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AN ASSESSMENT OF SEPARABLE FLUID CONNECTOR SYSTEM PARAMETERS TO PERFORM A CONNECTOR SYSTEM DESIGN OPTIMIZATION STUDY

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16. ABSTRACT

The engineer today is more than ever concerned with the optimum solution to highly complex problems involving numerous, and often conflicting or unspecific demands made by stringent requirements in the areas of environment, cost, materials, performance, date of delivery, reliability, safety, and numerous other constraints imposed upon the designer. In this paper the key to optimization of design where there are a large number of variables, all of which may not be known precisely, lies in the mathematical tool of dynamic programming developed by Bellman. This methodology can lead to optimized solutions to the design of critical systems and in a minimum amount of time even when there are a great number of acceptable configurations to be considered.

To demonstrate the usefulness of dynamic programming, an analytical method is developed for evaluating the relationship among existing numerous connector designs to find that configuration which is best(optimum). The data utilized in the study were generated from the some 900 flanges designed for six subsystems of the S-IB stage of the Saturn IB space carrier vehicle. The algorithm is general enough to be useful in optimizing design and systematizing decision-making in a variety of related technical fields such as aeronautics, construction, electronics, etc.

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LIST OF SYMBOLS

Symbol	Definition
A	flow area
A_g	gasket bearing surface
C_k	parameter value increment
C_k'	criterion function
C_p	permeation rate constant
D_c	compression ratio, degree of contact, for gaskets or seals
D_B	bolt circle diameter
D_G	gasket center diameter
DI	degree of importance
E	effectiveness
E_f	modulus for flange material
E_g	modulus for gasket material
E_s	modulus for seal material
E_y	modulus at yield stress
F	flange force
F_b	bolt force
F_p	pipe force
ID	inner diameter
L	flow path

LIST OF SYMBOLS (Continued)

Symbol	Definition
Max_k	maximum value of the kth parameter
$Max_k(j)$	maximum value of the kth parameter of a subsystem (j)
Min_k	minimum value of the kth parameter
$Min_k(j)$	minimum value of the kth parameter of a subsystem (j)
M	bending moment
OD	outer diameter
p	total number of parameters
P_k	a system level parameter
PI	performance index
$P_k(i, j)$	value of the kth parameter for the ith configuration of the jth subsystem
PP_k	parameter constraints
Q	leakage rate, quantity of medium
R_k	the kth parameter range
S	seal stress
S_c	seal contact stress
S_y	seal yield stress
S_k	return value
S_{min}	minimum seal stress
S	gasket or seal stress
W	weight

LIST OF SYMBOLS (Continued)

Symbol	Definition
X_k	input function
a	mean perimeter of annulus
a'	distance between force vector F_b and center-line of gasket
b	seal contact width
c_k	number of equally spaced points in a subset
d	a decision made at a stage
d_k	a decision
d_y	gasket depth penetrating into flange asperities
d_t	average height of flange surface asperities
e	bolt offset
f'	scale factor
f	some function
f_j	a quantity for the jth subsystem
f_w	factor weight
h	some function
h^3	conductance parameter
h_c	compressed gasket height
h_g	gasket height
i	any configuration
i_c	interface leakage conductance

Symbol	Definition
j	any subsystem
k	any parameter
ℓ'	scale factor
m_c	number of possible configurations
m	total number of subsystems
m_b	moment
n_j	number of configurations pertaining to the jth subsystem
n_k	number of points in a set c_k
Δp	pressure difference
$\bar{p}$	mean pressure
p_1	leak exit pressure
p_2	sealed fluid pressure
p_0	standard atmospheric pressure
p_{int}	internal pressure
p_{ext}	external pressure
p	number of parameters
r_m	radius between force vector F_b and pipe centerline
r_o	outer seal radius
r_i	inner seal radius
$\bar{r}_s$	middle seal radius

LIST OF SYMBOLS (Concluded)

Symbol	Definition
r_D	inner duct radius
r_B	bolt circle radius
t	flange thickness
t_c	clearance between plates (flanges)
w	flange width
w_s	gasket (seal) width
x	gasket distance compressed after F is applied
δ	flange deflection
$\bar{\lambda}$	mean free path of gas molecules of mean pressure $\bar{p}$
μ	fluid viscosity (absolute)
ϵ	molecular correction factor
σ	stress
ω	flange radial width
ϕ	a function

CHAPTER I

INTRODUCTION

The engineering profession is primarily concerned with design and design analysis, which have been closely associated with engineering by means of handbooks, standards, and experience. Today's designs are becoming more and more complex which creates unique decision patterns to find the best solution for each product. The design engineer has to focus his attention upon the study of the whole system and individual components involving large numbers of variables. To find the best solution for a complex design by exhaustive search becomes expensive and inefficient. To avoid this expense and inefficiency, mathematical programming was born. The research described herein uses dynamic programming developed by Bellmann [1], which is a method of mathematical programming. This method is used to determine the optimum design for a separable bolted connector system, which is a rather complex problem in aerospace applications.

The term "separable bolted connector" means the complete assembly, which consists of two flanges, the gasket or seal, the bolt, and the nuts. The term "flange" identifies the structural portion at the end of the pipe. Figure 1 is a schematic of a separable bolted connector system.

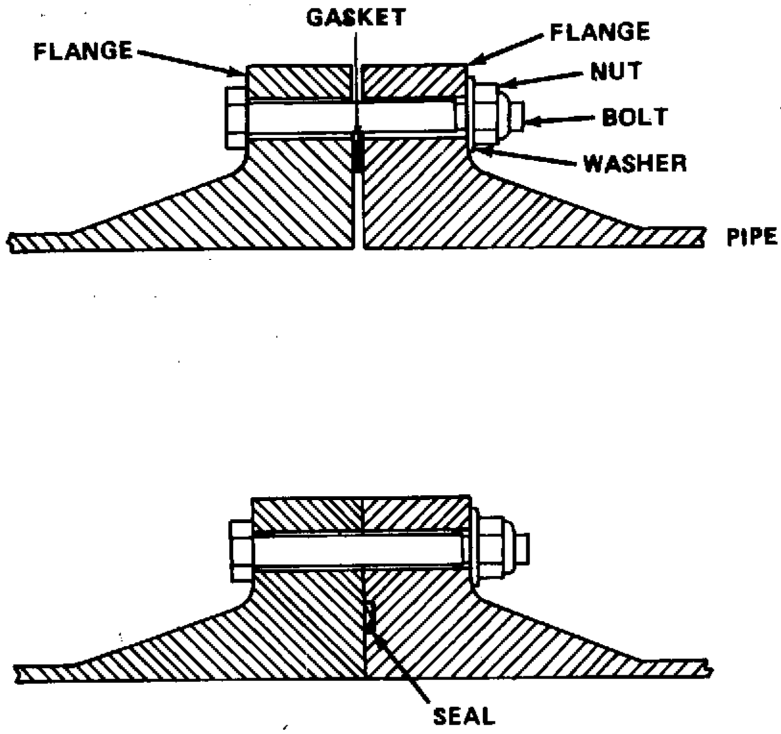


FIGURE 1. BOLTED SEPARABLE CONNECTOR

Separable bolted fluid connectors used in space vehicle and space module applications present potential hazards if they are not absolutely leak tight. During operation in these applications, strenuous environments are imposed and each connector has to withstand varying pressures, temperatures, and vibrations without losing its sealing ability. Zero leakage, minimum weight, and minimum envelope are the most important design parameters for separable connectors.

The "zero leakage" requirement for most of these connectors demands a special effort from the design and materials engineers to meet this goal. The dominating factors in the design of separable connectors are performance, reliability, and weight, whereas the costs are of a secondary nature. This rating of importance is typical for aerospace design because the failure of a single component can be the margin between success and failure of a mission. One such critical component is the separable connector.

Almost any leak that develops in a stationary or mobile terrestrial system can be easily detected and repaired. For example, detecting and replacing a leaking head gasket on an automobile does not present any technical problem with the exception of its unavailability during repair. Also, industrial production of hazardous propellants does not present serious leakage problems. Any leak at the plant is immediately detected by a system of sensitive gas sniffers or gas analyzers that are interconnected to a warning system. Remotely controlled shutoff valves immediately stop the flow of the media, and emergency drain and collection systems confine the leaking media to avoid mixing. Reliable fire detection systems turn on automatic fire fighting equipment to localize and extinguish fires.

Such precautionary systems cannot be incorporated into a space vehicle because of weight and space restraints imposed on the design. However, some engine connector systems have drain systems installed to collect leaking fuels. Past experience has proven that almost any amount of propellant leaking to the outside of a system presents a potential explosion hazard. There are many ignition sources onboard a vehicle, e.g., heat, electrical sparks, and static electricity, which might ignite such a propellant mixture. Another cause of losing a mission due to a leaking connector would be the loss of air pressure in a cabin and/or the loss of mission critical gas and liquid supplies.

The many lines for propellant feed, pressurization, venting fill and drain systems, tank openings, and feed-throughs needed for a vehicle stage require a great number of separable connectors. The performance criterion "zero leakage" imposed on almost every separable connector, makes this design very difficult, because the connectors are exposed to very strenuous environments. The long time period between connector system assembly and checkout and the launch of the vehicle causes a setting of the gasket material and some material creeps, resulting in a relaxation of the initial seal force. It is very difficult to predict such relaxation of the seal force that might result in a leak. Later, during launch, the wide temperature and pressure ranges that the connector is exposed to, coupled with the dynamic and static forces that apply substantial loads to the system, make the problem of designing separable connectors difficult. The fact that the magnitude of leakage increases with the extended exposure to high vacuum makes it almost impossible to design separable connectors meeting the "zero-leakage" requirements.

The numerous design handbooks, specifications, tables, and reports used by different organizations designing separable connectors contributed heavily to the many different and unnecessary configurations. The variables were weight, surface machining, facing, and dimensioning. As long as the connector did not leak more than the specified tolerable leakage rate, the connector was accepted. It is surprising that none of the 90 different flange drawings that were investigated showed the flange weight, which illustrates the point that the flange weight was not considered important. This philosophy for designing flanges was certainly costly and not the best way to solve the weight problem.

In this research, an effort is made to find the best separable fluid connectors among the existing ones used for the liquid oxygen system. The optimum configuration is that which, subject to a specific environment, satisfies and properly fulfills the imposed criteria.

The objective of this study is to develop an analytical method for evaluating the relationship among the separable connector design parameters subject to the analysis. The method developed is based on the principles of dynamic programming and permits evaluation of alternate design configurations in order to optimize the system and achieve maximum effectiveness.

CHAPTER II

STATEMENT OF THE PROBLEM — THE DESIGN DECISION

Given a function to be performed, a number of possible ways to achieve the task become apparent. Differences in performance, relaibility, configuration, shape, and material are just a few variables that are encountered in determining the optimum design for a separable bolted connector system. To make a design decision is therefore a very complex undertaking. A design decision process requires that a single strategy must be chosen from a great number of alternate possible strategies. A means must be found to discriminate between good and bad decisions. There is sufficient evidence that the commonly used decision process using only intuition, experience, and feelings is inadequate and has caused numerous failures. A typical example substantiating this statement is the design of separable bolted connectors used for space applications. The high leakage rates experienced and the many engineering change orders modifying the design of separable connectors reflect the fact that the design is in a state of flux, and the optimum design needs to be determined. The design engineer has to find the best, the optimum, configuration among the many existing ones. Because of the complexity of such a task, some procedure must be applied to evaluate all these configurations and some method has to be found to measure the parameter values of each configuration for comparison and tradeoff purposes to select the optimum design. The complexity of a separable connector system is clearly demonstrated in the design tree shown in Figure 2.

The data for this design tree were taken from the Saturn IB first stage separable connector designs. The tree is composed of four subsystems: (1) the flange, (2) the gasket, (3) the bolt, and (4) the nut. Each of these subsystems is composed of configuration, material, shape, mounting, and finish.

Under the assumption that interaction exists among the unique possible paths, a total of 2,654,208 alternatives have to be taken into consideration. One such path might be 1-4-6-8-15-23-30-38-39-41-44-46. It is true that many inferior and infeasible branches of the tree and their relationships and dependencies can be filtered out by visual inspection alone, but the reduced design tree still presents a problem of great magnitude.

The need to find a way to solve such a design problem provides the environment for the use of Operation Research (OR) techniques. The author has attempted to prove the usefulness of OR methods for the determination of the best separable connector system for the given criteria. Many reports on the subjects of optimization and effectiveness studies in systems engineering, value analysis, value engineering, and decision and value theory have been written, but they are difficult to apply to design decisions by the design engineer.

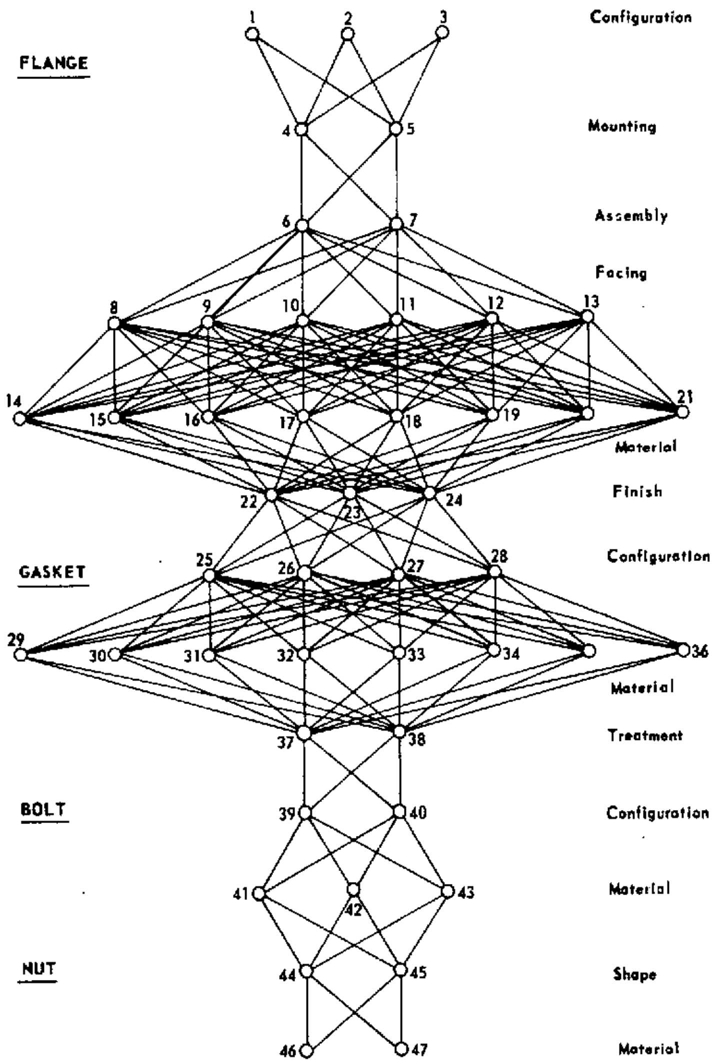


FIGURE 2. LOX SYSTEM DESIGN TREE FOR SATURN IB FIRST STAGE BOLTED SEPARABLE CONNECTORS

NOMENCLATURE

FLANGE

Configuration

1. Taylor Forge
2. Low Profile
3. Optional

Mounting

4. Fixed (Integral)
5. Loose

Assembly

6. Open
7. Metal-to-Metal

Facing

8. Flat
9. Raised
10. Male and Female
11. Groove
12. Tongue and Groove
13. Lap Joint

Material

14. AL 2024
15. AL 5052
16. AL 6061
17. AA-596 Series
18. CRES
19. AL 5456
20. ST 4130
21. ST 1020

Finish

22. 32 microinch
23. 65 microinch
24. 125 microinch

GASKET OR SEAL

Configuration

25. Flat Small
26. Flat Wide
27. O-Ring
28. Pressure Equalizing

Material

29. MS-29513-200 Series
30. Allpax 500
31. MIL-P-5516
32. Silicon Rubber
33. QQ-C-576
34. Narmco
35. AMS 7270C
36. Johns Manville

Treatment

37. Nontreatment
38. Luboil

BOLT

Shape

39. Hexagonal
40. Allen Head

Material

41. Aluminum
42. CRES
43. Steel

NUT

Shape

44. Hexagonal
45. Round

Material

46. Aluminum
47. Steel

As mentioned previously, this study should demonstrate that the application of OR methods results in maximum performance, optimum weight, and optimum costs of separable fluid connectors used for the LOX system of Saturn IB vehicles. For this research, a method was developed to find the optimum connector design from a family of different configurations that were designed, tested, and built for the prototype and flight articles.

CHAPTER III

HISTORY OF THE PROBLEM

The Saturn IB Separable Connectors

During the development of the Saturn and other space vehicles, a serious problem developed in the separable connector systems. Leaks developed to such an extent that vehicle tests and launches had to be postponed, and in some cases, the mission was abandoned.

To build the first stage of the Saturn IB vehicle, which develops a thrust of about 1,500,000 pounds, a total of approximately 900 flanges varying in diameter from 1.5 to 22 inches was designed for (1) the liquid oxygen (LOX) systems, (2) the gaseous oxygen (GOX) systems, (3) the fuel systems, (4) the fuel pressurization systems, (5) the eight engines with heat exchangers, and (6) the water-quench and LH_2 cooling system.

Flange Criteria and Configurations

A total of 118 different flange configurations was designed, as shown in Table I, to meet the requirements shown in Table II. As shown in Table III, 26 different materials were used for manufacturing the flanges, and 30 different materials were used for the gaskets, as shown in Table IV.

TABLE I

NUMBER OF DIFFERENT BOLTED SEPARABLE CONNECTOR CONFIGURATIONS USED FOR THE TYPICAL SATURN IB STAGES

System	No. of Different Configurations	No. of Connections per System	Total No. of Flanges
LOX	22	86	172
GOX	34	60	120
Fuel	22	81	162
Fuel Pressure	11	26	52
Engines	21	168	336
Others[a]	8	28	56
Total	118	449	898
Total number of flanges used for 15 stages			13,470

a. Water-quench and LH_2 cooling system.

More than 13,470 flanges of a nominal diameter from 1.5 inches to 22 inches were manufactured for the first stages of the 15 Saturn IB flight vehicles. The frequency of use for different diameters is shown in Figure 3. The large number of different flange configurations shows that there was neither an approach made to standardize nor any optimization study performed to reduce the number of configurations to a minimum. This confirms the previous statement that in almost every case, the flange design was a product of the designer's experience, intuition, and data retrieved from different specifications, publications, and handbooks.

TABLE II

NOMINAL DIAMETER, OPERATIONAL PRESSURE, AND TEMPERATURE CRITERIA FOR THE SATURN IB FIRST STAGE BOLTED SEPARABLE CONNECTORS

Nominal Diameter (inches)	1.5, 2.0, 2.25, 2.5, 3.0, 3.8, 4.0, 5.0, 5.75, 6.2, 6.82, 6.875, 7.0, 8.0, 9.0, 10.0, 11.0, 12.0, and 22.0
Pressure Range (psi)	0 to 15, 0 to 30, 0 to 80, 0 to 140, 0 to 150, 0 to 300, 0 to 600
Temperature Range (°F)	ambient to -100, -300, and -429 and ambient to +70, +700, and +1,000

TABLE III

FLANGE MATERIAL USED ON THE SATURN IB FIRST STAGE

System	Flange Material Callouts	No. of Different Materials
LOX	1 4 5 8 9 10 11 15 19	9
GOX	1 4 5 8 9 10 15 17 18 19	10
Fuel	2 4 5 8 9 10 13 15 16 17 19 20	12
Fuel Pressure	1 3 15 16 19 22	6
Engines and Others[a]	1 6 7 8 9 11 14 19 20 21 23 24 25 26	14

Material Nomenclature[b]

1. QQ-A-267	8. QQ-A-601	15. AA-5456 H321	22. MIL-A-19842
2. QQ-A-268	9. QQ-S-763	16. AA-5052	23. MIL-S-6758/416
3. QQ-A-318	10. QQ-S-766	17. AA-2024-T4	24. MIL-S-7952/1200
4. QQ-A-327	11. QQ-3-765	18. MIL-A-19842 H24	25. MIL-S-6758/4130
5. QQ-A-355	12. AA-5458	19. AA-CSTG	26. MIL-S-6721A
6. QQ-A-362	13. CRES 300	20. AA-356-T6	
7. QQ-A-596	14. NA5-26069B	21. AA-AN-A-9	

a. Water-quench and LH_2 cooling system.

b. These materials present a total of six separate groups of aluminum alloys and four groups of steel alloys.

TABLE IV

GASKET AND SEAL MATERIAL USED ON THE SATURN IB FIRST STAGE

System	Gasket Material Callouts										No. of Different Materials
LOX	1	2	4	30							4
GOX	1	3	4	6							4
Fuel	7	8	9	10	11	12	13	28	29		9
Fuel Pressure	4	11	14	15	16	17	18				7
Engine and Others[a]	19	20	21	22	24	25	26	27	28	29	10

Material Nomenclature

1. Allpax 500
2. MC-246
3. Johns-Manville No. 76
4. MC-252
5. 0-8-2857-1
6. MS-29513-268
7. MS-29513-268
8. MS-29513-262
9. MS-29513-254
10. MS-29513-513
11. MS-29513-236
12. MS-29513-224
13. MS-29513-261
14. AMS 727 0C
15. MS-29513 264
16. MS-29513 244
17. MS-29513 243
18. MS-29513 242
19. MIL-P-5516
20. AN6627B-19
21. Silicon Rubber
22. Copper QQ-C-576
23. Canadian Asbestos/304
24. MS-29513-251
25. Silicon AMS-3302
26. MS29513 251
27. MS-29513 258
28. MS-29513 450
29. MS-29513 248
30. Narmco

a. Water-quench and LH_2 cooling system.

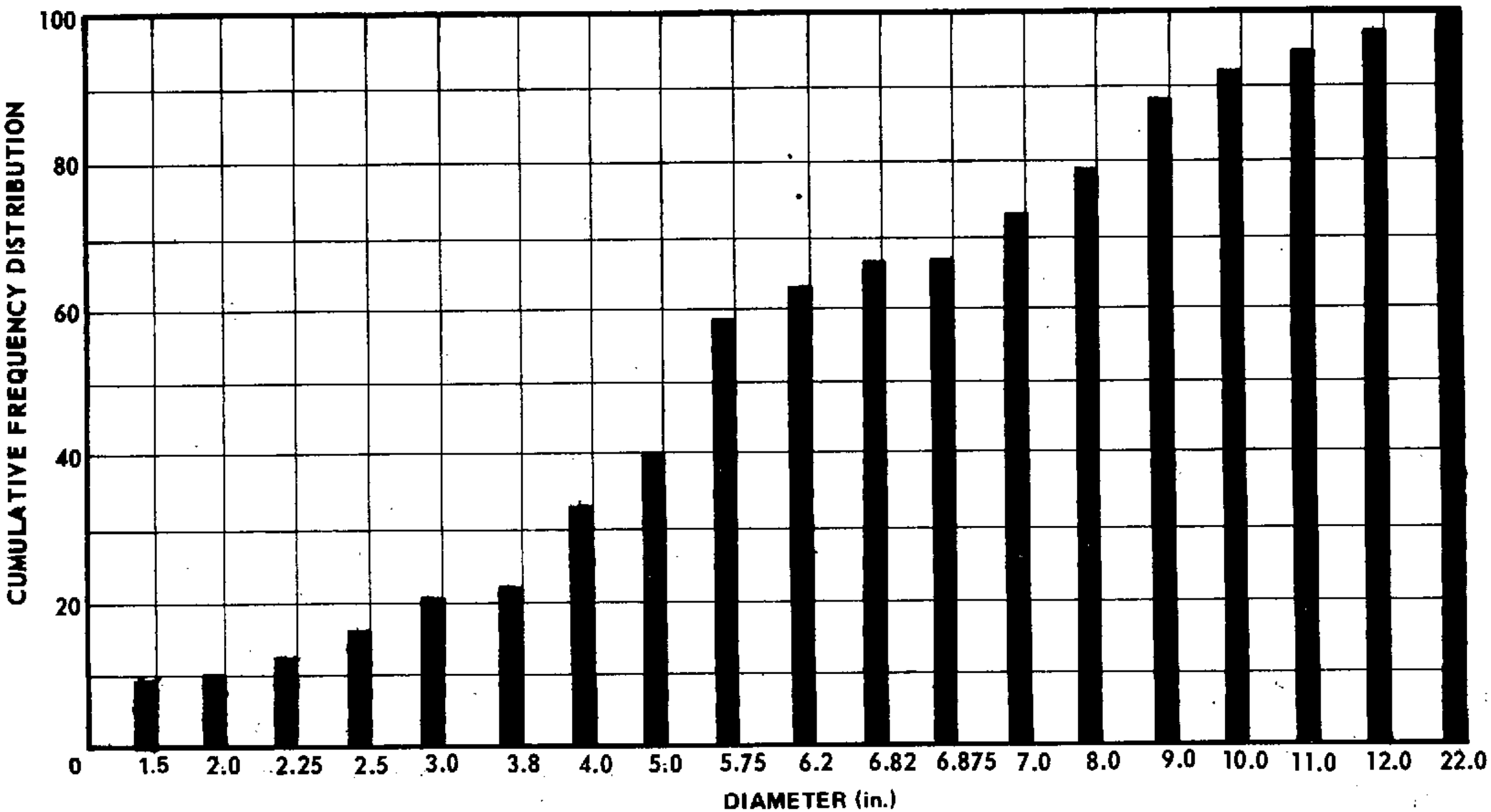


FIGURE 3. CUMULATIVE FREQUENCY DISTRIBUTION OF FLANGES 1.5 TO 22 INCHES IN DIAMETER FOR THE SATURN IB FIRST STAGE

Flange Configuration Examples

Figures 4 and 5 present a few examples of the 118 different separable connector configurations demonstrating the inconsistency of the design. The flange configuration examples are presented in two groups: (1) the LOX and GOX system, and (2) the fuel and engine system.

