion gap should be exercised. Although it is recognized that gland design in general and the materials to be used in particular undoubtedly will be considerably affected by high temperature requirements currently being investigated it is expected that the use of teflon or some heat resistant material back-up rings would be required, as the existing AN-6246 leather back-up rings are not entirely satisfactory in some installations at 3000 psi.

Examination of figure B-5 indicates that with many of the existing 3000 psi cylinders plotted thereon, column strength is not the critical design criteria. However, figure B-6 shows that this feature will become more critical when using the optimum pressure (4000 psi). For a given job of work, shorter strokes in proportion to cylinder diameter will be required. The use of shorter strokes will provide higher loads but also more compact designs which is in accordance with general design trends.

In flight control power systems the use of the optimum pressure (4000 psi) and consequent reduction in cylinder size would reduce the rigidity of the system. From figure 3-6 it is observed that piston deflection caused by an external load equivalent to full system pressure is 1.32% of working stroke at 3000 psi. This deflection would be increased to 1.78% at 4000 psi. This reduces the potential response of the servo and may lead to instability where this situation is ignored. In new systems, designed with this problem in mind, no serious trouble should be experienced.

4.d. Flexible Lines

The very serious problems to be overcome, due to high temperature hydraulic system requirements and current difficulties being experienced by hose manufacturers in meeting the high pressure hose specification MIL-H-5512 indicates that considerably less flexible rubber hose would be used in the optimum pressure (4000 psi) systems than are now being used in the standard 3000 psi systems.

Actually the industry, in many cases, is hard-put to obtain qualified high pressure hoses for use in 3000 psi systems. Although hoses are available with sufficient burst strength, 12000 psi for the 3000 psi system and over 16000 psi in sizes 1/4, 3/8 and 1/2 for the 4000 psi system inability to pass the impulse pressure test of 100,000 cycles at 1 1/2 times the working pressure causes rejection of much of the hose that is manufactured. In order to prevent shortages on many production airplanes the government has allowed deviations whereby inferior hose can be used in certain installations. It is recognized that high pressure hose is required in some installations where the peak pressure impulses are negligable, whereas in others they are quite severe. This policy will of course relieve the situation considerably.

While a more judicious evaluation of operating conditions of particular hose installations would justify the use of existing high pressure hoses in some places in the optimum pressure (4000 psi) system, it is believed that the use of swivel joints will provide a solution in almost all cases. These joints, made by several manufacturers, are now readily available for 3000 psi aircraft hydraulic systems. In fact many designers, unable because of limited space, to accommodate the bend radii required for flexible hoses, are using these joints.

Coiled rigid tubing has been used extensively by some designers for installations requiring small deflections in order to avoid hose troubles. This practice would be particularly applicable for the optimum pressure (4000 psi) system.

Considerable progress has been made in developing flexible metal hose for use in aircraft power plants to meet fire proofing requirements. In the smaller sizes some of this hose has been developed to withstand relatively high pressures, burst pressures as high as 11,000 psi in the 1/4 inch size having been reached in some cases. Although still inadequate through the range of sizes commonly used in high pressure hydraulic systems, it is believed that further progress will be made and that this hose will eventually be available for use in 4000 psi systems.

4.e. Tubing and Fittings

It is assumed that the same minimum ratio of burst pressure to working pressure, 4 to 1, for tubing and fittings will be maintained in the optimum pressure (4000 psi) system as used at present for the existing standard pressure systems. This provision precludes the use of 61ST tubing as the wall thicknesses of the tubes becomes excessive. Consequently corrosion resistant steel tubing will be used. One-eighth hard tubing, Specification MIL-T-6845 or AMS-5566, has been used extensively for 3000 psi systems and will be suitable for the optimum pressure (4000 psi) system.

Tube flaring has been, to a more or less degree, a process requiring careful shop control. If the tools are not maintained in top condition or if the tube ends to be flared are not smooth, cracks or marks will develop, causing many rejections. It is expected that with the heavier wall thicknesses required for a 4000 psi system these problems may become more acute. Also it is believed by some designers that the integrity of a flared tube joint for service at pressures exceeding 3000 psi is questionable. Extensive vibration-impulse tests have proven that the flared tube joint is quite inferior to the flareless fitting joint in this regard. For the above reasons it is expected that flareless fittings would be used extensively with the optimum pressure (4000 psi) system. These fittings are readily available, and are now widely used in 3000 psi systems for Naval and Commercial aircraft.