Typical differences among the flanges are:

1. Flange facings such as flat, tongue and groove, groove only, and male and female were used differently for the same applications.
2. Thicknesses of the left flange and the right flange.
3. Proportions of the bolt circle diameter and the outer flange diameter.
4. Location of the gasket with respect to the bolt circle diameter.
5. Surface finish of the left flange and the right flange.
6. Gasket configuration such as flat wide and flat small, with different thicknesses and seals of different shapes.
7. Different bolt and nut configurations.
8. Different materials for identical applications.

These variations in flange design resulted in different performances, weights, and costs.

Certainly it is very difficult to avoid such problems during the development of a new project, particularly if different design organizations working with different manufacturing facilities provide the design without having identical specifications for the design, manufacture, and testing. However, if OR methods had been used to evaluate all separable connector designs on the

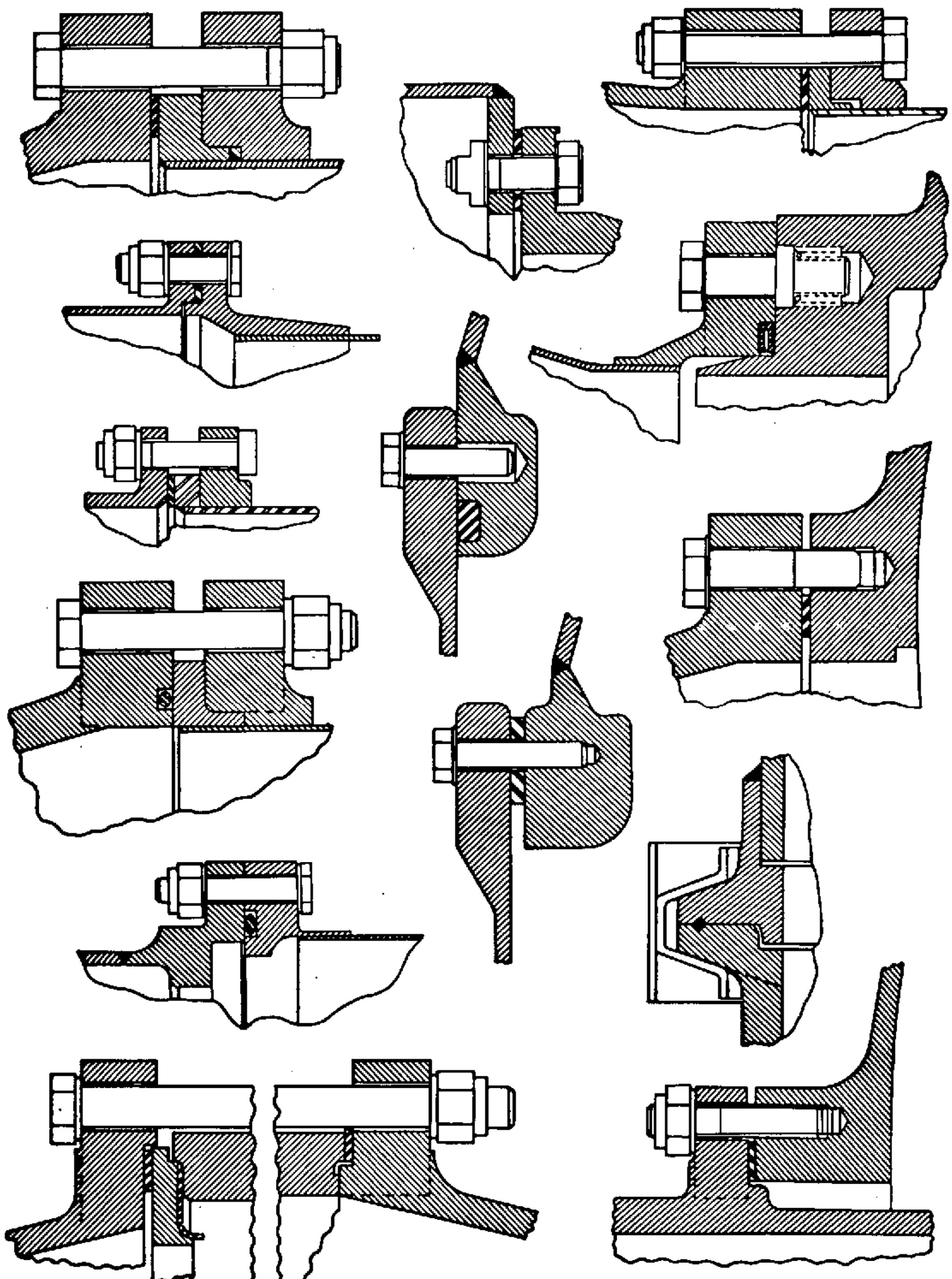


FIGURE 4. FLANGE CONFIGURATION SAMPLES OF THE LOX AND GOX SYSTEMS USED ON THE SATURN IB FIRST STAGE

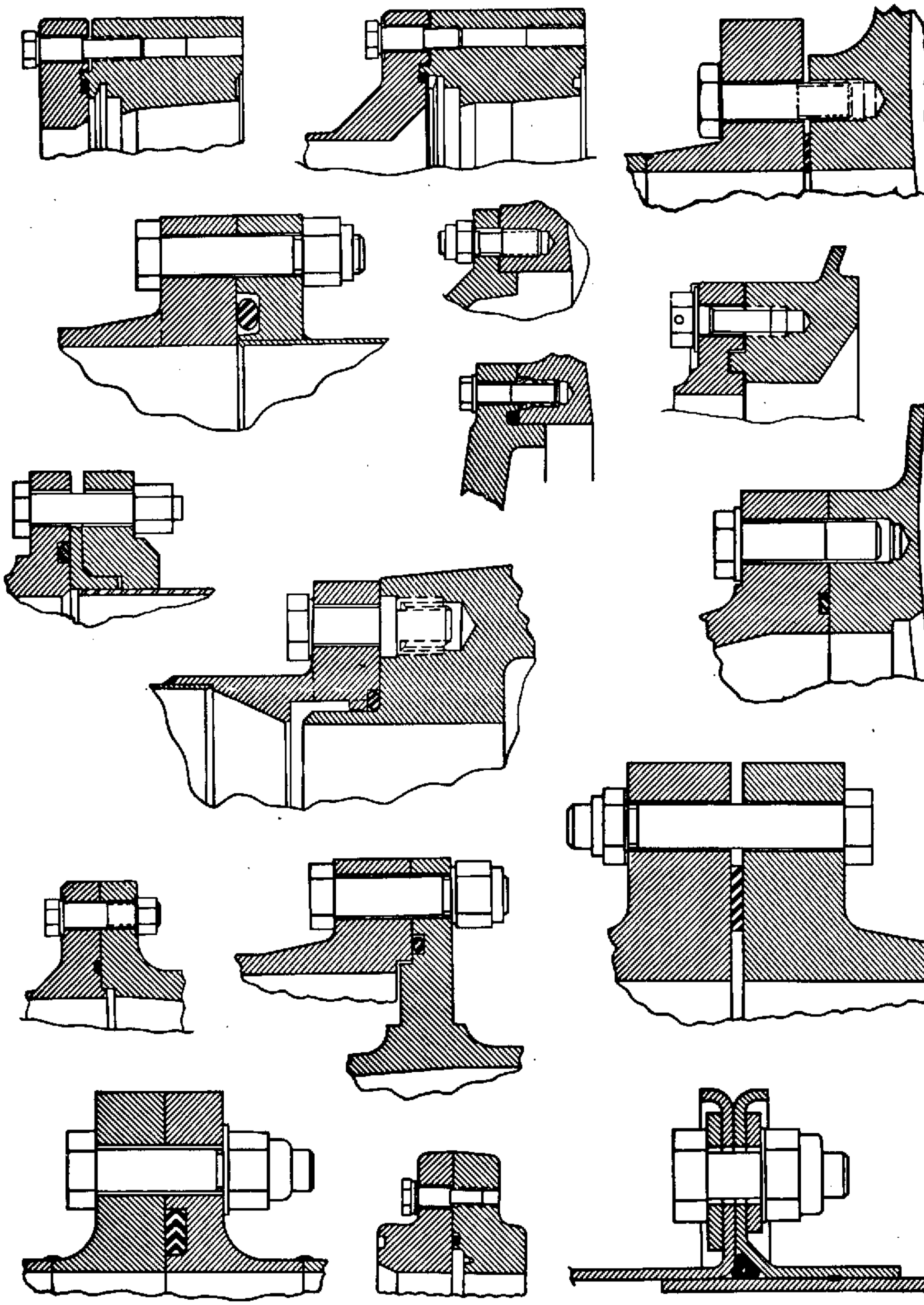


FIGURE 5. FLANGE CONFIGURATION SAMPLES OF THE FUEL AND ENGINE SYSTEMS USED ON THE SATURN IB FIRST STAGE

prototype for the flight stages, the result would have been (1) a higher performance level, (2) important weight savings, (3) fewer configurations, and consequently (4) lower costs.

The separable connectors for the second stage (S-IVB) and the payload of the Saturn IB pose the same problems as the evaluated connectors, but they are not evaluated in this research.

CHAPTER IV

FACTORS AFFECTING THE DESIGN OF SEPARABLE CONNECTORS

The goal of a separable connector design is to develop a lightweight connector without compromising its performance. For the design of connectors, a leakage rate of 1×10^{-4} cm^3/sec of helium per linear inch of seal was originally specified [2], but this resulted in excessive flange weights. This requirement was relaxed later to 1×10^{-3} cm^3/sec of helium per linear inch of seal.

The maintenance of a non-leak state during operation is a prime requirement for separable connectors. The interaction between the connector components, flange, gasket (seal), and bolts under flight conditions is of major importance.

A leak could develop during operation because of permeation, the porosity of the material, a reduction of the stress between the mated parts, some lateral shift between the sealing surfaces by shearing off previously mated asperities, and by inducing flange bending and deflection. A reduction of the sealing load may occur as a result of vibration, shock, internal pressure, thermal distortion, and misalignment of the flanges. A stress relaxation is always certain if the connector experiences elevated temperatures and pressures.

Calculation of the Leakage Flow Through a Connector

To meet the requirements for a reliable separable connector, it is advantageous to design it for the gaseous state of the medium for which it is used, which results in improved system performance.

The total amount of leakage flow through a connector is the sum of (1) flow resulting from permeation through the component material, (2) flow through the materials because of porosity, and (3) flow through the interface between mated components. The leakage resulting from permeation is considered for long duration performance in hard vacuum and varying temperatures and can be calculated from the following equation taken from Rathbun [2]:

$$Q = C_p \cdot A \frac{\Delta p}{L} \; (\mathrm{cm}^3/\mathrm{sec}) \quad ,$$

where

Q = amount of the medium at atmospheric pressure and temperature leaking through the connector ($\mathrm{cm}^3/\mathrm{sec}$),

C_p = permeation rate constant (cm^3 mm/kg · sec),

A = Flow area (cm^2),

L = flow path (mm),

and

Δp = pressure differential across the seal ($\mathrm{kg/cm}^2$).

For most metals and gases, $C_p \leq 10^{-8}$ cm^3 mm/kg · sec; for elastomers and plastics C_p ranges from 4×10^{-4} to 10^{-7} cm^3 mm/kg · sec, using helium gas. Leakage resulting from porosity is not considered a design parameter; it occurs rarely in welding areas, and can be easily detected by X-rays.

Modes of Leaks

Two modes of leaks are to be considered in designing a separable connector system: (1) the liquid interface leakage flow, and (2) the gaseous interface leakage flow, which can be distinguished either as viscous flow or molecular flow or both. If the medium is a liquid, it is called viscous flow. If the medium is a liquid but vaporized because of temperature variation, or if the medium is a gas, the flow is molecular. Molecular flow exists for extremely low flow rates in the range of $Q = 10^{-6}$ cm^3/sec or less at atmospheric pressure. If the smallest leak path dimension becomes large in comparison to the mean free path of the gas in question, viscous flow commences. At the beginning the flow will be laminar; later, depending on the pressure differential, the flow might become turbulent.

The most serious leakage is that which occurs because of improper mating between the sealing surfaces. Obtaining "zero leakage" for long duration space application is almost impossible because permeation and diffusion are always present.

The Viscous Flow of Gas

The flow rate of a gas leaking radially between two flat annular plates for laminar flow is given by Rathbun [2] as follows:

$$Q = 13.79 \times 10^{-11} \frac{a t_c^{\,3} (\Delta p) \bar{p}}{\mu\omega} \quad 1 + 6.383\, \epsilon \frac{\bar{\lambda}}{t_c} \quad ,$$

where

Q = leakage rate at atmospheric pressure (cm^3/sec),

a = mean perimeter of annulus (in.),

t_c = clearance between flanges (μin.),

Δp = pressure difference across seal (kg/cm^2),

$\bar{p}$ = mean pressure $(p_{int} + p_{ext})/2(kg/cm^2)$,

μ = absolute viscosity of gas (centipoise),

ω = flange (plate) radial width (in.),

ϵ = molecular correction factor, dimensionless (0.9 for a single gas and 0.66 for a mixed gas such as air),

and

$\bar{\lambda}$ = mean free path of the gas molecules at the mean pressure $\bar{p}$ (μin.).

Bauer [3] developed an equation to calculate the leakage Q of a typical pressure-energized seal as follows:

$$Q = \frac{\Pi (p_{ext}^{\;2} - p_{int}^{\;2})}{24\, p_0 \mu} \; \frac{r_i + b + r}{r_i + b - r_i} \; (h^3) \quad ,$$

where

Q = leakage rate (in.^{3}/sec),

μ = viscosity of medium (lb-sec/in.^{2}),

r_o = outer seal radius (in.),

r_i = inner seal radius (in.),

p_{int} = internal fluid pressure (psi),

p_{ext} = external fluid pressure (psi),

p_0 = standard atmospheric pressure (psi),

b = seal contact width (in.),

and

h^3 = conductance parameter (in.^{3}); its value depends on the flange face roughness.

The above equation is used to calculate the leak rate for K-seals (K-shaped seals) and similar configurations.

The Selection of Materials

The selection of materials for the components of a separable connector system is important to the performance of the system. There is a great variety of materials on the market, and making a selection is rather confusing. The first parameters to be considered are compatibility, and the stress-weight ratio of the materials. A material is said to be compatible if there is no corrosive attack by the medium it is in contact with and if there is no decomposition of the medium caused by the material.

Candidate materials for the separable connector systems must have, over the anticipated service time and environment, the following properties:

1. Good creep resistance.
2. Rupture and relaxation strength over good resistance to thermal shock.
3. Oxidation.
4. Embrittlement during cyclic service.
5. Ease of machining, welding, and molding (plastics).

Because of the interaction between the flanges, the gasket, and the bolts that determine the seal pressure, the properties of the materials utilized must be compatible with each other. The thermal expansion coefficients, for example, have to be compatible to avoid seal pressure relaxation during operation.

In addition to the selection of materials for gaskets and seals, the following factors should also be considered:

1. Short torquing sequence for the bolts.
2. Little or no cold flow.
3. Little compression setting.
4. Retention, to some degree, of flexibility at extreme temperatures.

A selection of the most commonly used construction materials for different propellants is shown in Table V.

The Flange

The design goal of a separable connector is to develop a lightweight system without compromising its performance. To design a separable

TABLE V

CONSTRUCTION MATERIALS FOR DIFFERENT PROPELLANTS

Medium	Metallic	Nonmetallic
Alcohol	Steel, Stainless Steel, Aluminum	Neoprene, Rubber, Kel-F, Teflon, Asbestos
Fluorine Gas	Nickel, Monel, Stainless Steel (300 Series), Aluminum, Titanium, Low Carbon Steel	Teflon, Fel-F
Fluorine Liquid	Monel, Stainless Steel (300 Series) (Not 347)	
Hydrazine	Stainless Steel (300 Series)	Teflon, Kel-F
Hydrocarbon Fuel	Steel	Teflon, Kel-F Buna-N, Vinyls
Liquid Hydrogen	Most Aluminum Alloys, Stainless Steel, Monel, Nickel Alloys	
Hydrogen Peroxide	Aluminum Alloys, Stainless Steel (300 Series)	Teflon
Fuming Nitric Acid	Aluminum Alloys, Stainless Steel (300 Series)	Kel-F, Teflon
Liquid Oxygen	Aluminum Alloys, Stainless Steel (300 Series) Monel, Nickel Alloys	Allpax Narmco Teflon-Coated K-Seal

connector, five major steps have to be taken to meet the requirements:

1. Identification of the medium, operational pressure, temperature, and environment.
2. Determination of the tolerable leakage rate.
3. The selection of the appropriate material.
4. Determination of the flange configuration.
5. Design of the necessary support for the connector system to force the sealing surfaces together and avoid lateral flange movement.

In addition to the identification of the operational condition to determine the total load distribution, it is necessary to consider the pipe forces including hydrostatic and dynamic pressure loads, axial and bending loads, and loads resulting from shock and the water hammer effect. Storage life, operational life, checkout, static testing, and reusability are parameters that will influence the design of the connector.

Flange Configurations

Flanges can be divided into the following groups:

1. Integral flanges with no contact and with contact outside the gasket.
2. Loose flanges with no contact and with contact outside the bolt circle.
3. Taylor Forge and similar flanges.
4. Low profile flanges.
5. Optional flanges.

Some of these configurations were shown previously in Figures 4 and 5 and some others are shown in Figure 6.

Six different flange facings for the adaptation of different gaskets and seals are shown in Figure 7. The male-and-female configuration centers the gasket and avoids lateral motion of the flanges. The raised-face design is used for flanges whose sealing effectiveness is provided by the bolt and load only. Flat-face configurations are used to adapt wider gaskets. The lap-joint flange is used for connectors where the pressure is low and welding is difficult. Tongue-and-groove flanges are built for gaskets that demand full confinement. Groove flanges are built for the adaptation of self-equalizing seals of different shapes.

Surface Finish

The surface finish of a flange influences the potential leak to a noticeable extent. Given that a flange surface has some asperity distributions on it and that each distribution varies in magnitude, direction, and type, the leak path is impossible to predict. These asperities will be deformed depending on the yield stress and the strain hardening characteristics of the material. Under load, the gasket material penetrates the asperities and fills the voids. By increasing the load a plastic deformation of the asperities occurs as they mate with the opposite asperities. The equilibrium stress level during deformation is approximately two to three times the yield stress. When the stress field in the solid is sufficient to produce yielding in a large portion of the material, geometrical deformation will occur.

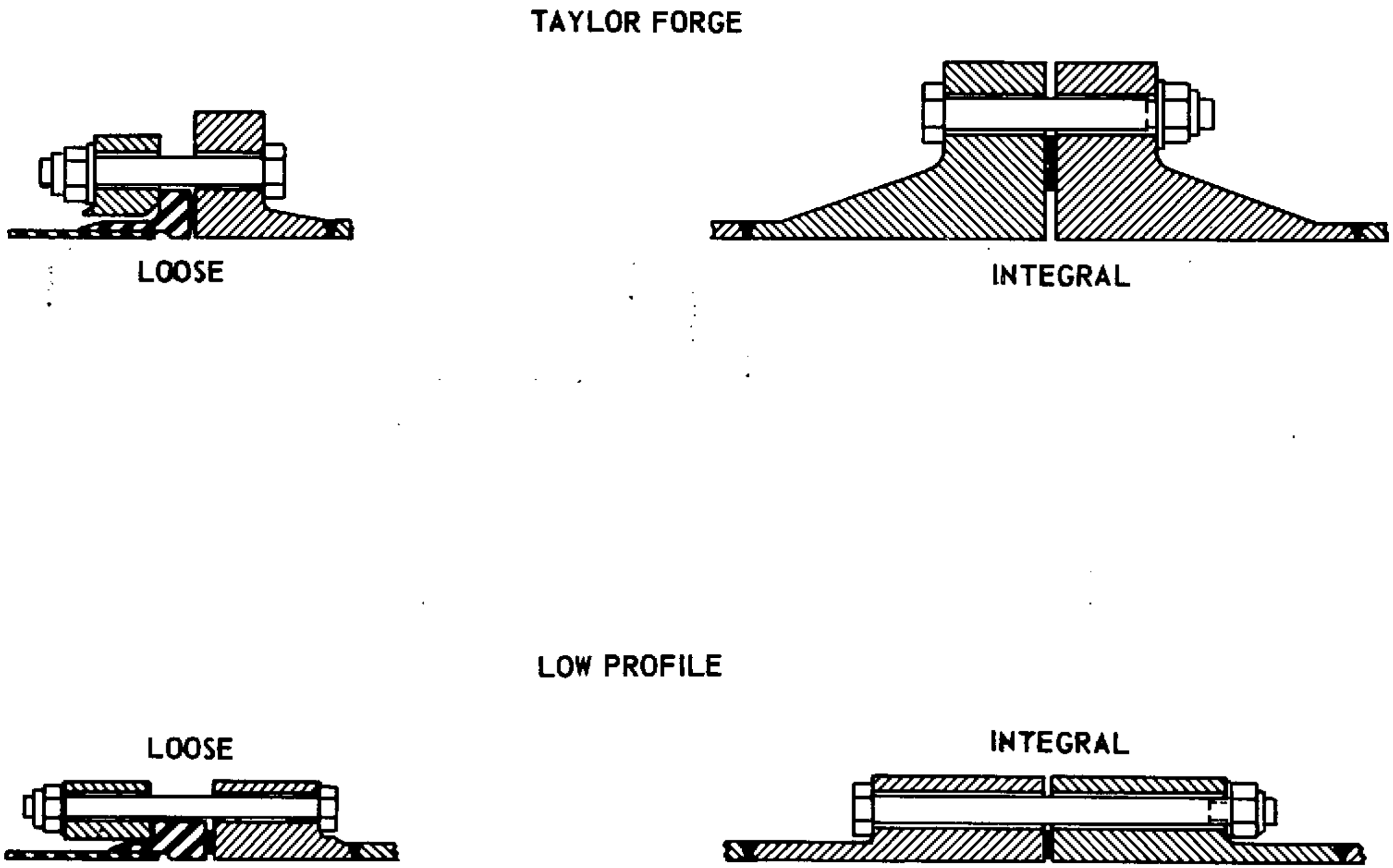


FIGURE 6. LOW PROFILE AND TAYLOR FORGE LIGHTWEIGHT FLANGE CONFIGURATIONS

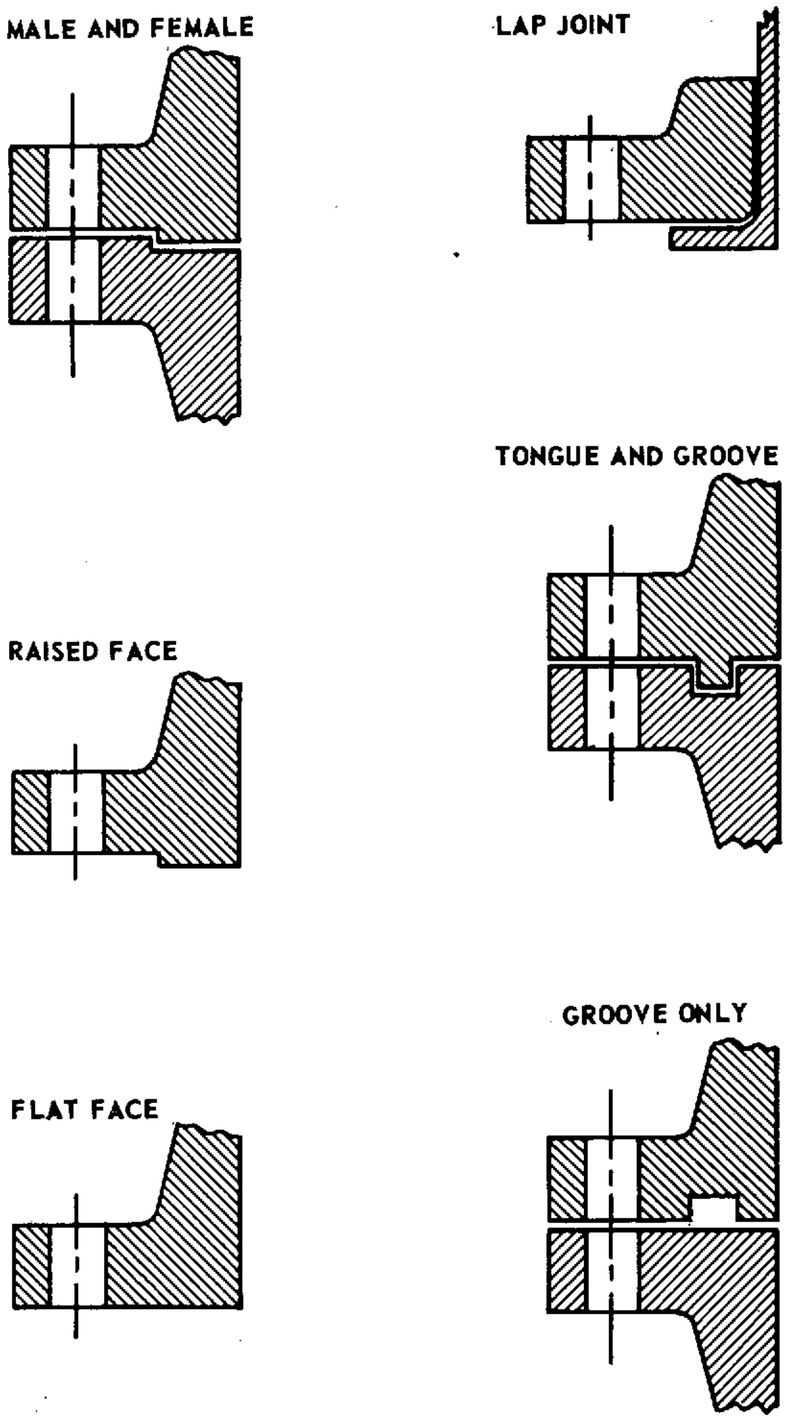


FIGURE 7. BASIC TYPES OF FLANGE FACINGS

Surface shear stresses tend to reduce the bulk flow during this regime, but the deformable gasket material will slide along the surface with which it is mated, assuring a positive seal. Figure 8, Broadstone [4], shows

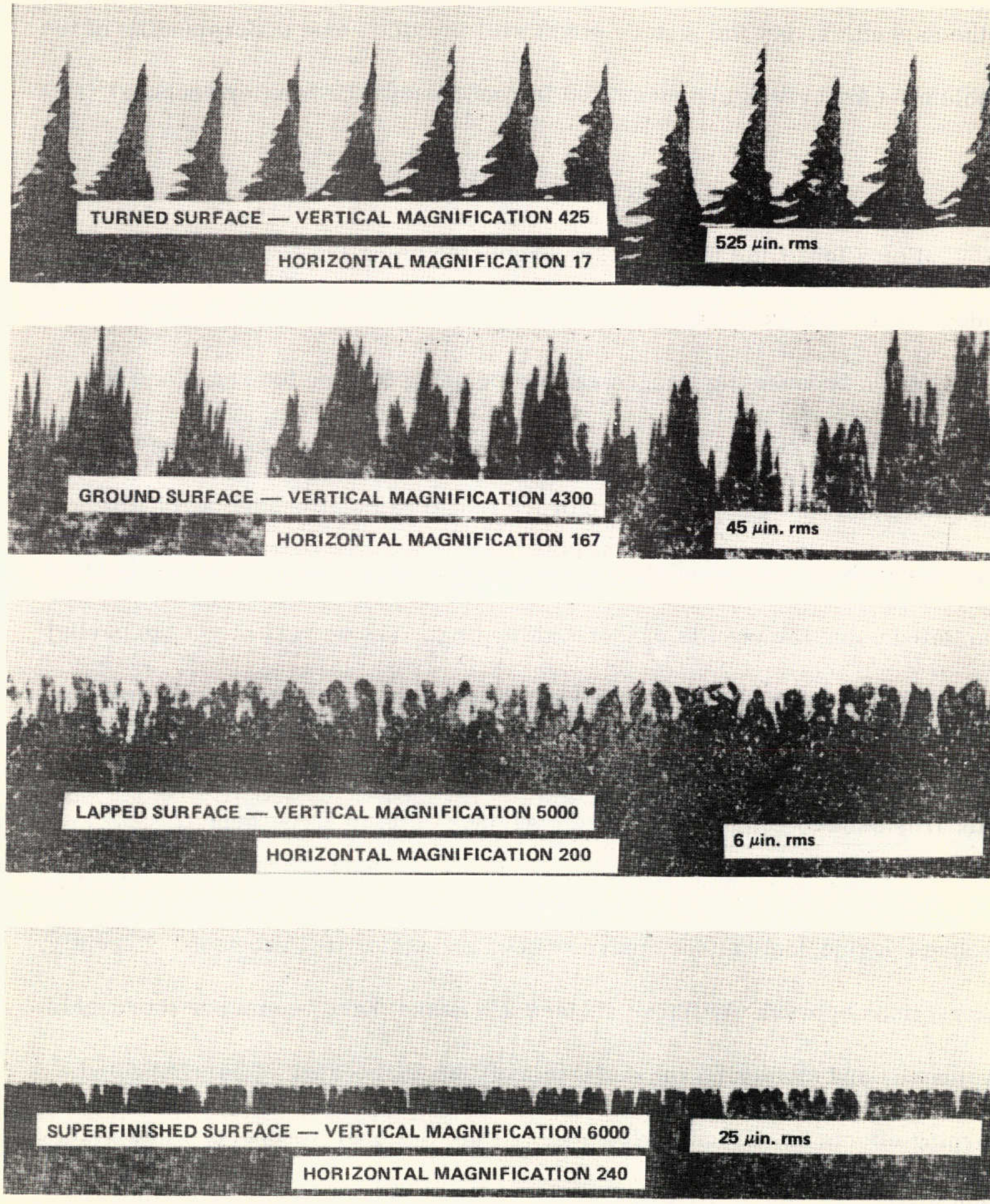


FIGURE 8. FLANGE SURFACE FINISH

differently machined surfaces. Unfortunately, the magnification of each surface picture is different, which misleads the reader in judging the magnitude of the asperities, but the picture might still provide some understanding of the problem. (Asperities are measured in root-mean-square microinches.)

Calculation of Flanges

For a long time flanges were calculated using the A.S.M.E. Pressure Code [5] as a guide, but for aerospace applications this calculation was unacceptable because the flanges calculated using it were too heavy and too bulky. A strong effort was made by the Government space-related agencies and among the space industry to improve flange calculations by developing new standards. The formulas, algorithms, and tables were empirically substantiated using the results of thorough testing. The acceptance of the partial plastic deformation theory, applied for the application of flanges, formed the base of the new lightweight flange design. Flanges designed to these new standards showed comparative weight savings of up to 25 percent.

A great number of newly developed flange calculations were performed by stage engine contractors, universities, and other organizations. Recently, interactions between the flange surface and gasket have been more thoroughly evaluated, and studies on the deflection of flanges because of barreling and warping effects have been performed. At the National Aeronautics and Space Administration, Marshall Space Flight Center, five computer programs are available for structural analysis. One of these, the "Separable Connector Design Handbook" [2], prepared for NASA-MSFC by General Electric was used for the calculation of some flanges under investigation in this study.

Since the deflection of a flange is a main contributor to the development of a leak, some deflection theories were studied, and for calculation purposes the following simplified equation from Field [6] was used:

$$\delta = \frac{3p_i \cdot k}{E_f} \frac{e}{t^3} \quad ,$$

where

δ = flange deflection (in.),

p_i = internal pressure (psi),

E_f = modulus of flange material,

t = flange thickness (in.),

e = bolt offset = $r_B - r_D$ (in.),

r_B = bolt circle radius (in.),

r_D = inner duct radius (in.),

and

$$k = \frac{t^2 \cdot r_D + 2t \cdot r_D}{\log_e \dfrac{t + t_D}{r_D}} \quad .$$

This equation is good for a first analysis of a flange.

Gaskets and Seals

Although gaskets and seals for separable connectors have been widely used for many years, only in the past few years has an attempt been made by manufacturers and users to establish standards. When the design of space hardware began, almost every stage and engine builder used different criteria to select gaskets and/or seals. Approximately 80 different gasket and seal configurations are commercially manufactured. The performance ratings for these gaskets and seals are based on test and flight evaluation data. The engineer's selection of one gasket from many existing gaskets is difficult, and most of the time the selection depends on data provided by the manufacturer or users and test experience. In case of doubt, only testing and performance verification will establish the required confidence level.

There are two ways of sealing a connector, one is with a gasket and the other is with a seal. Figure 9 shows some of the normally used gasket and seal configurations. When a gasket is used, the pressure required to seal the gasket flange interface is provided by the connector bolt load. If a seal is used, the sealing effect depends on the seal structure and on the seal interface. The seal structure directly influences the forces acting on the interfaces; thus, some of the sealing pressure comes from the pressure of the confined fluid. An elastomer O-ring or a K-seal, for example, is first compressed by the bolt force of the connector and is then further compressed into flange surface irregularities with the help of the fluid pressure.

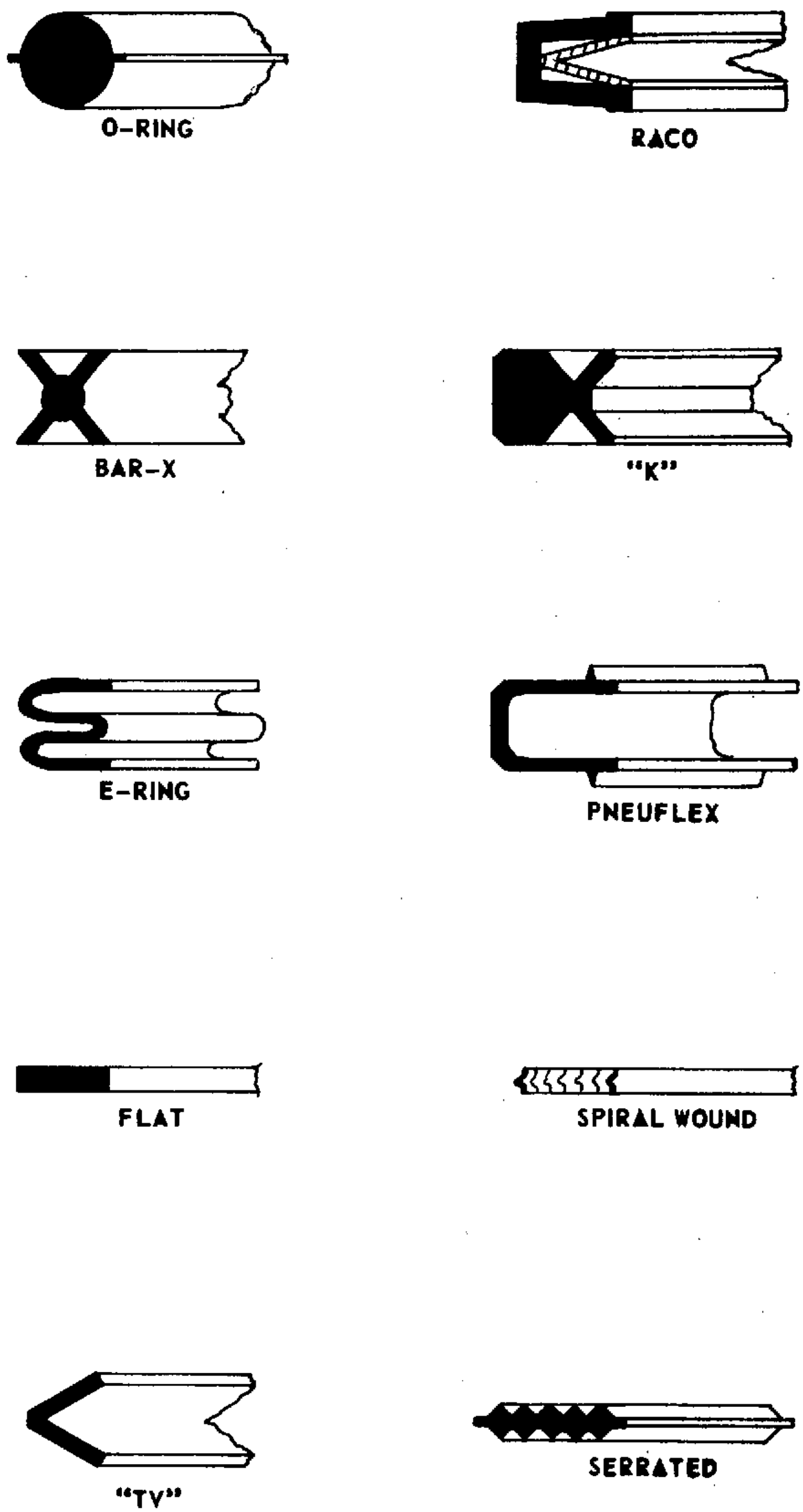


FIGURE 9. GASKET AND SEAL CONFIGURATIONS

The seal interface geometry, surface roughness, and the seal load determine the magnitude of the leak. Loading and contact stress of the sealing elements are the important parameter of the seal performance. It is of primary importance that the seal deflection capability is such that it compensates the radial and axial motion of the flange within the permissible tolerances. The total deflection capability of a gasket depends largely on the gasket height and the seal modulus. At the same time the gasket must be as thin as possible to reduce gasket extrusion and to minimize torque losses in the bolts. The sealing integrity also depends on the conformable material and the sealing pressure that forces it into intimate contact with the surface irregularities and asperities.

Softer gaskets normally permit a greater angular rotation of the flange under applied internal pressure and bolt load than stiffer gaskets. A wider gasket will restrain the flange rotation more than a smaller gasket. Differences in the form and material of a gasket will result in different stress distributions in the flanges.

The gasket forces for zero leakage must be obtained empirically. The gasket seal force is affected by (1) the bolt load, (2) the gasket (seal) material, (3) the surface finish of the flange, and (4) the flange face. Flat gaskets require a stress of about 2.75 times the yield stress to achieve zero leakage. From that point the gasket force can be reduced to approximately one-third before the zero-leakage state is lost. The initial gasket load must be large enough so that the application of pressure and pipe loads will not reduce it below the limiting minimum value as shown by Field [6].

A seal is normally compressed first with the help of the sealing force from the bolts and flanges and then further compressed into the irregularities by the fluid pressure. For high pressure systems, such as the engine application, seals are preferred over metal gaskets. A leak between the seal and flanges can only be prevented by providing good contact between the interfaces, which depends on the flange surface finish, the developed seal stress, and its modules.

Calculation of Gaskets and Seals

The stress S in a gasket can be calculated using Field's equation [6]:

$$S = \frac{F}{A_g} \quad \frac{F}{2r\pi w_s} = \frac{E_g x}{h_g} \quad ,$$

where

S = seal stress (lb/in.^{2}),

A_g = $2r\pi w_s$ = gasket bearing surface (in.^{2}),

F = flange force (lb),

w_s = gasket width (in.),

x = gasket distance compressed after F is applied (in.),

E_g = modulus of gasket material,

E_y = modulus at yield stress,

and

h_g = gasket height (in.).

If the gasket is compressed to its yield point, then the seal stress at this point will be

$$S_y = \frac{E_y \cdot x}{h_g} ,$$

and the distance compressed is

$$x = \frac{S_y \cdot h_g}{E_y} ,$$

which defines the total deflection capability of the gasket. This means that any material change, or a change in gasket height (thickness) h_g , will change the deflection capability of the gasket.

To calculate the sealing efficiency of the gasket, the degree of contact between the gasket (seal) and the flange must be known. The degree of contact achieved depends on the finish of the flange surface, the magnitude of the stress developed in the gasket, and the gasket material. The degree of contact D_c is defined as the ratio of the distance x that the gasket is compressed to the distance d_y that the gasket material must move to fill and seal the flange surface asperities. Using the value of x , we obtain

$$D_c = \frac{x}{d_y} = \frac{Sh_g}{E_y d_y} .$$

For calculation purposes d_y can be four to five times the asperities height measured in root-mean-square microinches.

The minimum stress S required to establish good contact is calculated by taking $x = d_y$ and $D_c = 1$, so that

$$S_{min} = \frac{E_y d_y}{h_g} ,$$

where

S_{min} = minimum seal stress (lb/in.^{2}),

D_c = degree of seal contact,

and

x = change in gasket height because of compression (in.).

Standard practice in gasket calculation is to assume that the seal penetration height can be four times the flange asperity average height. Normally the energy stored in a seal allows it to follow and maintain good contact as long as the joint does not separate more than $x-d_y$, which is the usable deflection capability of the seal.

The Bolt

The bolts of a separable connector provide the required sealing load on the gasket located between the two flanges. A problem is that the tensile load applied on the bolt cannot be measured exactly during torquing because the friction between the bolt and nut and the nut and washer cannot be isolated from the applied torque load. This results in different gasket loads which consequently results in leakage. Built-in feeler gauges are used in the aircraft industry to control the torque applied to the bolt. Unfortunately, these gauges are expensive and require special handling.

The rigidity of the bolt flange system must be less than the rigidity of the seal system to ensure that the bolt stress is relatively unaffected by the vibration forces and pressure changes.

A greater number of bolts present a more uniform stress distribution at the flange-gasket interface. An optimized flange design features the use of the smallest possible bolt size, which results in very close spacing between bolts because of the great number of them being used. Very accurate methods of determining bolt loads were published by Horsch [7] and McLure [8]. The influence of the bolt ultimate tensile strength (UTS) in saving flange weight is demonstrated in Figure 10. Although these values are purely theoretical and are calculated for stainless steel, for high pressures from 4,500 to 7,000 psi, they certainly demonstrate that a potential weight saving exists by increasing the bolt tensile loads.

The three major bolt configurations used for separable bolted connectors are (1) hex-head bolts, (2) internal wrenching bolts, and (3) 12-point external wrenching bolts.

The bolt material used for separable connectors has to be compatible with the flange material and the fluid medium. To avoid stress relaxation

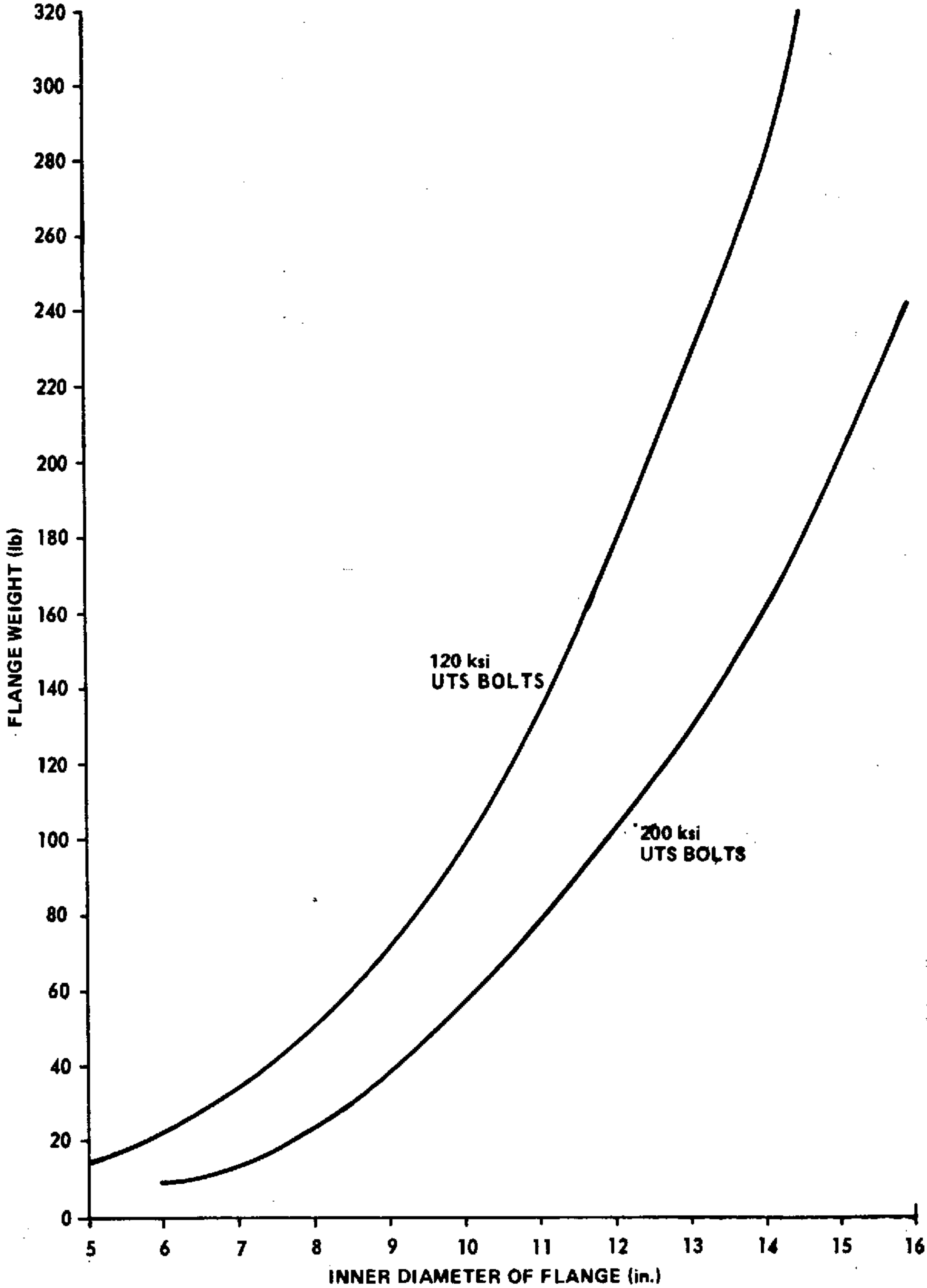


FIGURE 10. FLANGE WEIGHT VERSUS BOLT STRENGTH

caused by the operational environment, particularly in the high and low temperature ranges, the thermal expansion coefficient of the material is very important.

The common trade-offs in selecting the right bolts for a connector design are (1) the number of bolts versus allowable waviness of the flange and (2) the thickness and length of the bolts.

The evaluation of nut and washer designs was considered negligible for this research, but in the design of a connector system, it must be given fair consideration.

CHAPTER V

FORMULATION OF THE PROBLEM

Given a design objective a number of possible ways to achieve this goal usually become apparent. Each of the ways is an alternate strategy. Variants encountered in these strategies could include (1) differences in the material used, (2) tolerance, (3) shape, (4) form, and (5) stress-weight ratio. The decision problem is to choose that strategy which best satisfies the design objectives. For any complex design, the number of alternatives (strategies) expands beyond our comprehension, as previously shown by the design tree (Fig. 2). The decision to be made may be composed of several different decision problems where the individual decisions are either clearly independent from each other or are interacting with each other.

Types of Parameters

Parameters describe the characteristics and properties of components and systems. Parameters are important factors in determining the quantitative and qualitative values that are needed for the analysis and determination of system performance, weight, cost, profit, etc. In general, parameters are divided into:

1. Objective parameters that encompass operational capabilities.

2. Subjective parameters not directly related to operational properties, encompassing technical risk, growth potential, etc.

In addition, parameters can be categorized as follows:

1. Additive parameters.
2. Probabilistic parameters.
3. Nonadditive parameters.

If the system level parameter value is the sum or difference of the subsystem parameter values, then the parameter is considered to be an additive parameter. In the case where the system level parameter consists of the product of the subsystem values, the parameter is probabilistic. Nonadditive parameters are characterized by a maximal or minimal property, which means that the system parameter value is the maximum or minimum of the subsystem parameter values. For example, an assembly schedule is a nonadditive parameter; a system cannot be assembled until after the subsystem requiring the longest order time or longest manufacture time is completed. Parameters such as performance level, growth potential, and technical risk also are nonadditive parameters.

In selecting parameters for the design of components and systems, performance is normally given first consideration. If, for example, a connector of a vital supply system, such as the oxygen system, begins leaking excessively during the mission, the value of the remaining parameters, such as weight or cost, would be immaterial because the mission would be totally or partially lost.

To a certain extent, all parameters have theoretical or practical limits. Some parameters have a minimal level below which the system cannot perform, and others have upper limits or constraints imposed by practical considerations influencing the performance. Parameter values of this nature are important in determining the range in which satisfactory performance can be expected. A separable bolted connector configuration is limited by the state-of-the-art of the design, material, and manufacturing techniques. Available fundings determine the cost of the development, while the weight is limited by the boost capability of the vehicle as well as by the payload limitations. To achieve maximum effectiveness, trade-off among the system parameters within the limiting values is frequently accomplished until an optimum is reached. Trade-offs for optimization purposes could include:

1. Performance versus weight and cost.
2. Reliability versus weight.
3. Reliability versus cost.

In this study the following parameters are used to determine the optimization of the connector system:

1. The performance of the connector.
2. The weight of the connector.
3. The cost of the connector.

The performance parameter is a measure of the capability of the connector system to maintain its sealing characteristics which in turn depends on:

1. The distortion of the flange because of hoop stresses and meridional flange rotation.
2. The gasket or seal configuration and properties.
3. The bolt characteristics.
4. The interactions among flanges, gasket, and bolts.

The effectiveness (performance) E of any system normally is equal to some relationship between a set of controlled variables of the system X_i and a set of uncontrolled variables Y_i. The basic form of most OR models is

$$E = f(X_i, Y_i) \quad .$$

Restrictions on the values of the variables may be expressed in a supplementary set of equations.

The model can express symbolically a pattern of very complex interrelationships that would be difficult to explain completely in words. The model also can be manipulated to simulate changes to the system and to predict the effect of the changes on the system without tampering with the actual operations of the system in use.

Measure of System Effectiveness

A very difficult effort for an optimization model is the development of an adequate measure of the system performance or system effectiveness. The problem is to determine what is an optimum separable connector. Is it a lightweight system using exotic materials? Is it a system with a wide flange or a small one? Is it a system with a pressure equalizing seal? Is it a combination of some or all of these considerations? The measure of the system performance must reflect the relative importance among the multiplicity of objectives involved.