4.f. Seals

It is expected that the standard AN-6227 O-rings, used with teflon back-up rings, in standard grooves would be satisfactory for use in the optimum pressure (4000 psi) system. Information has been received from several airframe and equipment manufacturers that they have successfully used AN6227 O-rings in standard grooves at 5000 psi. The Air Force has run laboratory tests with standard AN-6227-28 O-rings and spiral teflon back-up rings wherein 50,000 cycles were completed at pressures of 15,000 and 20,000 psi. The leakage was approximately 2.6 cc per 1000 cycles or 1.3 drops per 25 cycles. The existing specification for 3000 psi cylinders allows

1 drop external leakage at the piston rod gland per 25 cycles. Extreme cold flow of the teflon rings was encountered on these tests which would not be present at 4000 psi pressure. It is believed, however, that some reduction in the diametrical clearances allowed in MIL-P-5514 would be made for use with 4000 psi pressure to provide maximum protection against seal extrusion. This could be done in most cases without causing serious manufacturing problems. The life of the O-ring as affected by normal wear should be somewhat less using 4000 psi pressure and the tendency of the O-ring toward spiral failure somewhat greater due to increased friction of the O-ring against the moving metal parts when under pressure. However, it is believed that ample life would be obtained if no extrusion occurred and that well lubricated rods would prevent spiral failures.

The results of current tests and investigations relative to the high temperature problem and the development of new materials for O-rings and back-up rings will of course be the principal factors affecting gland and seal design in the near future.

4.g. Valves

When considering the practical effects of using the optimum pressure (4000 psi) on various elements in the hydraulic system, perhaps it is an over simplification to include under one heading components with such various functions and characteristics as hydraulic system valves. However, in all cases these units are used to control the direction, flow rate, or pressure of the hydraulic fluid. Hence the problem of internal leakage is common to all valves; it is assumed that external leakage must be prevented in all cases.

With unbalanced poppet type valves of various kinds the increase in pressure may require tougher valve seats to provide adequate life. Judicious use of dampers however can alleviate to a large extent the dynamic effects of pressure increase. In balanced poppet valves no appreciable difficulty should arise.

With slide valves of various kinds it is expected that generally, the same fits and finishes for spools and bodies or sleeves would be used for 4000 psi valves as now used for the 3000 psi valves. It is assumed that this practice would be followed wherever possible in order to minimize manufacturing difficulties, so that only in some special cases would closer fits be required. Consequently more internal leakage would have to be tolerated due to the increase in pressure. The valve leakage horsepower when using 4000 psi pressure is 1.43 times the valve leakage horsepower when using 3000 psi pressure (See Equation 2D-7, Section 2D).

For use in 4000 psi systems valve bodies would have to be more rigid to prevent distortion. In many cases, however, the bodies will contain sufficient bulk, for manufacturing and mounting considerations, to accommodate the higher pressure without any change. In many slide valves, of course, considerable internal leakage can be tolerated and in some cases internal leakage is not only desirable but mandatory to contribute to the stability of the circuit. With valves of this type no problem should be encountered when using the optimum pressure (4000 psi).

Due to the increase in pressure internal parts in general will be smaller in size. Consequently somewhat more difficulty will be encountered during assembly of the various parts, particularly the installation of O-rings and back-up rings.

As several of the valve manufacturers have already built sample valves of various kinds for 5000 psi pressure, it is believed that 4000 psi valves could be available in one year if the need for them existed.

4.h. Accumulators

The cylindrical rather than the spherical accumulator appears more adaptable for use in the optimum pressure (4000 psi) system because of simpler design and manufacturing problems. Cylindrical accumulators for 3000 psi systems are readily available from several manufacturers some of whom have built 5000 psi accumulators for test purposes. It is believed that close control of the extrusion gap between the piston and bore and the use of teflon back-up rings will be required for best results. It is expected that the life of the piston O-rings will be slightly less than those now in 3000 psi accumulators due to the higher pressure but is expected to still be satisfactory providing the proper finish is maintained on the cylinder bore.

The preload air pressure would be increased to obtain maximum utility from the accumulators. As 3000 psi air compressors and intensifiers are now widely used, the increase in preload which will fall below 3000 psi pressure should create no serious service problem.

4.j. Reservoirs

Use of the optimum pressure (4000 psi) system will not affect hydraulic reservoirs except that in general the reservoirs will be 25% smaller in volume.

5. COST

When this optimum pressure study was undertaken it was realized that any change from the present standard pressures would involve additional costs. It was generally agreed that if costs were considered in the determination of an optimum pressure, they would only serve to weight the answer toward the present pressures and thus give a false indication. For this reason cost was considered constant in the optimum pressure determination.

Even though costs were considered constant in the choice of the optimum pressure it is important that some thought be given to the economic aspects of a change in the hydraulic system operating pressure. An analysis of the costs involved show that they can be roughly divided into two major classifications, the non-recurring costs and the recurring costs. The non-recurring costs are basically those expenses that would be incurred in bringing the industry up to an equivalent state at the new pressure. The recurring expenses are those added costs that may be involved in the manufacture of standard or proprietary components at the higher pressure as well as any added costs involved in the development and manufacture of new items.