The simplest possible types of system measures are given by their effectiveness, weight, and cost, where the system effectiveness can be defined as a measure of the extent to which it achieves a set of specific requirements or objectives. How well the connector system performs within its tolerances is its effectiveness. Usually, but not necessarily, its measure is given by a calculated value denoting the probability that the system objectives are met. Unfortunately the evaluation of a system usually involves more measure than the consideration of effectiveness and cost; reliability, maintainability, availability, and reusability would be appropriate additional measurements in this study, which make the whole process of system-measure even more complicated.

Parameter Optimization Methods

There are three widely utilized methods of optimizing a parameter subject to constraints on other parameters:

1. The Search Method.
2. The Lagrange Multipliers method.
3. Dynamic Programming method.

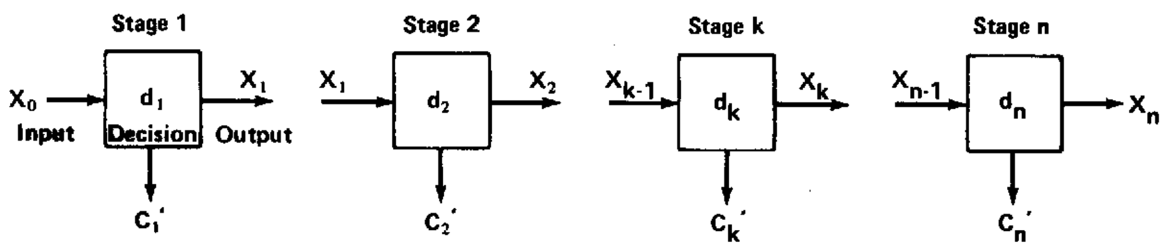
The Search Method consists of examining all combinations of the system and subsystems, and determining from those systems and subsystems that parametric value which is the most desirable. This method is used in small problems where the constraint values are specified. It is not suitable in system level parametric relationships because for each constraint value change, another search would have to be made.

The Lagrange Multipliers technique is an application of classical calculus. As in the Search Method, the individual solutions of specific values correspond to specific parameter constraint values; therefore, another solution would have to be developed for each combination of multiplier values during development of system level parameters.

Dynamic Programming (DP), developed by Bellman [1], is a mathematical technique useful for making a sequence of interrelated decisions. DP provides a systematic procedure for determining the combinations of decisions that maximize the overall effectiveness. There is no standard mathematical formulation of the DP problem solving technique. Particular equations must be developed to fit each individual situation. The characteristics of DP are:

1. The problem can be divided into stages with a policy decision required at each stage.
2. Each stage has a number of stages associated with it, which are the various possible conditions in which the system might find itself at that stage of the problem; the number of stages might be finite or infinite.
3. The effect of the policy decision at each stage is to transform the selected state into a state associated with the next stage.

To apply DP, the design system will be divided into a series of subsystems where each subsystem is characterized by (1) some input, (2) a decision that must be made, (3) some output, and (4) a criterion function. The criterion function C_k', also called the utility function for any stage or subsystem, measures the contribution of that particular stage to the overall system effectiveness, whether it be performance, cost, weight, or some other parameter. To form an n-stage system, the subsystems or stages are connected in series, head-to-tail, with no recycle. The schematic of an n stage problem is shown below.



The output of stage k-1, which is X_{k-1}, is also the input into stage k and the output X_k of stage k is the input for stage k + 1. Both the criterion function C_k' and the output X_k are functions of the input X_{k-1} and the decision d_k made at the stage.

$$C_k' = f(X_{k-1}, d_k)$$

and

$$X_k = h(X_{k-1}, d_k) \quad .$$

An example of such a problem is a separable connector system consisting of three components or subsystems, the flange, the gasket, and the bolts, and where at each stage one of several configurations may be used. Performance could be the criterion that the design engineer wishes to optimize. In this case, performance is the value of the stage criterion function C_k'. The stages are jointed because a decision made at one stage affects the criterion at succeeding stages. Considering all stages where each decision, d_k, involved choosing one configuration from m_c possible configurations, the number of alternative designs for the entire system is

$$\prod_{k=1}^{n} m_k = m_1 m_2 \ldots m_n \quad .$$

This problem is therefore combinatorial in nature. To completely enumerate all possible combinations and choose that configuration which optimizes the system's criterion function, would be, even with a computer, a time consuming task.

Dynamic programming provides a method of solving the problem in a relatively easy and efficient manner. The principle of discrete stage optimization has the property that, regardless of the previous decisions, the remaining decision must constitute an optimum sequence of decisions for the remaining problem. This principle permits one to solve multistage problems by working backward. Considering the end stage n, the optimal value of the criterion performance C_n when the input value is X_{n-1} can be defined as

$$C_n \text{ opt } (X_{n-1}) = \min f_n (X_{n-1}, d_n) \text{ or } = \max f_n (X_{n-1}, d_n) \quad .$$

Working backward to stage n-1 , the best decision d_{n-1} has to be determined for the input X_{n-2}. The best decision will be the one that optimizes the sum of the two stage returns, which are

$$C_{n-1} \text{ opt } (X_{n-2}) + C_n \text{ opt } (X_{n-1}) \quad ,$$

where X_{n-1} is a function of X_{n-2} and d_{n-1}. Determining the best (optimum) decision at each stage for all possible inputs can be continued until stage k is reached. The optimal return for stages k through n would be

$$S_k{}^* (X_{k-1}) = \sum_{m=k}^{n} C_m{}^* (X_{m-1}) = C_k{}^* (X_{k-1}) + S_{k+1}{}^* (X_k) \quad ,$$

where X_k is defined as

$$X_k = k (X_{k-1}, d_k)$$

and the asterisk (*) symbolizes the optimum. The above equations provide the recursive relationship necessary to link the stages together. Simply, at each stage, determine the optimal decision for each possible input; when stage 1 is reached, trace back through the process in a forward pass, making the optimal decision at each stage, thereby optimizing the system.

The objective of this technicque is to reduce the amount of effort required for the solution of this n dimensional problem. The numerical example shown in Chapter VII will enable a better understanding of the procedure. The proposed procedure should not be considered the only approach to optimize separable connector systems, but merely as one that will aid in future design optimization problems. To perform such tasks, it is necessary for the investigators to be cognizant of operational research methods, techniques, and algorithms. A knowledge of the technical problems and their solution is also necessary.

CHAPTER VI

THE RESEARCH PROBLEM

Collection of Data

To perform this research, a considerable effort was necessary to provide and develop the input data. It was first necessary to determine the causes of leakage. Secondly, the standards and calculation methods used for the design had to be analyzed. In support of the assessment of parameter values, dimensions, material properties, costs, and weights were taken from:

1. Existing flange drawings and parts lists.
2. Technical reports, catalogues, and brochures.
3. Experts in the field of concern.

To establish basic data necessary for separable connector evaluation and comparison purposes, a Taylor Forge lightweight flange (Fig. 11) and a low profile flange designed for the LOX system (Fig. 12) were structurally analyzed using the computer program for flanged connector design [2]. The two flange stress computer programs were performed to determine flange stresses in five equally spaced locations in the flange hub center beginning at the flange surface and ending at the interface of the flange hub and pipe. The different stress values are given as stress ratios which is the quotient of the computed actual stress over the allowable stress. The stress analysis was

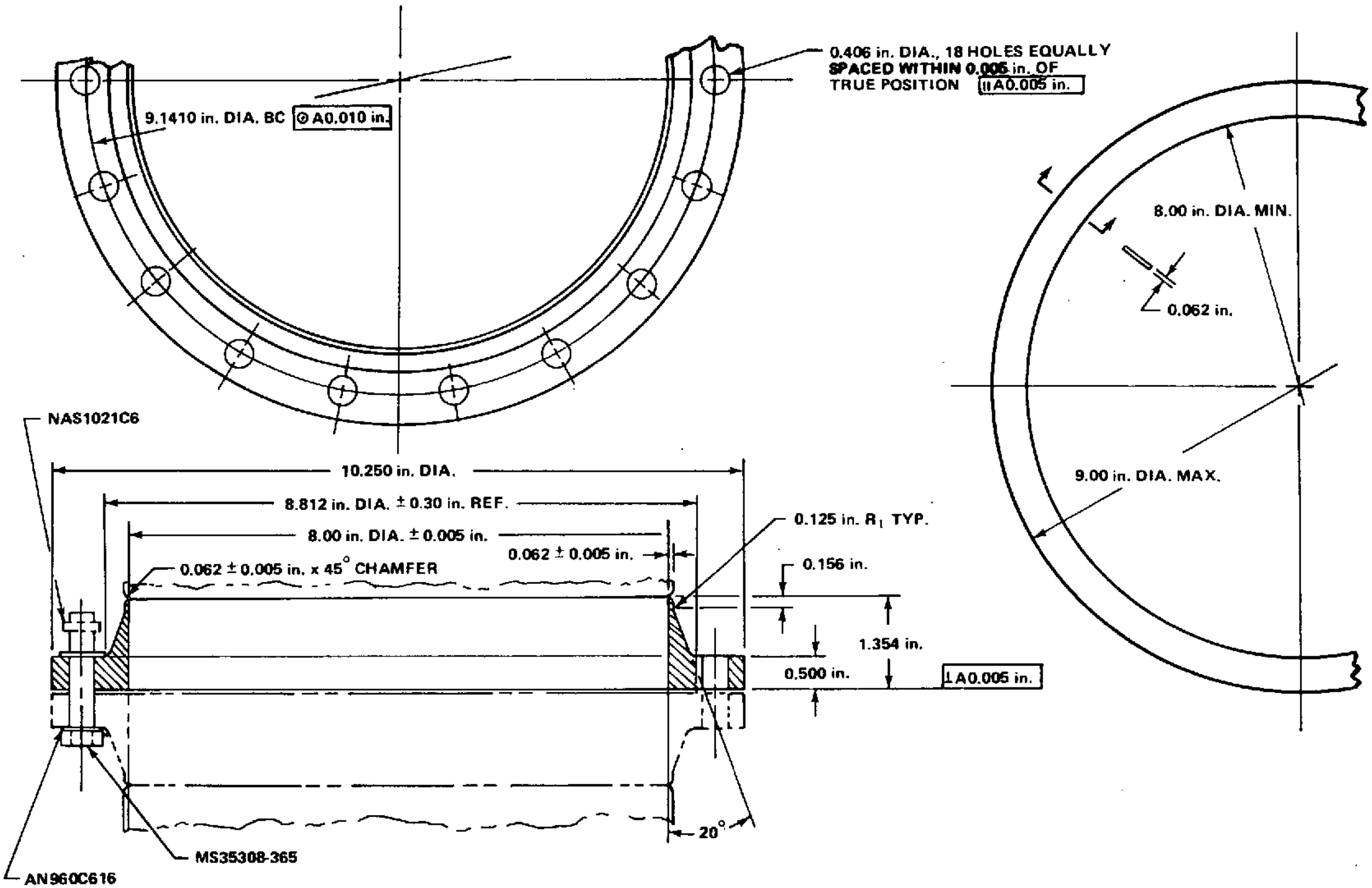


FIGURE 11. TAYLOR FORGE LIGHTWEIGHT FLANGE

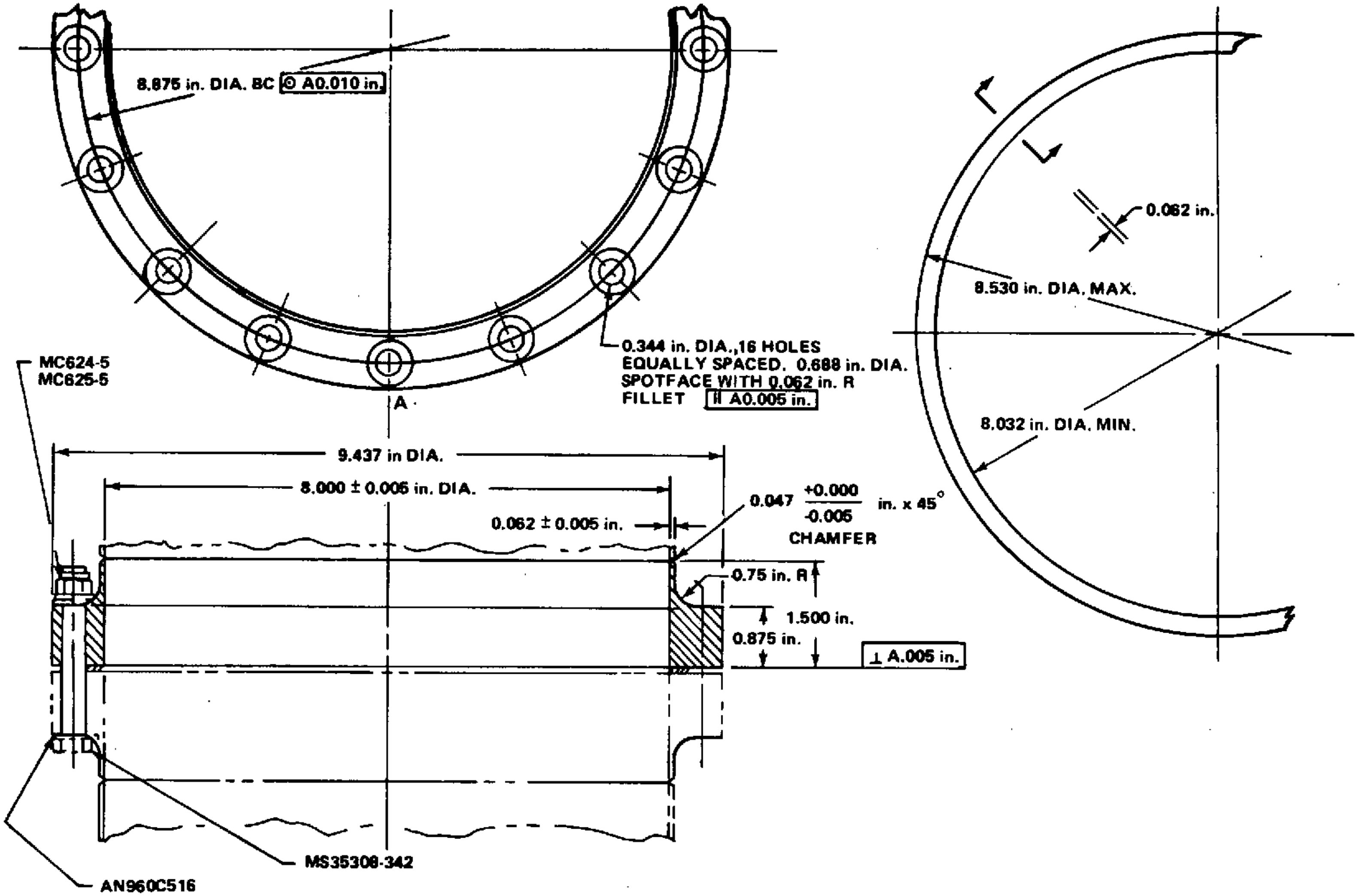


FIGURE 12. LOW PROFILE FLANGE

computed for five different pressure and temperature conditions:

1. Condition 1, atmospheric pressure at 70°F.
2. Condition 2, 200-psi pressure at 70°F.
3. Condition 3, 300-psi pressure at 70°F.
4. Condition 4, 200-psi pressure at -320°F.
5. Condition 5, 200-psi pressure at -450°F.

The two flange configurations, the Taylor Forge lightweight flange and the low profile flange, were structurally analyzed using an Allpax 500 gasket, a Butyl gasket and a steel gasket. The results of these stress analyses are shown in the appendix.

All Saturn IB first stage connector dimensions and data, from an inner diameter of 3.5 inches up to 22 inches were tabulated. For this purpose the connectors were divided into the following three groups, depending on the application:

1. The liquid oxygen system connectors.
2. The fuel system connectors.
3. The gaseous oxygen system connectors.

Using Saturn IB drawings and parts lists for flanges, gaskets, and bolts, dimensions such as inner diameter (ID), outer diameter (OD), flange width, thickness, material weight, and surface machining were obtained and shown in different tables.

The LOX system flange dimensions, and gasket (seal) and bolt dimensions are shown in Tables VI and VII, respectively. The fuel system flange

TABLE VI

FLANGE DIMENSIONS, LOX SYSTEM

Operating Pressure, psi	Flange Dimensions, in.					Finish, rms		Material		Weight, lb	
	Diameter		Width	Thickness							
	ID	OD		L[a]	R[b]	L	R	L	R	L	R
140	22.00	26.25	2.12	1.00	1.00	63	63	MIL-A-19842	—	21.75	21.75
	12.00	15.00	1.50	0.75	—	32	63	MIL-A-19842	—	5.46	—
	12.00	15.00	1.50	—	0.75	32	32	—	AL-5456	—	5.55
	12.00	15.00	1.50	0.81	—	63	32	AA-5456/321	—	6.02	—
n_1	8.00	10.75	1.40	0.87	—	63	32	QQ-A-601	—	3.97	—
n_2	8.00	10.75	1.40	0.75	—	32	32	CSTG	—	3.30	—
n_3	8.00	10.75	1.40	0.66	0.75	32	32	CRES	CSTG	3.39	—
n_4	8.00	10.75	1.40	0.87	—	125	63	QQ-A-106	—	4.25	—
n_5	7.78	10.38	1.30	0.63	0.75	63	32	QQ-A-601	QQ-A-601	3.92	—
n_6	7.82	10.50	1.30	0.66	—	32	32	QQ-A-601	—	3.62	—
90	6.00	8.25	1.17	—	0.63	32	63	—	QQ-A-355	—	1.68
	6.00	8.25	1.17	—	0.63	125	32	—	QQ-A-355	—	1.72

a. L = Left Flange
b. R = Right Flange
c. n_1 through n_6 Flanges Used for Numerical Example

TABLE VII

GASKET (SEAL) AND BOLT DIMENSIONS, LOX SYSTEM

Operating Pressure, psi	Gasket Dimensions, in.			Material	Bolt Dimensions, in.		
	ID	Width	Thickness		Bolt Circle Diameter	Bolt Diameter	Number of Bolts
140	23.25	0.625	0.125	Allpax 500	25.00	7/16	36
	12.00	0.380	0.062	Allpax 500	13.38	3/8	24
	12.80	0.380	0.062	Allpax 500	14.00	3/8	24
	12.80	0.380	0.062	Allpax 500	14.00	3/8	24
	8.50	0.380	0.062	K-Seal	9.50	3/8	12
	8.50	0.310	0.062	Allpax 500	9.50	3/8	12
	8.50	0.375	0.062	Allpax 500	9.75	3/8	18
	8.50	0.375	0.062	Allpax 500	9.75	3/8	18
	8.50	0.375	0.062	Allpax 500	9.75	3/8	18
	8.50	0.375	0.062	Allpax 500	9.75	3/8	18
	2.50	0.375	0.062	Allpax 500	3.64	5/16	8
90	6.25	0.375	0.062	Allpax 500	7.50	3/8	12
	6.25	0.375	0.062	Allpax 500	7.50	3/8	12

dimensions, and gasket (seal) and bolt dimensions are shown in Tables VIII and IX, respectively. The GOX system flange dimensions and gasket (seal) and bolt dimensions are shown in Tables X and XI. All weight data shown for the flanges were calculated. The weights for gaskets or seals and bolts were taken from catalogs or obtained from the specific manufacturer.

For the separable connector component, the flange, the gasket or seal, and the bolts, an effort was made to determine the cost parameter values. The following efforts and expenses determine the costs of flanges:

1. Engineering and managing effort to develop, test, and verify the prototype and flight article.
2. Manufacturing the prototype, test, and flight article.
3. Costs for materials, storage, and delivery.
4. Profit.

It can be said that the flanges used for the Saturn IB separable connectors are much more expensive than the ones built by flange and fitting manufacturers. The fact that each design organization developed, tested, and manufactured their own connectors for identical applications raised the cost tremendously. For example, a cost estimate made by a stage contractor for performing development and qualification testing of prototype connectors for his stage only, requests a total of \$500,000.00 for the program. Another example that demonstrates the high costs of connector testing is the one in which a contractor performed Taylor Forge lightweight and low profile configuration comparison tests at a cost of more than \$100,000. The connector hardware was provided by the Government. Also, one contributing

TABLE VIII

FLANGE DIMENSIONS, FUEL SYSTEM

Operating Pressure, psi	Flange Dimensions, in.					Finish, rms		Material		Area, in.^{2}	
	Diameter		Width	Thickness							
	ID	OD		L[a]	R[b]	L	R	L	R	L	R
140	10.00	12.50	1.75	0.75	0.64	63	32	QQ-A-601	—	0.93	0.64
	10.00	12.50	1.75	—	0.50	32	63	QQ-A-267	QQ-A-601	—	0.62
	8.00	10.80	1.40	0.87	0.87	32	32	QQ-A-601	MIL-A-19842	1.20	1.20
	8.00	10.80	1.40	0.87	0.75	32	32	QQ-A-601	QQ-A-267	1.20	0.93
	8.00	10.80	1.40	0.75	0.75	32	32	CSTG	QQ-A-267	1.03	0.89
	8.00	10.80	1.40	0.63	0.75	32	32	CSTG	QQ-S-766	0.86	1.03
	6.70	9.00	1.15	0.66	—	32	32	CRES 321	CSTG	0.76	—
	6.78	9.38	1.30	—	0.72	—	—	QQ-A-601	QQ-A-601	—	—
50	10.00	12.50	1.25	0.75	0.75	63	32	QQ-A-601	MIL-A-19842	0.83	0.83
	6.50	8.80	1.15	0.75	0.38	63	32	QQ-A-601	QQ-A-601	0.81	0.40
	6.50	8.80	1.15	0.38	0.75	32	63	QQ-A-327	QQ-A-601	Blind Flange	0.81
	6.50	8.80	1.15	0.75	0.75	32	63	QQ-A-355	QQ-A-601	2.15	0.75
	4.70	7.10	1.20	0.38	0.50	125	32	QQ-A-601	QQ-A-325	0.48	0.51
	4.00	7.00	1.50	0.50	0.38	—	—	QQ-A-355	QQ-A-601	0.69	0.55
	3.00	4.80	0.90	0.25	0.28	63	32	QQ-A-355	QQ-A-601	0.21	0.21

a. L = Left Flange
b. R = Right Flange

TABLE IX

GASKET (SEAL) AND BOLT DIMENSIONS, FUEL SYSTEM

Operating Pressure, psi	Gasket Dimensions, in.			Material	Bolt Dimensions, in.		
	ID	Width	Thickness		Bolt Circle Diameter	Bolt Diameter	Number of Bolts
140	10.00	0.75	0.62	MS-29513-450	11.75		
	10.00	O-Ring	—	MS-29513-450	12.12		
	8.00	O-Ring	—	MS-29513-268	9.75	3/8	18
	8.00	O-Ring	—	MS-29513-268	9.75	3/8	18
	8.00	O-Ring	—	MS-29513-268	9.75	3/8	18
	8.00	O-Ring	—	MS-29513-268	9.75	3/8	18
	6.70	O-Ring	—	MS-29513-261	8.13	3/4	8
	6.78	0.75	0.62	Johns-Manville	8.50		
50	10.00	O-Ring	—	MS-29513-450	11.50	3/8	24
	6.50	O-Ring	—	MS-29513-262	8.00	3/8	12
	6.50	O-Ring	—	MS-29513-262	8.00	3/8	12
	6.50	O-Ring	—	MS-29513-262	8.00	3/8	12
	4.70	O-Ring	—	MS-29513-254	6.40	5/16	12
	4.00	O-Ring	—	MS-29513-248	6.00	5/8	8
	3.00	O-Ring	—	MS-29513-236	4.20	1.4	8

TABLE X

FLANGE DIMENSIONS, GOX SYSTEM

Operating Pressure, psi	Flange Dimensions, in.					Finish, rms		Material		Weight, lb	
	Diameter		Width	Thickness							
	ID	OD		L[a]	R[b]	L	R	L	R	L	R
300	6.50	9.40	1.45	0.75	—	63	125	MIL-A-19842	—	2.98	—
	6.50	9.40	1.45	—	0.75	32	63	—	MIL-A-19842	—	2.39
	4.00	6.30	1.13	1.00	0.38	32	32	QQ-A-601	QQ-S-763	2.06	2.39
	4.00	6.30	1.13	—	0.38	32	32	QQ-S-763	QQ-S-763	2.41	2.41
	4.00	6.30	1.13	0.38	0.38	32	32	QQ-A-601	QQ-S-763	1.47	—
	4.00	6.30	1.13	0.38	—	32	63	QQ-S-763	QQ-A-335	5.93	1.50
	4.00	6.30	1.13	0.38	0.50	32	63	QQ-S-763	QQ-S-335	2.18	1.00
	4.00	6.30	1.13	0.38	0.38	32	32	QQ-S-763	QQ-S-763	—	2.26
100	22.00	26.25	2.12	1.00	1.00	63	63	MIL-A-19842	MIL-A-19842	—	21.51
	4.00	6.25	1.13	0.38	0.81	32	125	QQ-S-763	MIL-A-19842	2.00	1.66
80	7.00	9.50	1.25	0.75	—	32	—	QQ-A-601	—	2.46	—
	7.00	9.50	1.25	0.75	—	32	32	QQ-A-601	—	2.46	—
	6.90	9.75	1.42	0.38	0.81	32	125	QQ-S-766	MIL-A-19842	5.90	2.92
	6.88	9.50	1.31	—	0.75	—	63	—	QQ-A-601	—	2.68
	5.00	7.75	1.38	0.87	0.72	125	63	MIL-A-19842	QQ-A-601	2.92	—
	5.00	7.75	1.38	0.72	—	63	—	QQ-A-601	—	2.17	—
	4.75	7.50	1.38	—	0.75	—	125	—	QQ-A-327	—	2.17
	4.00	6.27	1.13	—	0.56	—	125	—	MIL-A-19842	—	1.34

a. L = Left Flange
b. R = Right Flange

TABLE XI

GASKET (SEAL) AND BOLT DIMENSIONS, LOX SYSTEM

Operating Pressure, psi	Gasket Dimensions, in.			Material	Bolt Dimensions, in.		
	ID	Width	Thickness		Bolt Circle Diameter	Bolt Diameter	Number of Bolts
300	7.35	0.370	0.062	Allpax 500	8.50	3/8	12
	7.30	0.400	0.062	Allpax 500	8.50	3/8	12
	4.25	0.375	0.062	Allpax 500	5.50	3/8	12
	4.25	0.375	0.062	Allpax 500	5.50	3/8	12
	4.25	0.375	0.062	Allpax 500	5.50	3/8	7
	4.25	0.375	0.062	Allpax 500	5.50	3/8	12
	—	—	—	Allpax 500	4.13	5/16	12
	3.00	0.344	0.062	Allpax 500	4.12	5/16	12
	3.00	0.344	0.062	Allpax 500	4.12	5/16	12
	3.00	0.344	0.062	Allpax 500	4.12	5/16	12
100	23.25	0.625	0.125	Allpax 500	25.00	7/16	36
	4.25	0.375	0.062	Allpax 500	5.52	3/8	8
80	7.50	0.310	0.62	Allpax 500	8.50	3/8	12
	7.50	0.310	0.062	Allpax 500	8.50	3/8	12
	7.50	0.362	0.062	Allpax 500	8.75	3/8	12
	7.50	0.310	0.062	Johns-Manville	8.50	3/8	12
	5.50	0.370	0.062	Allpax 500	6.75	3/8	12
	5.50	0.370	0.062	Allpax 500	6.75	3/8	12
	5.50	0.370	0.062	Allpax 500	6.75	3/8	12
	4.35	0.370	0.062	Allpax 500	5.52	5/16	12

factor to the high connector costs was the relatively small number of connectors built in comparison to the number of connectors built commercially by tube and flange manufacturers. The tabulated flange costs are certainly sufficient for this research because the objective of this study is to determine the optimum separable connector design without concentrating too much on exact values that are difficult to obtain and are not necessary to demonstrate the techniques of optimization.

To better understand the costs involved in building flanges, tables and curves are provided. Additional machining costs for different flange facings other than flat are shown in Figure 13, matching costs for different surface finishes are shown in Figure 14, and a flange finish cost comparison curve is shown in Figure 15. The flange costs shown in Table XII were obtained from True Dimensions, Inc. of Huntsville, Alabama [9]. These represent actual manufacturing costs of the flanges evaluated in this research. The prices are used as a base for the cost values necessary for the numerical example. The costs for flange facing and surface finish have to be added to the base prices shown in Table XII. Approximate gasket and seal costs of some commonly used configurations are shown in Table XIII. These values were obtained from engineers of the MSFC Propulsion Division.

Analysis and Evaluation of the Input Data

To enable a better understanding of the method used to determine the optimum separable connector design, the LOX system will be treated in

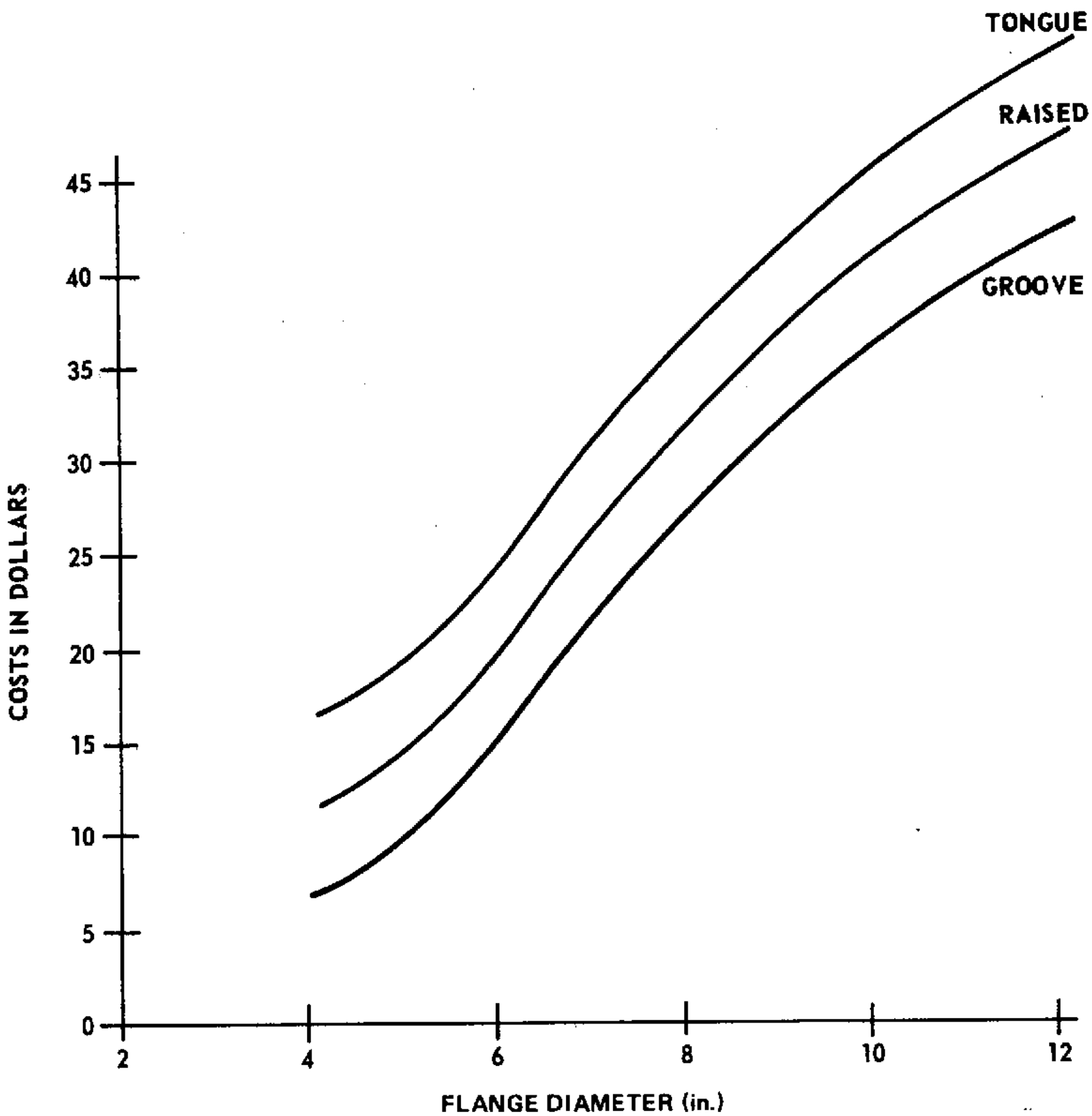


FIGURE 13. ADDITIONAL COSTS FOR FLANGE FACINGS OTHER THAN FLAT

detail. For this purpose all connectors under consideration used for the LOX system were analyzed and a design tree was established. Figure 16 shows the design tree structure. As shown by the design tree, there are 1,444 feasible combinations of configurations. The decision problem to be solved is to choose that configuration which best satisfies the design objectives.

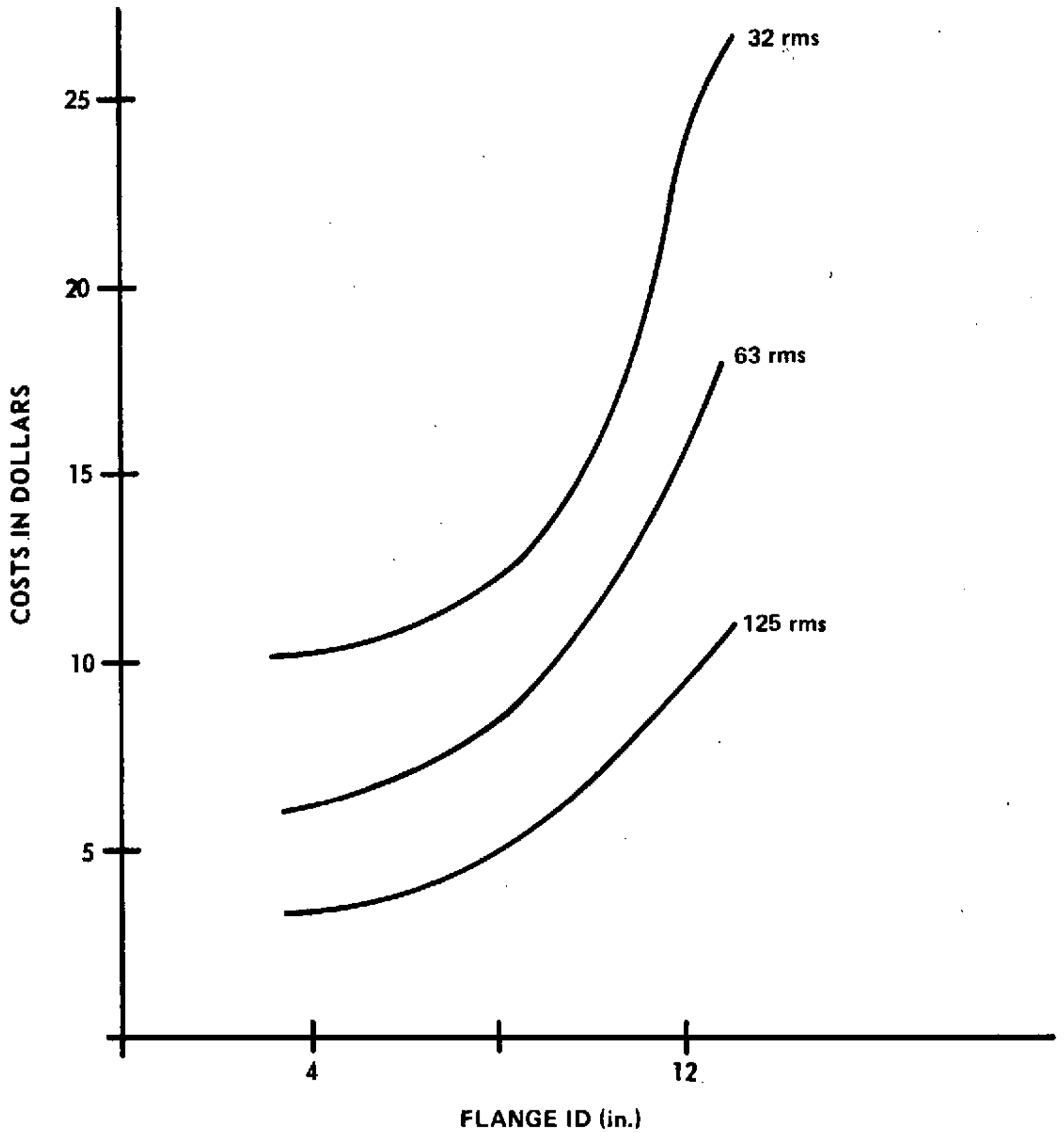


FIGURE 14. MACHINING COSTS FOR DIFFERENT SURFACE FINISH (rms)

The optimum design will be determined by the following parameters: performance, weight and costs, and any combination of them, depending on the criteria imposed. Performance is the system level parameter that is desirable to optimize. Any other parameter could be chosen for the purpose of demonstrating the usefulness of the method being used to determine

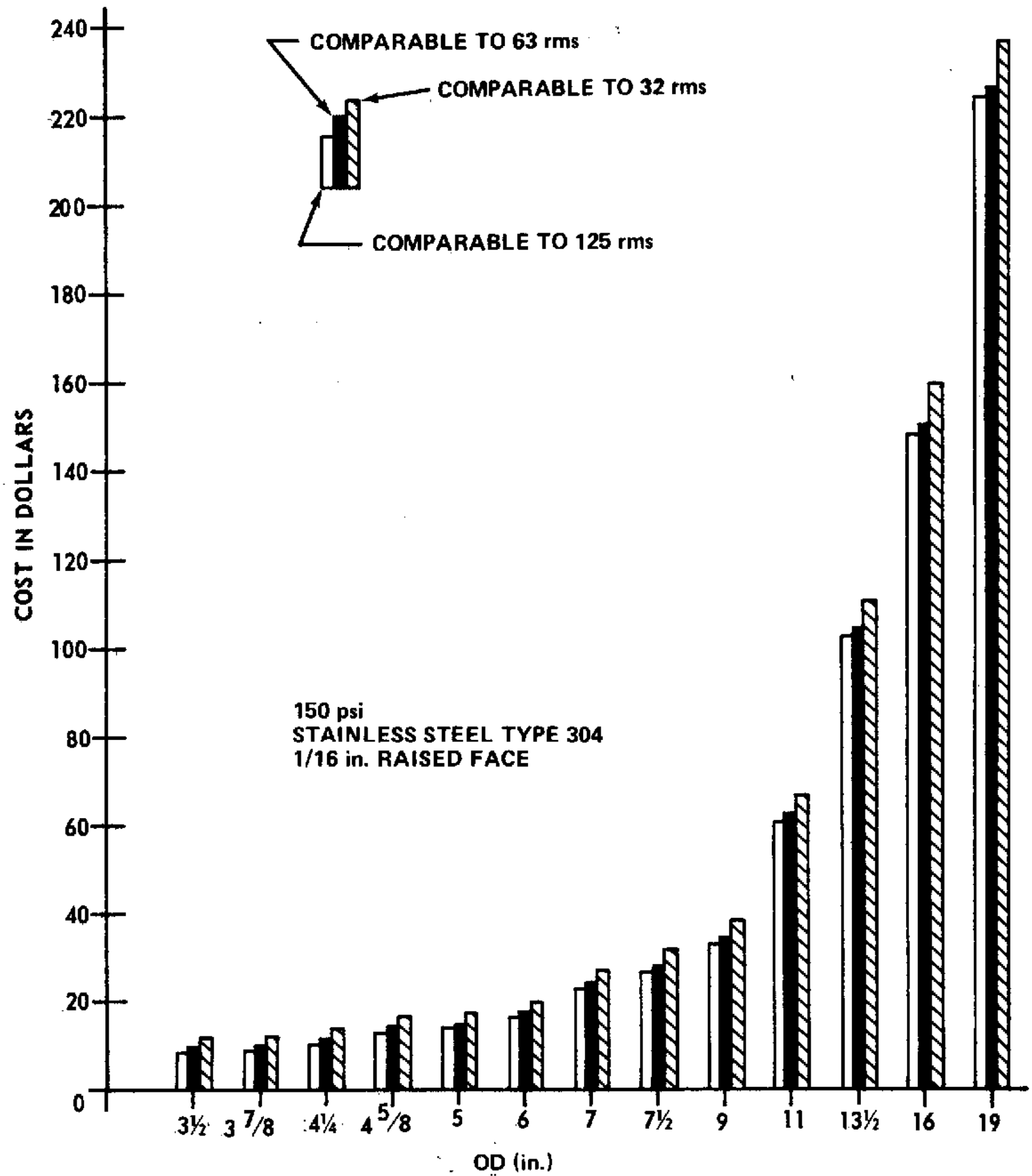


FIGURE 15. FLANGE FINISH COST COMPARISON
(STAINLESS STEEL)

the optimum separable connector design. If, for example, performance is less critical than weight, then that configuration has to be chosen where the performance stays within the tolerable leakage rate and the weight is at a minimum. The optimum could also be a connector system with the highest

TABLE XII

PRICES FOR COMMERCIAL LIGHTWEIGHT ALUMINUM FLANGES

ID, in.	OD, in.	Thickness, in.	No. of Holes	Diameter of Holes, in.	Price in $ each for	
					50 units	800 units
6.88	9.50	0.70	12	3/8	95	50
8.00	10.75	0.87	18	3/8	115	62
8.00	10.75	0.75	18	3/8	114	61
7.78	10.38	0.65	18	3/8	113	60
7.82	10.50	0.72	18	3/8	113	60
12.00	15.00	1.50	24	3/8	140	80
22.00	26.25	2.50	36	1/2	200	135

TABLE XIII

APPROXIMATE GASKET AND SEAL COSTS

Gasket or Seal	Cost in Dollars for			Material
	4-in. Diameter	8-in. Diameter	40-in. Diameter	
O-Ring	1.50	6.00	20.00	Synthetic, Organic, Plastic
Bar-X	30.00	78.00	530.00	CRES, Various Platings such as Gold, Silver, etc.
E-Ring	23.00	193.00	—	Multimet. N-155, Various Platings
Flat	2.50	3.20	15.00	Synthetic, Organic or Metallic
V-Seal	10.00	27.00	1000.00	CRES or Aluminum Alloys
RACO	13.00	34.00	250.00	CRES Spring, Teflon (FEP or TFE), KEL-F Coated
K-Seal	13.00	27.00	1300.00	CRES, Various Platings
NA-FLEX	35.00	69.00	250.00	CRES or Aluminum Alloy, Various Platings
Spiral Wound	1.50	4.00	25.00	CRES, Filler: Synthetic, Organic, Plastic
Serrated	10.00	21.00	250.00	CRES or Aluminum Alloy
Narmco	15.00	20.50	200.00	Fiberglass
Allpax	2.50	4.00	15.00	Asbestos
Johns-Manville	3.50	5.60	22.00	Asbestos

NOMENCLATURE

FLANGE

Configuration

1. Taylor Forge
2. Low Profile

Mounting

4. Fixed (Integral)

Assembly

6. Open

Facing

8. Flat
9. Raised
10. Male and Female
12. Tongue and Groove

Material

15. AL-5052
16. AL-6061

Finish

22. 32 microinch
23. 65 microinch

GASKET OR SEAL

Configuration

25. Flat Small
26. Flat Wide

Material

30. Allpax 500
34. Narmco
36. Johns Manville

Treatment

37. Nontreatment
38. Luboil

BOLT

Shape

39. Hexagonal
40. Allen Head

Material

41. Aluminum

NUT

Shape

44. Hexagonal
45. Round

Material

46. Aluminum
47. Steel

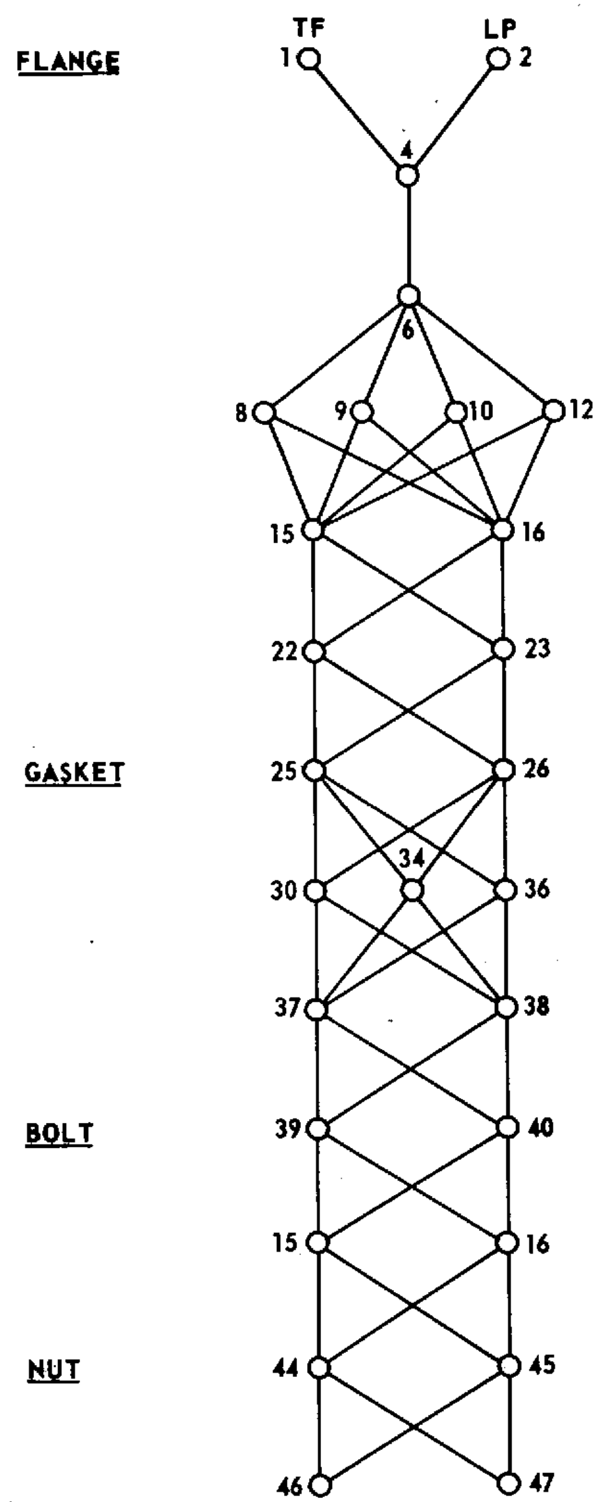


FIGURE 16. DESIGN TREE FOR SATURN IB FIRST STAGE LOX SYSTEM

performance rating possible and minimum weight, whereby the costs are of second nature.

The parameters selected for this research are performance, weight, and cost of the connector. The weight parameters are given in pounds and the cost parameter in dollars. The performance parameter was determined to be dimensionless and composed of the following values:

1. The stress values of the connector components.
2. The properties of the materials.
3. The performance data (leak rates) obtained from test and flight analysis.

The assessment of the performance parameter value was the most difficult and frustrating to establish. Questions had to be answered such as: How much does flange distortion influence the development of a leak? To what extent does gasket or seal pressure relaxation influence the sealing characteristics? Does degradation of the materials contribute to leakage and if so, when and after what period of time? In selecting and evaluating the bolts, the bolt load and torquing procedure posed questions concerning this degree of influence in the connector's performance. A more difficult question, which requires a science of its own to be answered, was to assess the effect of interaction of the various individual components on the performance of the connector system under operational conditions.

At the beginning of the evaluation of the connector systems under investigation, it was assumed that all connectors performed satisfactorily within an established performance value range. This assumption was substantiated by development testing, static firing, and to some extent by flight evaluation data. For stress analysis purposes, a way had to be found to obtain scaling methods to compare similar connector configuration stress values without analyzing each and every connector under investigation. One of the existing scaling methods used for stress analysis is the Buckingham pi theorem explained by Focken [10]. This theorem presents a logical and simple procedure for comparing stresses of geometrically identical bodies, introducing ratios as independent arguments of the unknown functions. The advantage of the pi theorem is that one does not have to know a mathematical relationship between the variables that define the phenomenon under investigation. If n variables define a phenomenon and if each variable may be expressed in m dimensions, the general equation for the phenomenon may be expressed in $n-m$ terms. Each dimensionless term may be composed of $m+1$ variables, m of which will be common to all terms.

Consider, for example the stress distribution over the flange plate width using the equation,

$$\sigma = \frac{6M}{wt^2} \quad ,$$

where

$$\sigma = f(M, w, t) \quad ,$$

where

σ = stress (lb/in.^{2}),

w = flange width (in.),

t = flange thickness (in.),

and

M = bending moment (in.-lb).

The basic dimensions are F and L where

F = some force

and

L = some length;

the variables then can be expressed in these two basic dimensions as follows:

$\sigma = [F \cdot L^{-2}]$,

$w = [L]$,

$t = [L]$,

and

$M = [F \cdot L]$.

The equation

$$\sigma = \frac{6M}{wt^2}$$

could be expressed by a dimensionless function,

$$\phi = \phi[M, w, t, \sigma] \quad .$$

The number of variables is $m = 4$, and the number of basic dimensions is $n = 2$. The number of dimensionless pi terms is

$$m - n = 2 \quad .$$

Each dimensionless pi term is composed of $m+1$ variables, m of which will be common. M and t are chosen as the common variables. The two pi terms are

$$\pi_1 = M^{a_1} t^{b_1} \sigma^{c_1} = (FL)^{a_1} L^{b_1} FL^{-2} = F^0L^0$$

and

$$\pi_2 = M^{a_2} t^{b_2} w^{c_2} = (FL)^{a_2} L^{b_2} L = F^0L^0 \quad .$$

In these terms, $c_1 = c_2 = 1$. By comparing the exponents of F and L , the unknown a's and b's can be found to be

$$a_1 = -1 \quad ,$$

$$b_1 = 3 \quad ,$$

$$a_2 = 0 \quad ,$$

and

$$b_2 = -1 \quad .$$

If the scaling law is expressed as

$$L = \ell' \overline{L} \text{ and } F = f' \overline{F} \quad ,$$

the scale factor relating σ and $\overline{\sigma}$ is found from the dimensionless expression $\pi_1 \pi_2$,

$$\frac{\sigma t^3}{M} \; \frac{w}{t} = \frac{\overline{\sigma} \overline{t}^3}{\overline{M}} \; \frac{\overline{w}}{\overline{t}}$$

and

$$\sigma = \frac{\left(\frac{\overline{t}}{t}\right)^2 \frac{\overline{w}}{w}}{\frac{\overline{M}}{M}} \; \overline{\sigma} \quad .$$

When the scale factors ℓ' and f' are substituted, this relation reduces to

$$\sigma = \frac{(\ell')^2 (\ell')}{(f' \ell')} \; \overline{\sigma} = \frac{\ell'^2}{f'} \; \overline{\sigma} \quad .$$

Let $\ell' = 1.2$ and $f' = 1.4$, then

$$\sigma = \frac{1.44}{1.4} \; \overline{\sigma} = 1.03 \; \overline{\sigma} \quad .$$

Since the flanges under investigation are geometrically similar bodies, the Buckingham pi theorem can be used for comparing stress distribution on these flanges. With the establishment of the scale factor, the number of calculations is reduced by an appreciable amount.