The fixed or non-recurring costs would represent a very sizable sum that would have to be spent during the transition stage of the adoption of the higher pressure. First new specifications and design requirements would have to be set up. A complete series of standard items such as check valves, accumulators, relief valves, shuttle valves, filters, unloader valves, etc., as well as proprietary items such as solenoid valves, pumps, brake valves, motors, flow regulators, etc., must be designed by the many component manufacturers, prototypes built, tested and when applicable qualified by the services. New production tooling would have to be designed and built, new forging dies, patterns, extrusion dies are required. Testing equipment in the various laboratories throughout the industry as well as the quality control test equipment and the portable test stands and auxiliary hydraulic power equipment used in servicing the aircraft would have to be rebuilt to handle 4000 psi. When one considers the number of companies both airframe as well as component manufacturers involved, as well as the number of items that each of the component manufacturers would have to redevelop and the changes involved in the test and service equipment belonging to these companies as well as the services, then one can begin to appreciate the expense involved in this change.

Those expenses classified as recurring costs are those added manufacturing costs occasioned by the change. It is assumed that the general level of manufacturing technique would remain the same. No general improvement of quality level of precision components can be expected. To keep the control forces at the same level as in present valves the pilot sections of various control components would of necessity have to be smaller and because of the handling problems associated with these already small parts the cost would undoubtedly rise. As these increased costs would only be associated with a small percentage of individual component parts the overall effect, of increased pressure on manufacturing costs, would be very small.

The overall cost of this pressure change to the industry would be reduced to a large degree if the change is made concurrent with the development of high temperature components.

The major development, testing, and tooling costs would have to be expended to get a standard line of high temperature components into production and these costs could rightly be charged to the high temperature program. The purchase of new 4000 psi inspection test stands and auxiliary hydraulic power equipment for servicing the aircraft would be the major item chargeable to the high pressure program.

BIBLIOGRAPHY

1. AFTR No. 5997, James E. Cambell: Investigation of the Fundamental Characteristics of High Performance Hydraulic Systems, June 1950

2. Lionel S. Marks: Mark's Mechanical Engineers Handbook, McGraw-Hill Book Company, Fifth Edition, 1951

3. Harold W. Adams: Aircraft Hydraulics, McGraw-Hill Book Company, 1943, page 80

4. Raymond J. Roark: Formulas of Stress and Strain, McGraw-Hill Book Company, 1938

5. Fred Habit: "Critical Buckling Loads for Hydraulic Cylinders," Product Engineering, July 1950, pp 108-112

6. Alfred S. Niles, Joseph B. Newell: Airplane Structures, Third Edition, John Wiley and Sons, 1946

7. R. S. Cumbee: "Higher Pressure Trend in Modern Presses," Applied Hydraulics, August 1950, pp 15-17

8. "Hydraulics," Flight, August 17, 1950, pp 191-196

9. WADC Tech. Report No. 52-37, E. A. Heusel: Investigation of Pressure Surges in Aircraft Hydraulic Systems, March 1952

10. W. C. Trautman, R. E. Middleton: Comparison of Hydraulic and Electrical Accessory Systems in Aircraft, S.A.E. Journal, March 1944, pp 94-99

11. AFTR No. 6685, Neil S. Furby: Development of High-Temperature Aircraft Hydraulic Fluids, and Supplements 1 and 2, November 1951, October 1952 and September 1953

12. J. B. Giacoble: "Tips on Stainless for Aircraft Hydraulic," Aviation Age, July 1952

13. Merrell R. Fenske: Fluids, Lubricants, Fuels and Related Materials PRL 5.2, 2.62, 5.3, 5.4, and 5.5, March 10, 1952, October 25, 1951, September 25, 1952, November 22, 1953, February 3, 1953

14. Ivan H. Driggs: "The Airplane Growth Factor and How to Control It," Aeronautical Engineering Review, September 1952

15. M. D. Stone: "Large Hydraulic Forging Presses," Transactions of the A.S.M.E., July 1948

BIBLIOGRAPHY

16. O. A. Wheelan: "Titanium, Its Processing and Subsequent Mechanical Properties," SAE Journal, March 1952

17. E. L. Carletta and E. M. Hebein: Limitations of Synthetic Rubber Packings for High Temperature Application, SAE Report No. 72, April 1953

18. George E. Keller: Adventures in Extreme Temperature Hydraulics, SAE Report No. 71, April 1953

19. Frank C. Mittell: High Temperature Problems Associated With Aircraft Hydraulic Systems, SAE Report No. 70, April 1953

20. J. C. Mosteller, John A. King: The Development of High Temperature Hydraulic Fluids, SAE Report No. 73, April 1953

21. Third Annual Report on The Development of a High Temperature Aircraft Hydraulic Fluid, California Research Corporation, February 1953