Another approach to compare flange stresses was developed in this research using ratios of flange dimensions with a common denominator. This method was applied first to compare the angular flange deflections that were calculated using the equation

$$\delta = \frac{F_b \cdot a' \cdot r_m^2}{E_f \frac{w \cdot t^3}{12}} ,$$

where

δ = angular flange deflection (dimensionless),

F_b = bolt force (lb),

F_p = pipe force (lb),

a' = distance between the force vector F_b and center of gasket (in.),

r_m = radius between force vector F_b and centerline of pipe (in.),

t = flange thickness (in.),

w = flange width (in.),

and

E_f = flange material modulus.

From this equation the angular flange deflection δ (roll) also can be expressed with the following ratios:

$$\delta = \frac{F_b \cdot a' \cdot r_m^{\;2}}{E_f \frac{w \cdot t^3}{12}} = \frac{12\, F_b \cdot a' \cdot r_m^{\;2}}{E_f \cdot w \cdot t^3} \quad \frac{12\, \frac{F_b}{r_m} \cdot \frac{a'}{r_m}}{E_t \frac{\frac{w}{r_m} \quad \left(\frac{t}{r_m}\right)^3}{12}} \; .$$

The ratios obtained from this calculation are

$$\frac{a'}{r_m} \;, \quad \frac{w}{r_m} \;, \quad \frac{t^3}{r_m} \;, \quad \text{and} \quad \frac{F_b}{r_m} \; .$$

Of these dimensionless ratios it was found advantageous to employ the following three ratios for flange comparison purposes:

1. The ratio of flange thickness t to the inner diameter I.D., t/ID.
2. The ratio of flange width w to the inner diameter I.D., w/ID.
3. The ratio of the bolt center diameter D_B to the gasket center diameter; D_G, D_B/D_G.

These ratio values were computed and are shown in Tables XIV, XV, and XVI. The ratio values were also plotted for each operational pressure of each system shown in Figures 17, 18, and 19. The established reference values of the Taylor Forge lightweight flange are marked with a cross in Figures 17, 18, and 19. These curves were used to determine the performance parameter value of the flanges. The performance parameter value P_1

TABLE XIV

FLANGE EVALUATION, LOX SYSTEM

Operating Pressure, psi	ID, in.	Thickness, t, in.		x, in.	y, in.	Width, w, in.	Gasket Center Diameter, D_G, in.	Bolt Circle Diameter, D_B, in.	$\frac{D_B - D_G}{ID}$	t/ID		w/ID	D_B/D_G
		L[b]	R[c]							L	R		
Aluminum	22.00	1.00	1.050	0.650	1.500	2.12	23.875	25.00	0.0513	0.046	0.046	0.096	1.05
140	12.00	0.750	—	0.812	0.687	1.50	12.380	13.38	0.0830	0.025	—	0.125	1.08
	12.00	—	0.750	0.500	1.000	—	13.180	14.00	0.0685	0.073	0.062	0.125	1.06
	12.00	0.813	—	1.00	1.50	—	13.180	14.00	0.0710	0.088	0.068	0.125	1.06
n_1[a]	8.00	0.872	—	0.500	0.875	1.38	8.875	9.75	0.1219	—	0.109	0.173	1.14
n_2	8.00	0.750	—	0.500	0.875	1.38	8.875	9.75	0.1220	0.094	—	0.173	1.14
n_3	8.00	—	0.750	0.500	0.875	—	8.875	9.75	0.1220	—	0.082	0.173	1.14
n_4	8.00	0.870	—	0.500	0.875	1.37	8.875	9.75	0.1210	0.109	—	0.173	1.14
n_5	7.78	—	0.720	0.440	0.880	—	8.817	9.50	0.1380	—	0.107	0.192	1.09
n_6	7.82	0.660	—	0.440	0.880	1.30	8.817	9.50	0.1020	0.095	—	0.192	1.09
Steel 140	6.82	0.650	—	0.500	0.840	1.34	6.850	8.50	0.2420	0.095	—	0.193	1.24
Aluminum	6.00	—	0.625	0.375	0.750	—	6.625	7.50	0.1460	0.103	—	0.195	1.14
90	6.00	—	0.625	0.375	0.750	—	6.625	7.50	0.1460	0.103	—	0.195	1.14

a. n_1 through n_6 Flanges Used for Numerical Example
b. L = Left Flange
c. R = Right Flange

TABLE XV

FLANGE EVALUATION, FUEL SYSTEM

Operating Pressure, psi	ID, in.	Thickness, t, in.		x, in.	y, in.	Width, w, in.	Gasket Center Diameter, D_G, in.	Bolt Circle Diameter, D_B, in.	$\frac{D_B-D_G}{ID}$	t/ID		w/ID	D_B/D_G
		L[a]	R[b]							L	R		
Aluminum	10.00	0.750	0.640	0.375	0.875	1.250	10.35	11.75	0.142	0.075	0.064	0.125	1.13
140	10.00	—	0.500	0.375	1.060	1.435	11.08	12.12	0.104	—	0.050	0.144	1.09
	8.00	0.870	0.870	0.500	0.875	1.375	8.60	9.75	0.130	0.110	0.110	0.172	1.04
	8.00	0.870	0.750	0.500	0.875	1.375	8.55	9.75	0.150	0.110	0.094	0.172	1.04
	8.00	0.750	0.750	0.500	0.867	1.375	8.55	9.75	0.150	0.094	0.094	0.172	1.04
	8.00	0.625	0.750	0.500	0.870	1.300	8.55	9.75	0.150	0.078	0.110	0.162	1.04
Steel	6.78	—	0.720	0.440	0.860	0.720	7.80	8.50	0.103	—	0.107	0.127	1.01
140	8.00	—	0.750	0.500	0.870	—	8.55	9.75	0.150	—	0.094	0.170	1.04
	6.70	—	0.760	0.440	0.950	—	7.06	8.12	0.158	—	0.114	0.207	1.16
Aluminum	10.00	0.750	0.750	0.500	0.750	1.250	10.50	11.50	0.100	0.075	0.075	0.125	1.09
50	6.50	0.750	0.380	0.400	0.750	1.150	6.94	8.00	0.154	0.116	0.057	0.177	1.15
	6.50	0.750	0.750	0.400	0.750	1.150	6.94	8.00	0.154	0.116	—	0.177	1.15
	6.50	0.750	0.750	0.400	0.750	1.150	6.94	8.00	0.154	0.116	0.116	0.178	1.15
	4.70	0.380	0.500	0.360	0.870	1.230	5.60	6.40	0.170	0.081	0.106	0.260	1.14
	4.00	0.500	0.380	0.500	1.000	1.500	4.70	6.00	0.325	0.126	0.098	0.375	1.28
	3.00	0.250	0.280	0.300	0.600	0.900	3.72	4.20	0.166	0.084	0.093	0.300	1.14

a. L = Left Flange
b. R = Right Flange

TABLE XVI

FLANGE EVALUATION, GOX SYSTEM

Operating Pressure, psi	ID, in.	Thickness, t, in.		x, in.	y, in.	Width, w, in.	Gasket Center Diameter D_G, in.	Bolt Circle Diameter D_B, in.	$\frac{D_B - D_G}{ID}$	t/ID		w/ID	D_B/D_G
		L[a]	R[b]							L	R		
Aluminum 300	6.50	0.750	—	0.50	1.00	1.500	7.720	8.50	0.120	0.115	—	0.232	1.06
	6.50	—	0.750	0.45	1.00	1.450	7.700	8.50	0.123	—	0.115	0.232	1.10
	4.00	1.000	—	0.38	0.75	1.130	4.625	5.50	0.220	0.250	—	0.283	1.18
	4.00	0.375	—	0.38	0.75	1.130	4.625	5.50	0.290	0.094	—	0.283	1.18
	3.00	1.100	—	0.38	0.55	0.937	—	4.13	—	0.368	—	0.313	—
	3.00	1.100	—	0.38	0.56	0.938	3.344	4.12	0.260	0.368	—	0.313	1.23
Steel 300	4.00	—	0.375	0.38	0.75	1.130	4.625	5.50	0.280	—	0.094	0.283	1.18
	4.00	—	0.375	0.38	0.75	1.130	4.625	5.50	0.280	—	0.094	0.283	1.18
	4.00	—	0.375	0.38	0.75	1.130	4.625	5.50	0.280	—	0.094	0.283	1.18
	4.00	0.375	—	0.38	0.75	1.130	4.625	5.50	0.280	0.094	—	0.283	1.18
	3.00	0.375	—	0.38	0.56	0.940	3.344	4.12	0.260	0.126	—	0.314	1.23
	3.00	0.375	—	0.38	0.56	0.940	3.344	4.12	0.260	0.126	—	0.314	1.23
Aluminum 100	22.00	1.000	1.000	0.65	1.50	2.120	23.875	25.00	0.051	0.045	0.045	0.096	1.05
	4.00	—	0.812	0.37	0.76	1.130	4.625	5.52	0.224	—	0.203	0.283	1.20
Steel 100	4.00	0.375	—	0.37	0.76	1.130	4.625	5.52	0.224	0.094	—	0.283	1.20
Aluminum 80	7.00	0.750	—	0.50	0.75	1.250	7.810	8.50	0.094	0.107	—	0.178	1.09
	7.00	0.750	—	0.50	0.75	1.250	7.810	8.50	0.094	0.107	—	0.178	1.09
	6.90	—	0.812	0.50	0.92	1.420	7.862	8.75	0.129	—	0.118	0.206	1.11
	6.90	—	0.720	0.50	0.81	1.310	7.810	8.15	0.050	—	0.104	0.190	1.09
	5.00	0.870	0.720	0.50	0.87	1.375	5.870	6.75	0.176	0.144	0.144	0.276	1.15
	5.00	0.720	—	0.50	0.87	1.375	5.870	6.75	0.176	0.144	—	0.276	1.15
	4.80	—	0.750	0.37	1.00	1.370	5.870	6.75	0.176	—	0.156	0.286	1.15
	4.00	—	0.562	0.37	0.76	1.130	4.720	5.52	0.200	—	0.140	0.283	1.17
	7.00	—	0.812	0.50	0.92	1.420	7.860	8.75	0.127	—	0.116	0.203	1.11

a. L = Left Flange
b. R = Right Flange

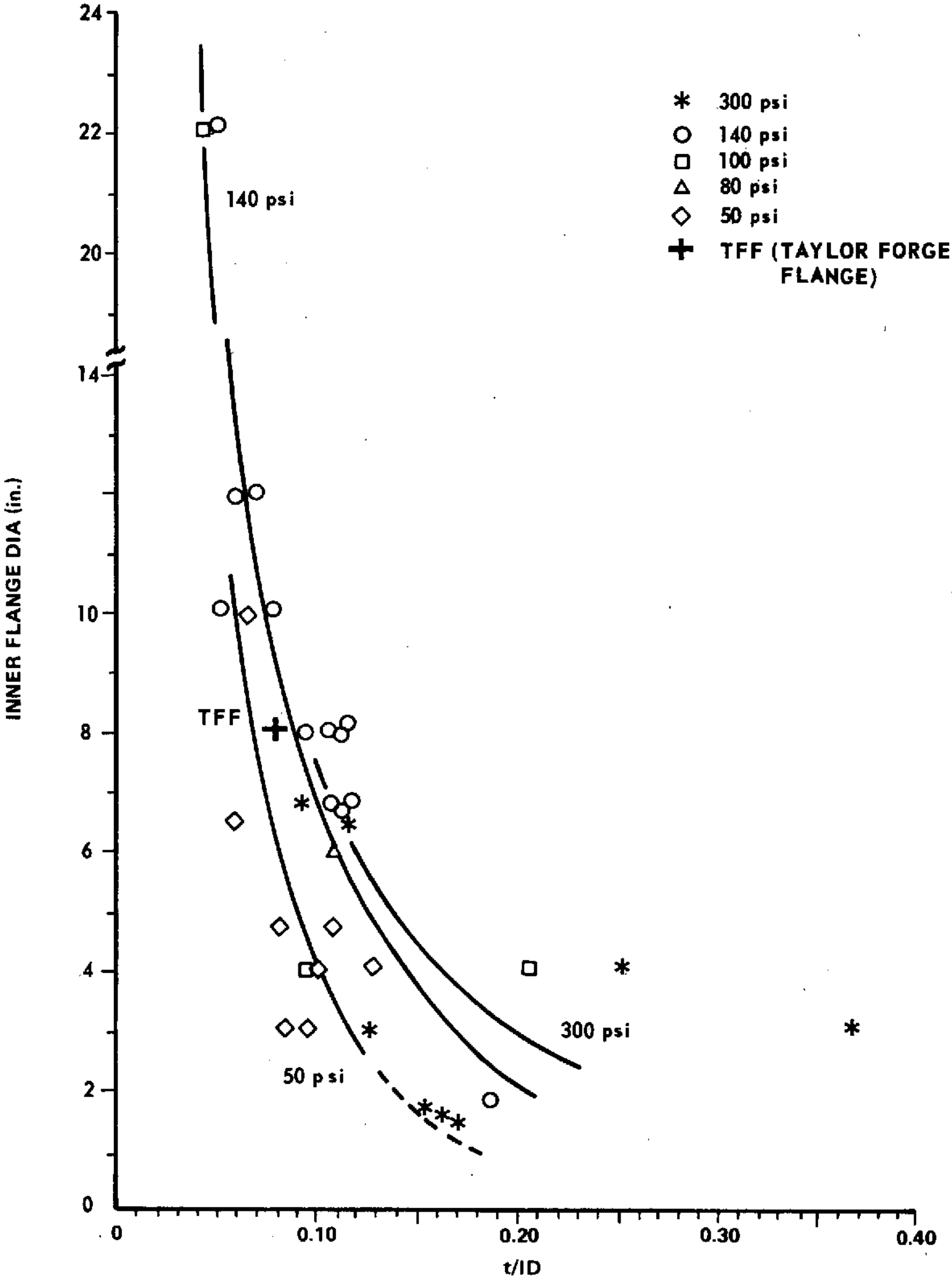


FIGURE 17. FLANGE THICKNESS t VERSUS ID FOR LOX, FUEL, AND GOX SYSTEMS

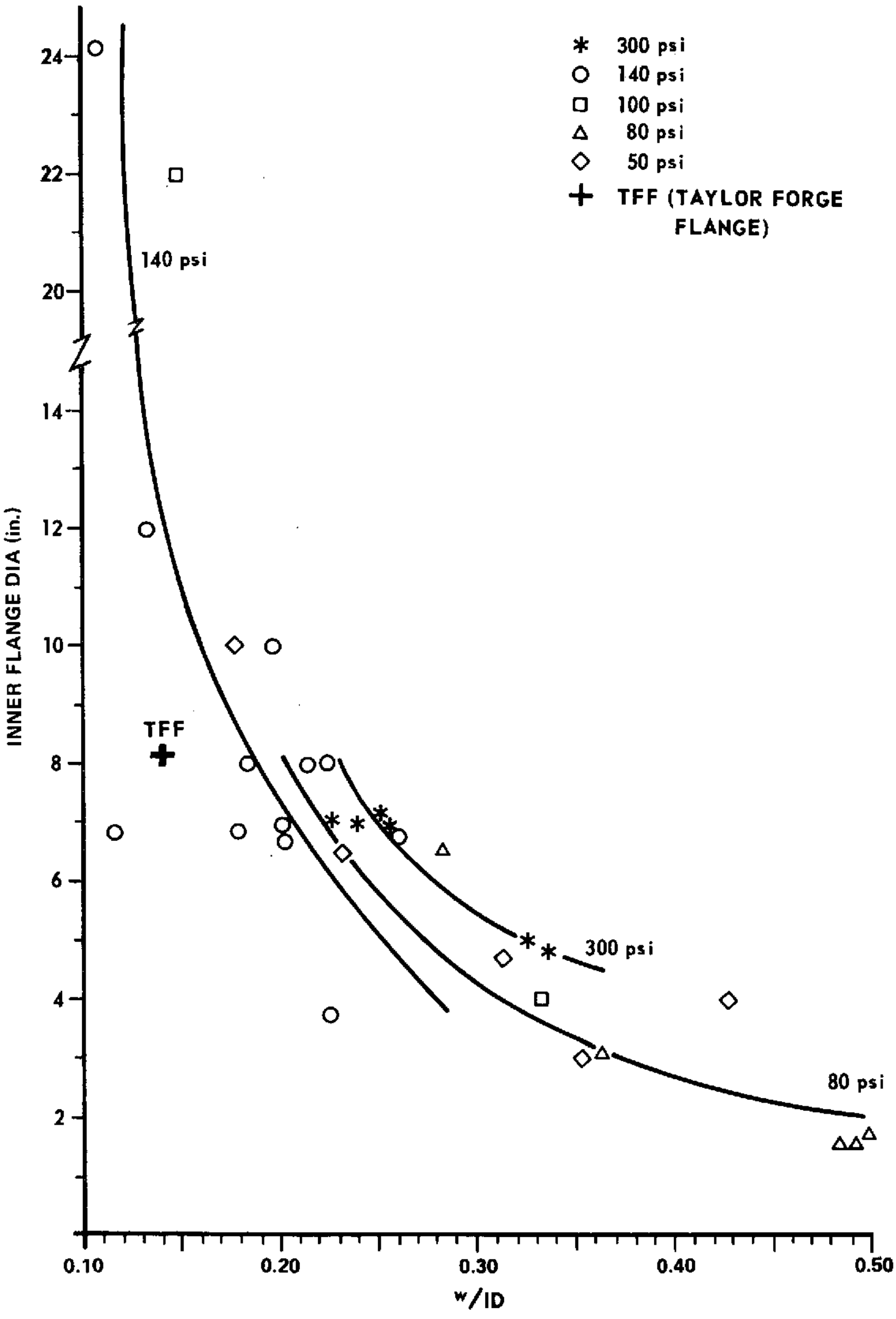


FIGURE 18. FLANGE WIDTH w VERSUS ID FOR LOX, FUEL, AND GOX SYSTEMS

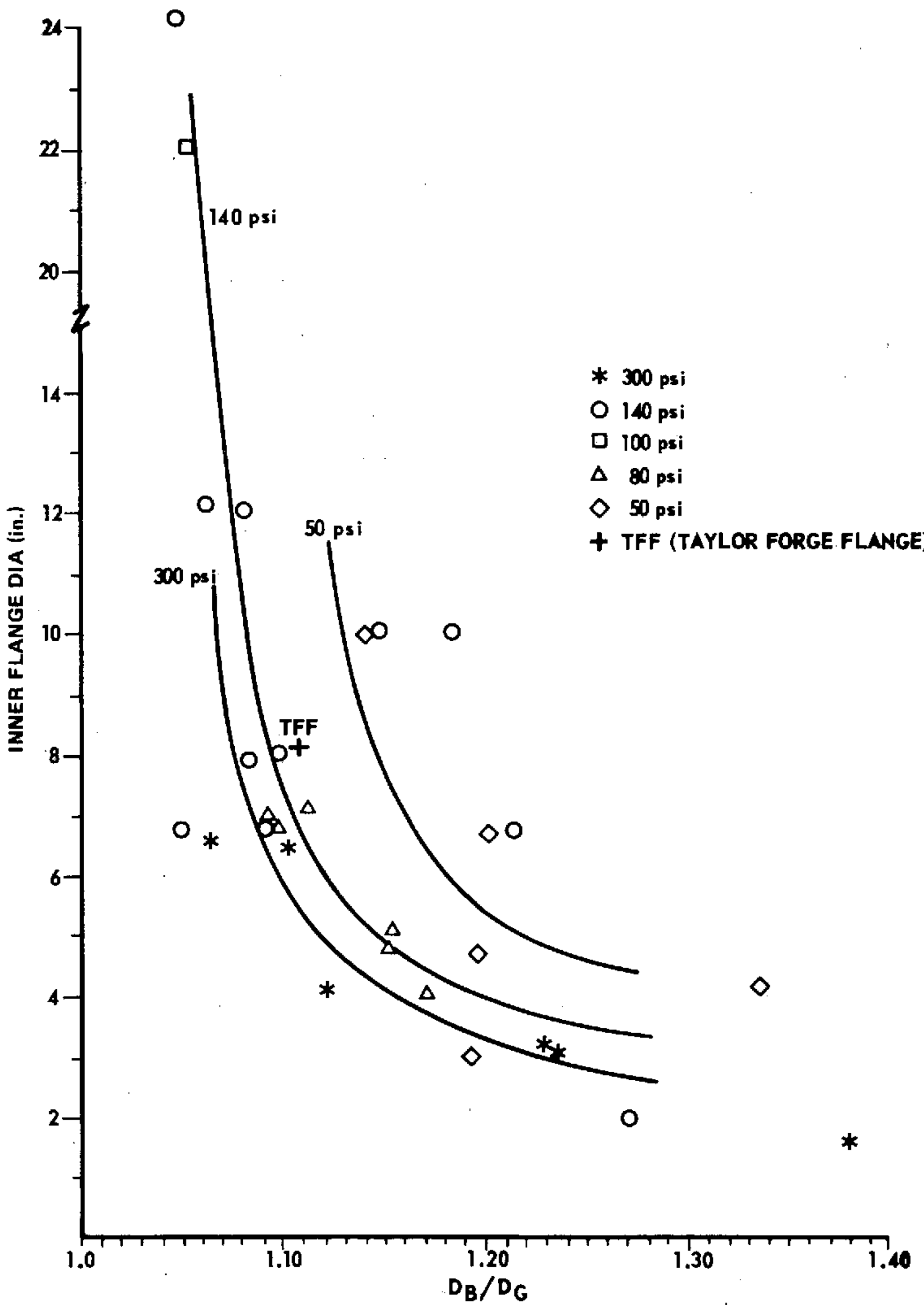


FIGURE 19. BOLT CIRCLE DIAMETER D_B VERSUS GASKET CENTER DIAMETER D_G FOR LOX, FUEL, AND GOX SYSTEMS

was obtained by the product of the three ratios: t/ID, w/ID, and D_B/D_G, as explained later in the numerical example.

The established curves represent the calculated ratios obtained from all analyzed flanges and exclusively are not sufficient to determine flange configurations. Again, the wide dispersion of the ratio values shown in the three figures clearly indicates the inconsistency in the design and calculation of flanges. It should be mentioned that for flanges smaller than 5 inches, the slope of the curves becomes more meaningless because of the design requirement to provide a torque wrench clearance between bolt heads and flange neck and also provide space for the gasket or seal. The implementation of these requirements results in relatively wider flanges.

To make this study meaningful, assumptions had to be made where data could not be obtained from documents, but the number of them were held to a bare minimum, and such values were substantiated by empirical data and/or by subjective judgement.

Figure 20 was developed to organize and document the important parameters and subparameters needed for gasket and seal performance evaluation. Table XVII was developed to assess parameter values for gaskets and seals. The major criteria such as gasket loading, allowable flange deflection, distortion, surface finish, surface flatness, and operating temperatures and pressures were determined. Figure 20 should be used in connection with Table XVII for a better prediction of gasket and seal performance in a given environment.

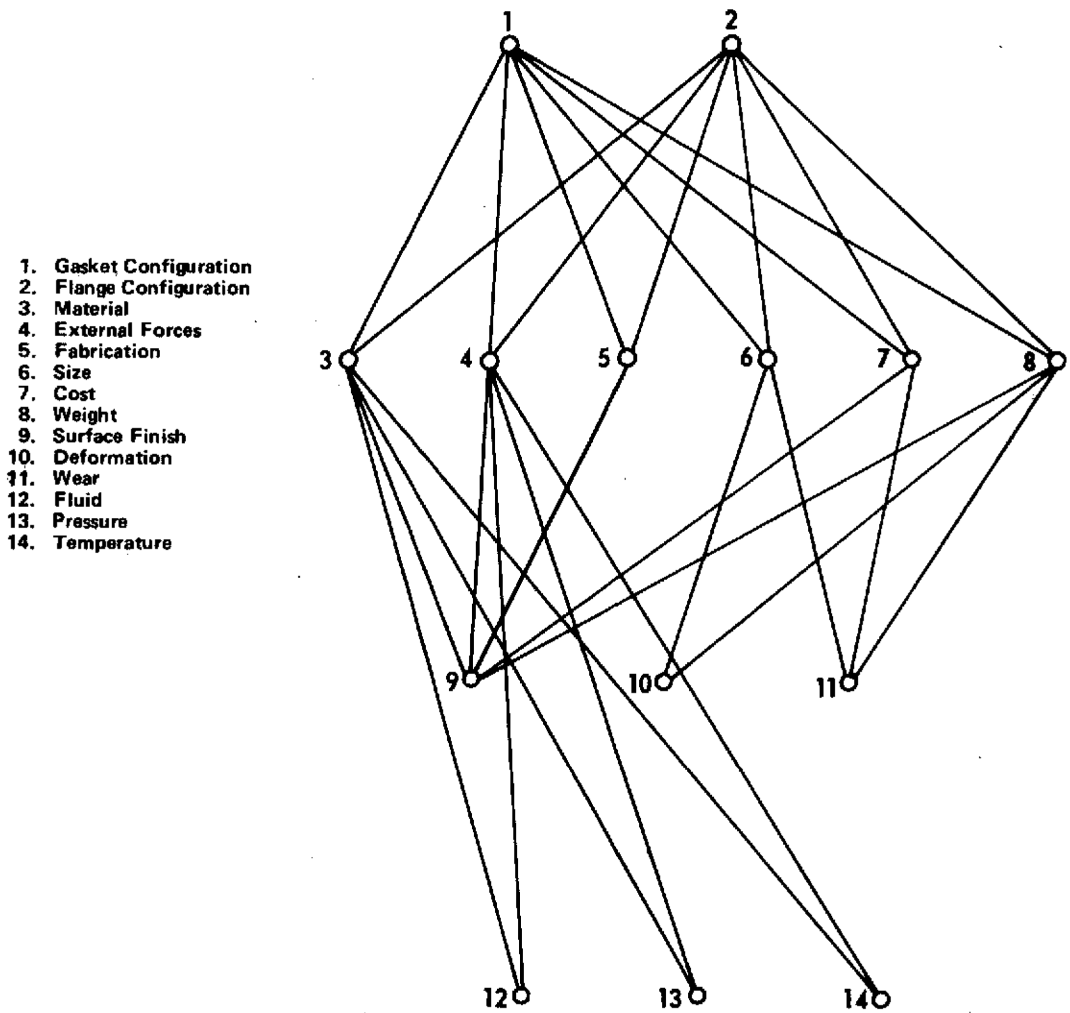


FIGURE 20. PARAMETERS INFLUENCING GASKET OR SEAL PERFORMANCE

To develop Table XVII, data for gaskets and seals were collected from various sources. The most detailed information on seals was found in References 3 and 11. The data presented for gasket loading, allowable flange deflection capabilities, and machining requirements were provided primarily by the manufacturer. Two flat gaskets, an asbestos Luboil-coated gasket and a Narmco gasket which is a fiberglass-epoxy type gasket, and two seals were analyzed (see Table XVII). One of the two seals is an O-ring made from

TABLE XVII

GASKET AND SEAL PROPERTIES

Configuration	Material		Gasket Loading, lbs/linear in.	Allowable Flange Deflection, in.			Maximum Allowable Distortion	Surface Finish	Surface Flatness	Operating Conditions	
	Base	Coating		Axial	Circular	Radial				Temperature, °F	Pressure, psi
Flat	Asbestos	Luboil	2000-6000	0.0000	0.0005	0.006	0.0000	32,125	0.001	-423 +2000	2000
Flat	Narmco	None	2000-6000	0.0000	0.005	0.0000	0.0000	65,125	0.001	-423 +1000	2000
O-Ring	Synthetic, Organic, Plastic	None	1300	0.010	0.010	0.010	0.010	32-64	0.003	-300	7000
K-Seal	CRES	Various Platings: Gold, Silver, Teflon	30-60	0.004	0.005	0.010	0.001	16-32	0.0002	-423 +1500	6000

synthetics or plastics, and the other is a K-shaped stainless steel body coated with Teflon. The O-ring and the K-seal belong to the family of pressure-actuated seals, where the operating pressure deforms the seal which then increases the contact load between the flange seal interfaces and results in a better sealing capability. One can see the different performance characteristics by comparing seals and gaskets. Flat gaskets for example can not compensate flange deflection, and they need very high seal loads. O-rings and K-seals need lower seal pressures and can compensate flange deflection better than gaskets. Gaskets can tolerate surface irregularities, scratches for example, better than seals. Gaskets can be reused; seals are not reusable.

The reader should remember that the accuracy of the result of research such as this depends on the accuracy of the input data. The approach toward converting the multistage decision process into a series of single stage problems is also important. The conversion is based on the principle that whatever the first decision is, the remaining decision must be optimal to the outcome that results from the first decision.

Parameter Value Assessment

Decision modeling was used where quantitative values are attached to parameters. An importance rating was performed, showing that performance is the most important parameter. The separable connectors under investigation were evaluated with consideration given to the anticipated leakage rate and the connector's reliability for maintaining its sealing characteristics during operational life. The data provided for this research were used to measure the performance of the connectors by evaluating flange stresses such as rolling of the flange, radial extension and contraction, the flange surface waviness and the interaction between flanges, the gasket or seal, and the bolts.

The complexity of the system dictated that the decision could not be derived from intuition, guess, and experience alone, but where experience was needed for the evaluation, experts from different engineering departments were consulted to assist in the determination of parameter values. A measure had to be established for the "goodness" of a separable bolted connector. To determine a measurement of goodness, several design approaches such as the following were considered. Where should redundancy be incorporated to meet the requirements? Is a higher price and longer leadtime for high reliability components justified? If any of the requirements should be changed, what would be the effect on the whole system? When designing a separable connector, the constraints and performance limitations include the state-of-the-art of the connector and gasket design, the weight limitations, and the limitation of funds. For the appropriation of funds, one must consider not only the amount of money required to make a system workable, but also the money that is lost if the system fails.

The correct assessment of the rating values is the key to a valid successful optimization study; for this reason, the author believes it is absolutely necessary to consult experienced engineers and experts for the analysis and evaluation. The analysis must reflect the importance of the parameter value itself for the component, and its importance within the total system.

The LOX System Design Tree Analysis

An analysis of the design tree previously shown in Figure 16 shows that under the assumption that all interconnecting branches represent feasible interactions, a total of 1,444 combinations have to be considered. This simplified design tree demonstrates the complex problem of finding the best, optimum configuration from the existing ones. We cannot simply pick the best flange design from the feasible flange designs and then select the best gasket design while ignoring the choice of the flange design, and continue in the same way to choose the other components. We must evaluate the system as a whole to ensure that all relevant dependencies and interactions are assessed. Consequently, the design decision will be composed of several decision problems where the individual decisions are independent of each other. In evaluating Figure 16 it was found that only integral flanges, either Taylor Forge flanges (1)[1] or some type of low profile flanges (2), were used. The majority of the flanges were open, which means that the gasket separates both flanges from each other (6). A total of four different flange faces were designed; flat (8), raised (9), male and female (10), and tongue and groove (2). Because the material for the suction lines was already predetermined by the propulsion system group, the design engineer had to select the material for flanges in

1. Numbers in parentheses refer to the Nomenclature of Figure 16.

only two cases. The two materials selected were aluminum AL 5052 (15) and AL 6061 (16). For the design of the gaskets two configurations, a small flat (25) and a wide flat (26), were used. Allpax 500(30) treated with Luboil (38), Narmco (34), and Johns-Manville (36) were the materials used for the gaskets. The bolts had either hexagonal (39) or Allen heads (40), and the materials used were aluminum AL 5052 (15), and AL 6061 (16). Aluminum (46) and steel (47) were used for the nuts.

Component Simplicity Rating

In support of the optimization calculation, a method to evaluate the design, called the "Component Simplicity Rating Method" [12], was used. The objective of the simplicity rating system is to achieve less complex design by means of comparison, where cost figures alone do not decide the component configuration to be used. Simplicity rating preferably is used prior to making the part, which in turn reduces the cost of manufacturing the part. This method also is very helpful in comparing mechanical components in accordance with machining processes, its compatibility within the system, complexity of the component, and advantages and disadvantages of the material, as well as manufacturing processes, evaluations, and weight determinations.

Using this method, in addition to the flange geometry ratios w/ID, t/ID, and D_B/D_G for flange performance value assessment, the following factors for flanges were established:

1. Factor A, flange flatness (parallelism) (in.).
2. Factor B, flange finish (rms) (μ in.).
3. Factor C, flange facings.
4. Factor D, flange envelope.

For each of the factors A, B, C, and D, an importance rating measured in degree of importance and a weighting measure of the factor was established. The factor weight was measured depending on the design properties such as the flange facing, surface machining, and the flange shape. The information below was developed for evaluation purposes for the flanges under investigation.

Factor	Degree of Importance	Dimension or Characteristic	Factor Weight
A, Flange Flatness	3	0.002 in. 0.005 in. 0.010 in.	3 2 1
B, Flange Finish	3	32 rms 65 rms 125 rms	1-3 1-3 1-3
C, Flange Facing	2	Tongue and Groove Raised Plate	1-3 1-3 1-3
D, Flange	1	Small Medium Large	3 2 1

The factor weight value for factors B and C varies from 1 to 3 depending on the criteria to be measured. If, for example, the selection of the flange facing or the flange finish was poor, then the lower value is used; however, the selection is such that a better finish than needed is selected, then the factor weight is selected high but the costs are raised for unnecessary machining of the flange surface. The product of the degree of importance value and the factor weight value equals the performance index value. The performance index value in its normalized form is used as a multiplicative parameter value for the overall performance parameter value assessment, (Performance Index) = (Degree of Importance) × (Factor Weight Value).

Table XVIII was developed for the evaluation of gasket and seal performance. This table shows the criteria needed to predict gasket and seal performance in given environments. For this table the different rating values were established as discussed in the following. A weighted percentage was assigned to each of the three criteria according to existing data and the way experts rated them. (The sum of the percentage must equal unity.) In this 0.40 percent was the weighing number for reliability, 0.31 for performance, and 0.29 for deformation adaptability. Then a number of 100 points were assigned to each major criterion weighing the importance, performance, and other considerations of its subcriteria. Where the performance or compatibility of a subcriterion was not perfect, a weighted number of points was subtracted from the assigned value of 100. This number was then normalized by multiplying it by the weighted percentage assigned to the criterion and shown in Table XVIII. For example, the flat gasket Allpax 500 reliability evaluation was calculated as follows: First, the leakage rate a under the heading Reliability 1 was determined to be acceptable based on development test data; for this reason no points were deducted. Secondly, for 1.b, Experience,

TABLE XVIII

PERFORMANCE PARAMETER VALUE ASSESSMENT FOR GASKETS AND SEALS

Criteria for LOX Application	Flat Gasket, Allpax 500	Flat Gasket, Narmco	O-Ring	K-Seal	Flat Gasket, Johns-Manville
1. Reliability (0.40)					
a. Leakage Rate	Acceptable (-0)	Acceptable (-0)	Acceptable (-0)	Acceptable (-0)	
b. Experience, Test	Extensive experience. Maintains good sealing capability. (-20)	Extensive testing was done in laboratories. Only a few tests were run on flight hardware. (-20)	Not much experience. Maintained good sealing capability. (-20)	Maintained good sealing capability. (-10)	Extensive experience. Good sealing capability. (-10)
c. Experience, Flight	Extensive experience on Saturn IB. Design is favorable. (-10)	Two flights only had some gaskets. (-20)	Not favorable for LOX. (-50)	Extensive application in high pressure cryogenic applications (engines). (-10)	Lesser application on Saturn IB than Allpax. (-15)
Rating	70	60	50	80	75
Normalized	28.0	24.0	20.0	23.0	30.0
2. Performance (0.31)					
a. Leakage Rate	Acceptable (-0)	Acceptable (-0)	Acceptable (-0)	Acceptable (-0)	Acceptable (-0)
b. Experience, Test	Demonstrated good sealing capability throughout testing. (-10)	Demonstrated good sealing capability throughout testing. (-10)	Demonstrated good sealing capability throughout testing. (-10)	Demonstrated good sealing capability throughout testing. (-10)	Demonstrated good sealing capability throughout testing. (-10)
c. Experience, Flight	Information from flight measurements; no pressure loss. (-10)	Not much experience. (-20)	Extensive experience for fuel application. (-20)	Very good flight performance. (-10)	Information from flight measurements; no pressure loss. (-10)

TABLE XVIII

(Concluded)

Criteria for LOX Application	Flat Gasket, Allpax 500	Flat Gasket, Narmco	O-Ring	K-Seal	Flat Gasket, Johns-Manville
Rating	80	70	70	80	80
Normalized	24.8	21.7	21.7	24.8	24.8
3. Gasket Deformation Adaptability (0.29)					
a. Axial Separation	None (0.000 in.); High gasket loading required. (-30)	0.006 in.; High gasket loading required. (-15)	Very flexible; compensates 0.010 in. (-5)	Fair (0.004 in.). (-10)	None (0.000 in.); High gasket loading required. (-30)
b. Radial	Gasket tolerates 0.006-in. deflection. (-0)	Gasket tolerates 0.006-in. deflection. (-0)	Very flexible; compensates 0.010 in. (-5)	Not sensitive (0.010 in.) to radial deflection. (-5)	Gasket tolerates 0.006-in. deflection. (-0)
c. Maximum Distortion	None (0.000 in.). (-30)	None (0.000 in.). (-30)	Very good; tolerates 0.010 in. (-5)	Tolerates 0.001 in. in., which is fair. (-15)	None (0.000 in.). (-30)
Rating	40	55	85	70	40
Normalized	11.6	16.0	24.6	20.3	11.6
Total	64.6	61.7	66.3	77.1	66.6
Remarks	For LOX application this gasket has been used for many years. It is not acceptable for high pressure.	For LOX application the Narmco gasket has been newly developed. It has a potential future and will probably replace the Allpax gasket because of its superior properties.	The application of O-rings (Butyl) for cryogenics has not been used extensively but reports from the National Bureau of Standards in Boulder, Colorado, recommend their use.	The requirements for flange surface machining and the sensitivity to scratches makes the K-seal, besides its high rating, less preferable.	For LOX application this gasket has been used for many years. It is not acceptable for high pressure.

points were deducted from 100 based on qualification testing, and for 1.c, 10 points were deducted, based on stage static firing and flight data. A total of 30 points was deducted; therefore, 70 points is the total rating value for this evaluation.

The normalized rating was then calculated as follows:

$$\text{Reliability Weighting Percentage} \times \text{Gasket Rating} = \text{Normalized Rating.}$$

The result of this calculation is $0.40 \times 70 = 28$, which is the number shown as the normalized value. The same procedure is used for performance and gasket deformation adaptability and the result is shown as the total value, which is 64.6 for the Allpax 500 gasket.

Evaluations such as these can be performed in many different ways. The procedure explained above demonstrates one way of establishing a comparison matrix. If there are no historical data available, then probability numbers could be used to describe likelihoods of success. The numbers can be obtained simply by making an educated evaluation as to their values. Another way to determine reliability estimates is through a failure mode analysis, which is a widely used tool in decisionmaking. It reflects the importance of the parameters under consideration. In practice, these values are entered as percentages. With all these values established, the computation necessary to find the optimum separable connector system from a family of alternate configurations can be performed.

CHAPTER VII

THE SOLUTION TECHNIQUE

The Analytical Problem

The problem in this research is said to be dynamic; that is, the problem cannot be fully stated until some portion of the solution is available. The partial solution then provides data from which the problem can be stated. Problems having these dynamic qualities require a special effort to obtain a solution. The complexity of such problems needs exceptional care to organize and document the analysis to avoid confusion.

Consider the three-stage process as shown in Chapter IV, and replace the stages by subsystems where each subsystem consists of a certain number of configurations and each of these configurations has different values for the predetermined parameters. The problem to be solved is to determine that configuration which is the optimum for the whole system.

The technique using dynamic programming for solving optimization problems, developed by Neuner and Miller [13], was extended in this research to find the optimum separable connector for the LOX system. The symbols used in this technique are defined as follows:

P_1 is the objective system level parameter to be optimized, subject to specified constraints placed upon the remaining system level parameters,

P_2, P_3, ..., P_p. The parameters under consideration could be performance, weight, cost, schedule, etc. Any one of them, depending on its importance, could be the objective system level parameter; all others would be the remaining system level parameters.

p is the total number of parameters.

k could be any one of the parameters, where $k = 1, 2, \ldots, p$.

m is the total number of subsystems.

j could be any subsystem, with $j = 1, 2, \ldots, m$.

n_j is the total number of configurations pertaining to the jth subsystem.

i could be any one of the n configurations, where $i = 1, 2, \ldots, n$.

P_k is a system level parametric value which depends on all kth parameter values of all subsystems and all configurations.

$P_k(i, j)$ is the parametric value of the kth parameter for the ith configuration of the jth subsystem.

$Max_k(j)$ is the largest value of the parameter k of the subsystem j.

$Min_k(j)$ is the smallest value of the parameter k of the subsystem j.

Max_k is the maximum value of the system level parameter, which is the sum of all $Max_k(j)$ values.

Min_k is the minimum value of the system level parameter, which is the sum of all $Min_k(j)$ values.

R_k is the range between the Min_k and Max_k parametric values.

C_k represents a predetermined parametric value increment that is equally spaced over the range value R_k. Its value should be close to the actual parametric increment value. For example, if the weight of eight configurations differs by 10 to 15 lb for each consecutive configuration, then C_k should be chosen to be near 15 lb.

n_k is the total number of increments C_k.

ℓ is the total number of values within the range R_k divided by increments C_k is 0.00, C_k, $2C_k$, $3C_k$, ..., $n_k \cdot C_k$, where $\ell = n_k + 1$.

$PP_k = P_k(i, j) - Min_k(j)$. The constraint on the remaining parameter is obtained by subtracting each configuration parametric value of the kth parameter $P_k(i, j)$ from the lowest parametric value of the jth subsystem.

The Dynamic Program Algorithm

The problem to be solved is one of optimizing a system level objective parameter P_1 subject to the constraints placed upon the remaining system level parameters P_2, P_3, .., P_p. The objective system level parametric value P_1 is a function of all system level parametric values for all configurations and subsystems. The objective system level parameter can be additive or multiplicative.

The dynamic programming algorithm employed is based upon a recurrence relation; i.e., the solution of the problem can be obtained by an iterative procedure. The system variables used in the problem formulation are expressed in terms of each other, so that by following an iterative procedure, the variables will be eliminated one at a time. The variables can be numerical quantities or functions and are subject to constraints. The problem is structured by means of a network consisting of stages, where the output of one stage represents the input to the next stage. The procedure is continued until the last stage to be considered has been evaluated. If, for example, the minimum cost of the first subsystem has been determined subject to constraints such as weight and schedule, the cost of the next subsystem is determined by the minimum cost of the first subsystem. The quantity cost f_j for the jth subsystem is determined by the quantity cost $f_{(j-1)}$ of the $(j-1)$ subsystem, where $j = 1, 2, .., m$. The best attainable quantity for the objective system level parameter is obtained by determining the quantities f_j for all combinations of the remaining system level parametric values. These quantities are then tabulated in a payoff matrix.

There are essentially three types of matrices needed. One is established for each subsystem showing the actual parametric values, called the initial matrices. The second type of matrix shows the coded parametric values, and the third matrix is the parametric payoff matrix. This matrix shows all possible combinations of parameter value trade-offs for the whole system.

The Numerical Example

The Saturn IB first stage LOX system as shown in Figure 21 consists of five tanks, one large tank and four smaller outer tanks that are clustered around the inner tank. The tanks are connected by interconnect lines. Attached to these interconnect lines are eight suction lines that are connected at the aft end to feed the eight LOX turbines of the engines. These lines, valves, spacers, and couplings were assembled with approximately 90 separable bolted connectors.

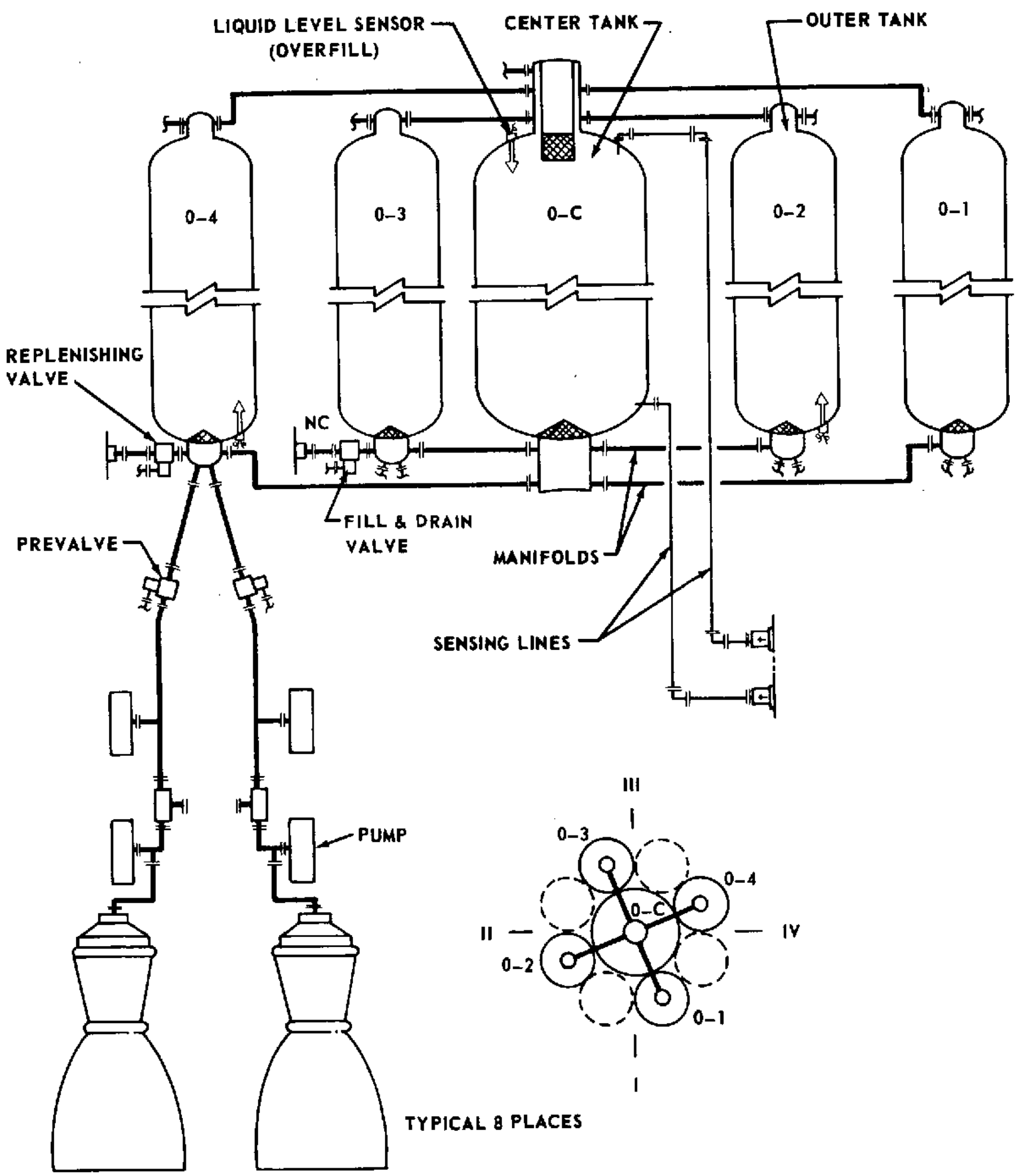


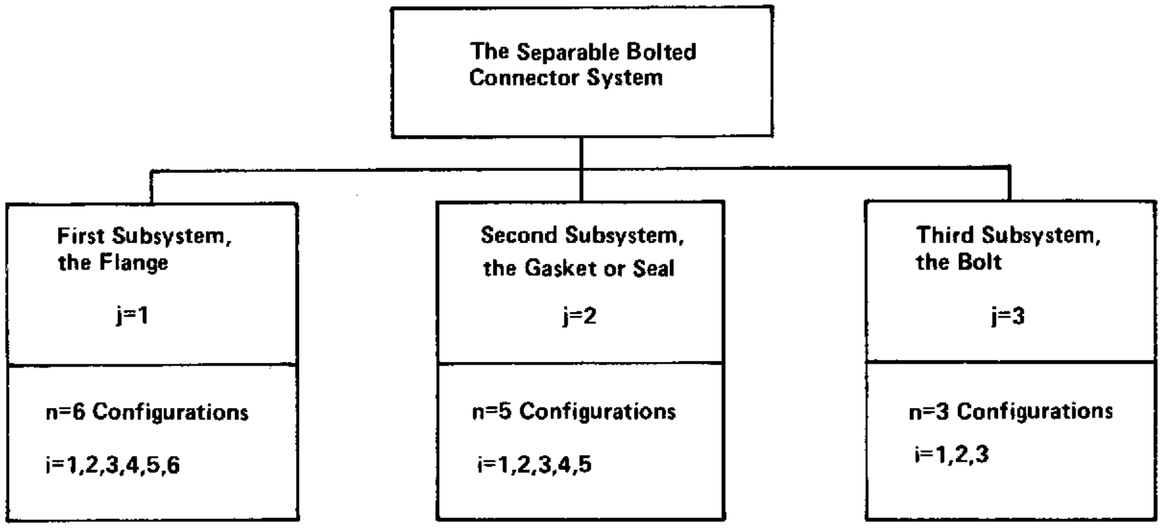
FIGURE 21. LOX SYSTEM SCHEMATIC.

An effort was made to find the optimum configuration among the many existing connector systems. To keep the numerical example from becoming too large, only six connector designs, four gaskets, one seal, and three bolt configurations were selected to be evaluated, which totaled ninety combinations of configurations. Three parameters were used for the analysis; namely, performance, weight, and cost. The reliability values were incorporated into the performance values to simplify the problem. The result of this optimum calculation was shown in a matrix that enabled the design engineer to select the connector design which meets the imposed criteria. For example, if performance is the primary criterion then this should be the system level parameter and a trade-off study can be made between weight and cost. Should weight be considered the primary criteria, then performance and cost should be used in the trade-off study.

The numerical example serves to demonstrate that smaller optimization problems can be solved numerically by the design engineer without the aid of a computer.

All parameters used in this analytical method maintained their natural content. The test and flight data substantiated the assumption that all connectors under investigation performed satisfactorily within a tolerable leakage rate range. Since geometric similarity exists among the selected separable connectors, scale factors were used for comparison purposes.

The LOX connector system under investigation is composed as shown in the following diagram.



The three parameters selected for the optimization computation were (1) performance P_1, (2) weight P_2, and (3) cost P_3. The system level parameter to be optimized was described to be the performance parameter P_1; therefore, the remaining parameters are P_2 (weight) and P_3 (cost). The performance parameter value P_1 is treated as a multiplicative parameter; P_2 and P_3 are treated as additive parameters.

The Simplicity Rating Matrix

The six flange configurations n_1 through n_6 shown in Table XIV were evaluated by the author first using the evaluation procedure in accordance with the simplicity rating system technique [12] described previously. The values in the matrix presented in conjunction with the discussion entitled "Component Simplicity Rating" are actual values established by the author and cognizant engineers. The resulting values are shown below.

Factor	DI[a]	n_1		n_2		n_3		n_4		n_5		n_6	
		f_w[b]	PI[c]	f_w	PI	f_w	PI	f_w	PI	f_w	PI	f_w	PI
A	3	1	3	1	3	2	6	1	3	1	3	1	3
B	3	3	9	3	9	1	3	1	3	3	9	1	3
C	2	3	6	3	6	3	6	3	6	3	6	2	4
D	1	1	1	1	1	2	2	1	1	1	1	2	2
Performance Index		19		19		17		13		19		12	

a. DI = Degree of Importance
b. f_w = Factor Weight
c. PI = Performance Index (The performance index values were used in support of the performance value assessment.)

The matrix above shows that the performance index values lie between 12 and 19, which represents 37 percent or 7 points difference between lower and upper extremes.

Preparation of the $P_k(i,j)$ Matrices

The next step in this analysis was the preparation of a matrix for each subsystem, showing the three parameter values for performance, weight, and cost. The performance parameter values for the flanges were calculated as discussed in the following.

For each flange the three dimensionless ratios t/ID, w/ID, and D_B/D_G were taken from Table XIV; then the ratios for the same flange diameter, operating pressure, and material were taken for t/ID from Figure 17, for w/ID from Figure 18, and for D_B/D_G from Figure 19. All curves shown on these figures represent the approximating curves fitted to the flange ratios that were obtained from 60 evaluated flange designs. The curves represent the average values and were used for base value references. For the determination of the performance value, one calculates the difference between figure value and table value and expresses the difference in percentage. For example, for configuration n_1, which is an 8.00-in. ID aluminum flange operating under 140-psi pressure, the ratios taken from Table XIV are as follows:

$$t/ID = 0.109,$$

$$w/ID = 0.173,$$

and

$$D_B/D_G = 1.140.$$

The ratios taken from Figures 17, 18, and 19 are

$$t/ID = 0.090,$$

$$w/ID = 0.190,$$

and

$$D_B/D_G = 1.095.$$

Then, the calculated differences between curve values and table values are

$$\Delta t/ID = 0.090 - 0.109 = -0.019 \ (-21.15 \text{ percent}),$$

$$\Delta w/ID = 0.190 - 0.173 = 0.017 \ (8.95 \text{ percent}),$$

and

$$\Delta D_B/D_G = 1.095 - 1.140 = -0.045 \quad (4.12 \text{ percent}) .$$

These three values show that flange n_1 is 21.15-percent thicker than average, 8.95-percent smaller than average, and the distance between bolt center and gasket center diameter is 4.12-percent wider than average. These values were used for the performance assessment of the flanges. Continuing the example for flange design n_1, one obtains

$$t/ID = 1.00 - 0.2115 = 0.7885 \text{ , rounded up } = 0.79 ,$$

$$w/ID = 1.00 - 0.00895 = 0.9105 \text{ , rounded down } = 0.91 ,$$

and

$$D_B/D_G = 1.0 - 0.0412 = 0.9588 \text{ , rounded up } = 0.96 .$$

The calculation of the flange performance was determined to be the product of the calculated ratio values with the performance index value PI from the simplicity rating added,

$$P_k(i, j) = t/ID \times w/ID \times D_B/D_G + PI .$$

The performance index value was assessed by subtracting the actual value from the lowest value in the simplicity rating matrix presented under the topic "Component Simplicity Rating" which is 19 for flange n_1 ; $19 - 12 = 7$. The number 7 was added directly in percentage to the subjective performance parameter,

$$P_1(1, 1) = (0.79 \times 0.91 \times 0.96) + 0.07 = 0.69 + 0.07 = 0.76$$

or

$$P_1(1, 1) = 0.76 .$$

This process was continued for all flanges and the values were inserted into the first row of the $P_k(i, 1)$ which follows.

The next step was to determine the weight figures of the six flanges; these were taken from Table VI and inserted into the second row of the $P_k(i, 1)$ matrix. The last step needed for the $P_k(i, 1)$ matrix was the establishment of the cost figures for each of the flanges. The costs of flanges shown in Table XII were established in support of this research by True Dimensions, Inc., Huntsville, Alabama, [9] in order to have some basic realistic cost values. The cost for each flange is calculated in three steps: the basic cost is taken from Table XII; the additional cost for a flange facing other than flat is taken from Figure 13; and the cost for surface finish other than 125 rms microinches is taken from Figure 14. For flange n_1 the basic cost taken from Table XII is \$62.00, for 8.00-inch ID and 0.872-inch thickness. Flange n_1 has a machined groove; therefore, an additional cost taken from Figure 13 of \$26.73 must be added to the basic cost. The surface finish of the flange is 32 rms microinches, and, from Figure 14, this cost is determined to be \$12.30. The total cost for flange n_1 is therefore $P_3(i, 1) =$ \$62.00 + \$26.73 + \$12.30 = \$101.03, for an order of 800 flanges. This value was then inserted into the third row of the $P_k(i, 1)$ matrix. With all other values inserted, the $P_k(i, 1)$ matrix for the first subsystem ($j = 1$), i.e., the flange, is complete.

$P_k(i,1)$	$i \backslash n_1$	n_2	n_3	n_4	n_5	n_6
P_1	0.76	0.83	0.80	0.69	0.83	0.95
P_2	3.97	3.30	3.39	4.25	3.92	3.62
P_3	101.03	73.30	109.10	97.23	68.50	97.40

The $P_k(i, 2)$ matrix for the second subsystem $(j = 2)$, the gasket or seal, was established in the following manner. The performance parameter values were taken from Table XVIII for n_1, a Narmco gasket; for n_2, an Allpax gasket; for n_3, an O-ring; for n_4, a Johns-Manville gasket, and for n_5 a K-seal. The assessed performance values $P_1(i, 2)$ were expressed in percentages, rounded up, and inserted into the P_1 row of the $P_k(i, 2)$ matrix. Then the weights, measured in pounds and obtained from drawing parts lists, were inserted into the second row, P_2. The third step was to insert the costs taken from Table XIII into the third row, P_3, of the matrix. With all values available, the $P_k(i, 2)$ matrix for the second subsystem $(j = 2)$ was completed as shown below.

$P_k(i,2)$	$i \backslash n_1$	n_2	n_3	n_4	n_5
P_1	0.62	0.65	0.66	0.67	0.77
P_2	0.03	0.04	0.02	0.04	0.07
P_3	20.50	4.00	6.00	5.60	27.00

The matrix for the third subsystem $(j = 3)$, the bolt, was established in a manner similar to the other two matrices. The performance parameter values were assessed based on the tensile strength of the material. The following data were obtained from Standard Press Steel Co., Jenkintown, Pennsylvania. For the first bolt configuration, n_1, an MS-35-308, the tensile stress was 65,000 psi, the weight for eight bolts was 0.5 pounds, and the cost for eight bolts was \$40.50. The second bolt configuration, n_2, was a 12-point high strength bolt with a tensile stress of 70,000 psi; the weight for eight bolts was 0.54 pounds, and the cost for eight bolts was \$40.50. The third bolt configuration, n_3, was an NAS-624 bolt with a tensile strength of 90,000 psi; the weight for eight bolts was 0.72 pounds, and the cost for eight bolts was \$79.20. The matrix for the $P_k(i,3)$ subsystem is shown below.

$P_k(i,3)$	i / n_1	n_2	n_3
P_1	0.65	0.70	0.90
P_2	0.50	0.54	0.72
P_3	40.50	40.50	79.20

The next step to be performed was the calculation of the range R_2 of the weight. Using the values of the three established matrices, $P_k(i,1)$, $P_k(i,2)$, and $P_k(i,3)$, R_2 was calculated by subtracting the sum of the lightest configuration of each subsystem from the sum of the heaviest configuration of each subsystem. The low weight total represents the lightest separable connector system; the high weight total represents the weight of the heaviest total separable connector system.

$$R_2 = Max_2 - Min_2 ,$$

where

$$\text{Max}_2 = \sum_{j=1}^{3} \text{Max}(j)$$

$$\text{Max}_2 = 4.25 \text{ lb (flange } n_4) + 0.07 \text{ lb (gasket } n_5) + 0.72 \text{ lb (bolt } n_3),$$

$$\text{Max}_2 = 5.04 \text{ lb (weight of the heaviest separable connector system)},$$

$$\text{Min}_2 = 3.30 \text{ lb (flange } n_2) + 0.02 \text{ lb (gasket } n_3) + 0.50 \text{ lb (bolt } n_1),$$

and

$$\text{Min}_2 = 3.82 \text{ lb (weight of the lightest separable connector system)}.$$

The range $R_2 = 5.04 - 3.82 = 1.22$ lb.

The cost range R_3 was calculated in a manner similar to the calculation of R_2:

$$R_3 = \text{Max}_3 - \text{Min}_3 ,$$

where

$$\text{Max}_3 = \sum_{j=1}^{3} \text{Max}(j)$$

$$\text{Max}_3 = \$109.10 \text{ (flange } n_3) + \$27.00 \text{ (gasket } n_5) + \$79.20 \text{ (bolt } n_3),$$

$$\text{Max}_3 = \$215.30 \text{ (most expensive separable connector system)},$$

$$\text{Min}_3 = \sum_{j=1}^{3} \text{Min}(j)$$

$$\text{Min}_3 = \$68.50 \text{ (flange } n_5) + \$4.00 \text{ (gasket } n_2) + \$40.50 \text{ (bolt } n_1 \text{ or } n_2),$$

and

$$\text{Min}_3 \ \$113.00 \text{ (least expensive separable connector system)}.$$

The range R_3 = \$215.30 - \$113.00 = \$102.30.

The next value to be calculated was $C_{\ell 2}$, the weight increment value which was determined by dividing the weight range R_2 = 1.22 pound into a number n_k of equal weight increments necessary for the analysis. The value of the increments was selected to be as small as the average increment value of the actual weight differences measured among the configurations under consideration. If, for example, the average weight difference from one connector system to the next heavier was 0.5 pound, then $C_{\ell 2}$ should be chosen as 0.5 pound or as close as possible. If $C_{\ell 2}$ is smaller than the actual average weight increase among the configurations, a great number of unnecessary calculations have to be made; if $C_{\ell 2}$ is greater than the actual average weight difference, the accuracy of the evaluation becomes unacceptable.

The weight increment $C_{\ell 2}$ was calculated as follows. First the number of intervals of the range R_2 was selected to be $n_k = 5$, because increment value $C_{\ell 2}$ was supposed to be near 0.25. The calculated $C_{\ell 2}$ was $1.22/5 = 0.244$ pound. This value represents the weight increase beginning from $\text{Min}_2 = 3.82$ pounds, which is the lowest weight of the whole separable connector system. The next heavier system weight is $3.82 + 0.244 = 4.064$ pounds. This is continued until the maximum weight $\text{Max}_2 = 5.04$ pounds is obtained. The index ℓ represents the number of steps beginning at the lowest weight and ending at the maximum weight. In the example there are $\ell_2 = 6$ steps, which are 0.0, 0.244, 0.488, 0.732, 0.976, and 1.220.

The first step (0.0) represents Min_2, and the last step Max_2 is 5.04 - 3.82 = 1.220.

The cost increment value $C_{\ell 3}$ was calculated in a similar way to the weight increment with R_3 = \$102.30; the number of increments was selected to be $n_k = 6$, and the number of steps $\ell_3 = 7$, so that $C_{\ell 3}$ = \$102.30/6 = \$17.05. The whole range R_3 then is composed of the following increments: 0.0, 17.05, 34.10, 51.15, 68,20, 85.25, and 102.30. To simplify the calculation, data from all three matrices $P_k(i,1)$, $P_k(i,2)$, and $P_k(i,3)$ were coded, and the new matrices were established, showing only the remaining parameter values P_2 and P_3.

<u>First Subsystem (j = 1)</u>

$P_k(i,1)$	i / n_1	n_2	n_3	n_4	n_5	n_6
P_2	0.67	0.00	0.09	0.95	0.62	0.32
P_3	32.53	4.80	40.60	28.73	0.00	28.90

<u>Second Subsystem (j = 2)</u>

$P_k(i,2)$	i / n_1	n_2	n_3	n_4	n_5
P_2	0.01	0.02	0.00	0.02	0.05
P_3	16.50	0.00	2.00	1.60	23.00

Third Subsystem $(j = 3)$

$P_k(i, 3)$ \ i	n_1	n_2	n_3
P_2	0.00	0.04	0.22
P_3	0.00	0.00	38.70

Using the three matrices with coded weight and cost values, Table XIX, the performance payoff matrix, was established in the following steps:

Step One. The six steps for the R_2 weight range were inserted in column 1, and each step ℓ_2 was repeated as many times as there are steps in range R_3 (cost) which is a total of $6 \times 7 = 42$ combinations. The corresponding $C_{\ell k}$ increments for weight were inserted in column 3 and the cost was inserted in column 4. For example, the increment value C_ℓ for first step $\ell_2 = 1$ (weight) is 0.00; therefore, all increment values for ℓ_2, first step, are 0.00. The increment values $C_{\ell 3}$ for the first step $\ell_3 = 1$ (cost) is $C_{\ell 3} = 0.00$. For the second step $\ell_3 = 2$, $C_{\ell 3} = \$17.05$. For the third step $\ell_3 = 3$, $C_{\ell 3} = \$34.10$. For the fourth step $\ell_3 = 4$, $C_{\ell 3} = \$51.15$. This process is continued until all $C_{\ell 3}$ values are inserted into the matrix.

Step Two. There are four columns occupied for the first sybsystem, the flange $(j = 1)$. Column 5 shows that configuration number whose value is closest to the weight figure $C_{\ell 2}$. In the example, looking at the coded $P_k(i, 1)$ matrix, the second configuration n_2 has the value of 0.00 pounds; therefore, configuration n_2 is shown in column 5 of the matrix. Similarly, in column

TABLE XIX

MAXIMUM PERFORMANCE PAYOFF MATRIX

ℓ_k		$C_{\ell k}$		Flange (j = 1)				Gasket (j = 2)				Bolt (j = 3)				P_1 Optimum System	P_1 Optimum System (Rounded Off)
ℓ_2	ℓ_3	$C_{\ell 2}$	$C_{\ell 3}$	i_2	i_3	Selected Configuration	$P_1(i,1)$ Selected Configuration	i_2	i_3	Selected Configuration	$P_1(i,2)$ Selected Configuration	i_2	i_3	Selected Configuration	$P_1(i,3)$ Selected Configuration		
1	1	0.0	0.00	n_2	n_5	n_5	0.83	n_3	n_2	n_3	0.66	n_1	n_2	n_2	0.70	0.38346	0.38
1	2		17.05	n_2	n_2	n_2	0.83	n_3	n_1	n_3	0.66	n_1	n_3	n_3	0.90	0.49302	0.49
1	3		34.10	n_2	n_1	n_2	0.83	n_3	n_5	n_5	0.77	n_1	n_3	n_3	0.90	0.57519	0.58
1	4		51.15	n_2	n_3	n_2	0.83	n_3	n_5	n_5	0.77	n_1	n_3	n_3	0.90	0.57519	0.58
1	5		68.20	n_2	n_3	n_2	0.83	n_3	n_5	n_5	0.77	n_1	n_3	n_3	0.90	0.57519	0.58
1	6		85.25	n_2	n_3	n_2	0.83	n_3	n_5	n_5	0.77	n_1	n_3	n_3	0.90	0.57519	0.58
1	7		102.30	n_2	n_3	n_2	0.83	n_3	n_5	n_5	0.77	n_1	n_3	n_3	0.90	0.57519	0.58
2	1	0.244	0.00	n_3	n_5	n_5	0.83	n_5	n_2	n_5	0.77	n_3	n_2	n_3	0.90	0.57519	0.58
2	2		17.05	n_3	n_2	n_2	0.83	n_5	n_1	n_5	0.77	n_3	n_3	n_3	0.90	0.57519	0.58
2	3		34.10	n_3	n_1	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.55440	0.55
2	4		51.15	n_3	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.55440	0.55
2	5		68.20	n_3	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.55440	0.55
2	6		85.25	n_3	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.55440	0.55
2	7		102.30	n_3	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.55440	0.55
3	1	0.488	0.00	n_6	n_5	n_6	0.95	n_5	n_2	n_5	0.77	n_3	n_2	n_3	0.90	0.65835	0.66
3	2		17.05	n_6	n_2	n_6	0.95	n_5	n_1	n_5	0.77	n_3	n_3	n_3	0.90	0.65835	0.66
3	3		34.10	n_6	n_1	n_6	0.95	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.65835	0.66
3	4		51.15	n_6	n_3	n_6	0.95	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.65835	0.66
3	5		68.20	n_6	n_3	n_6	0.95	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.65835	0.66
3	6		85.25	n_6	n_3	n_6	0.95	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.65835	0.66
3	7		102.30	n_6	n_3	n_6	0.95	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.65835	0.66

TABLE XIX

(Concluded)

ℓ_k		$C_{\ell k}$		Flange (j = 1)				Gasket (j = 2)				Bolt (j = 3)					
ℓ_2	ℓ_3	$C_{\ell 2}$	$C_{\ell 3}$	i_2	i_3	Selected Configuration	$P_1(i,1)$ Selected Configuration	i_2	i_3	Selected Configuration	$P_1(i,2)$ Selected Configuration	i_2	i_3	Selected Configuration	$P_1(i,3)$ Selected Configuration	P_1 Optimum System	P_1 Optimum System (Rounded Off)
4	1	0.732	0.00	n_1	n_5	n_5	0.83	n_5	n_2	n_5	0.77	n_3	n_2	n_3	0.90	0.57519	0.58
4	2	↓	17.05	n_1	n_2	n_2	0.83	n_5	n_1	n_5	0.77	n_3	n_3	n_3	0.90	0.57519	0.58
4	3	↓	34.10	n_1	n_1	n_1	0.76	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.52668	0.53
4	4	↓	51.15	n_1	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.55450	0.55
4	5	↓	68.20	n_1	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.55450	0.55
4	6	↓	85.25	n_1	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.55450	0.55
4	7	↓	102.30	n_1	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.55450	0.55
5	1	0.976	0.00	n_4	n_5	n_5	0.83	n_5	n_2	n_5	0.77	n_3	n_2	n_3	0.90	0.57519	0.58
5	2	↓	17.05	n_4	n_2	n_2	0.83	n_5	n_1	n_5	0.77	n_3	n_3	n_3	0.90	0.57519	0.58
5	3	↓	34.10	n_1	n_1	n_1	0.76	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.57519	0.58
5	4	↓	51.15	n_4	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.57519	0.58
5	5	↓	68.20	n_4	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.57519	0.58
5	6	↓	85.25	n_1	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.57519	0.58
5	7	↓	102.30	n_1	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.57519	0.58
6	1	1.220	0.00	n_4	n_5	n_5	0.83	n_5	n_2	n_5	0.77	n_3	n_2	n_3	0.90	0.57519	0.58
6	2	↓	17.05	n_4	n_2	n_2	0.83	n_5	n_1	n_5	0.77	n_3	n_3	n_3	0.90	0.57519	0.58
6	3	↓	34.10	n_4	n_1	n_1	0.76	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.57519	0.58
6	4	↓	51.15	n_4	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.57519	0.58
6	5	↓	68.20	n_4	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.57519	0.58
6	6	↓	85.25	n_4	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.57519	0.58
6	7	↓	102.30	n_4	n_3	n_3	0.80	n_5	n_5	n_5	0.77	n_3	n_3	n_3	0.90	0.57519	0.58

6, that configuration number is shown whose cost value is closest to the corresponding cost value $C_{\ell 3}$, 0.00, which is configuration n_5.

Step Three. Select from configuration i_2 and i_3 the one which has the higher performance value $P_1(i, 1)$ shown in the initial matrix $P_k(i, 1)$ and insert the number of the selected configuration into column 7 under Selected Configuration. The performance value $P_1(i, 1)$ of this configuration is then inserted into column 8 under $P_1(i, 1)$ Selected Configuration. Using the value taken from the initial $P_k(i, 1)$ matrix, the configuration n_2 has a performance value of $P_1(2, 1) = 0.83$ and configuration n_5, $P_1(5, 1) = 0.83$. In this case either n_2 or n_5 can be used. Configuration n_5 was chosen; this configuration number n_5 was inserted into column 7 and its performance value 0.83 was inserted into column 8 under $P_1(i, 1)$ Selected Configuration.

Steps two and three were repeated for all 42 combinations of weight $C_{\ell 2}$ and cost $C_{\ell 3}$ for the first subsystem, the flange. Likewise, steps two and three were carried out for the two other subsystems, the gasket and the bolt. The gasket values are shown in columns 9, 10, 11, and 12 and the bolt values in columns 14, 15, 16, and 17. Column 17 shows under P_1 Optimum System the product of the flange $P_1(i, 1)$ gasket $P_1(i, 2)$ and bolt $P_1(i, 3)$ performance values of the selected configurations. The figures from column 17 were rounded off and shown in column 18 under the same heading as column 17. These values represent the objective system level performance parameter values for all combinations of configurations and subsystems.

The performance payoff matrix, Table XIX, gives the user the possibility to choose that separable bolted connector configuration which best suits the imposed criteria. There are 42 combinations of different designs tabulated, each having the best attainable performance value. Trade-offs can be performed between weights and costs depending on the criteria. The weight range of the whole assembly varies between 3.82 and 5.04 pounds and the cost varies from $113.00 to $215.30, from which the selections can be made. If for example the highest performance that is requires is $P_1 = 0.66$ in the matrix and the weight must be a minimum, the following combination of configurations is suggested:

1. Flange configuration i_{max} is n_6.
2. Gasket configuration i_{max} is n_5.
3. Bolt configuration i_{max} is n_3.

The performance value $P_1 = 0.66$, the weight $P_2 = 3.820 + 0.488 = 4.308$ pounds, and the cost $P_3 = \$133.00$. This matrix can be used for any combination of the performance, weight, and cost parameters, depending on the established criteria; i.e., which parameters are important.

CHAPTER VIII

CONCLUSIONS AND RECOMMENDATIONS

This research illustrates the usefulness of Operations Research in determining the optimum design for a complex system based on the utilized parameters. It explains the pattern to be followed in simplifying the decision-making process. The algorithm utilized takes into account the interactions of individual components of the system. The result is given in quantities that may be of importance to design-oriented organizations. The computation can easily be accomplished by hand for small problems. This is of benefit where computer time is not readily available. Larger problems can easily be programmed for the computer. Great latitude is given the design engineer in assessing the parameter values. Depending on the application, the user of the procedure has many options for choosing the criteria best suited to his problem. Its broad application allows nearly any design decision to be performed and evaluated for its impact upon the variables that must be balanced to achieve system design optimization. The technique employed saves many engineering hours and results in true system design optimization. The result gives the design engineer the assurance that the product designed meets the imposed criteria to the maximum extent possible. It is the author's opinion that this effort is only a small step forward in demonstrating the usefulness of

Operations Research for making design decisions, and hopefully, in the future it will become a standard tool for use by design-oriented organizations. It is the author's intention to find a way to improve the parameter value assessment, especially for performance of systems where the values cannot be measured directly in standardized units. It is also intended to improve the assessment of parameter values where complicated interactions exist between the system and the environments.

If this algorithm had been used before hardware was built for the Saturn IB and Saturn V, an enormous amount of money would have been saved, particularly that spent for testing so many different configurations used for identical systems. Also the performance levels would have been increased and the weights would have been remarkably reduced. Use of the algorithm is strongly recommended in determining the optimum design in many fields, such as aeronautics, construction, electronics, etc. The author is presently involved in the application of this method to Stratoscope III and Sortie Laboratory design optimization. The result will be published in early 1973. These optimization studies will be very simple, using a computer while keeping the paper volume down to an absolute minimum.

APPENDIX

THE LOW PROFILE TAYLOR FORGE FLANGE STRESS COMPARISON ANALYSES

The result of the stress analyses for the Taylor Forge lightweight configuration (Fig. 11) and the low profile configuration (Fig. 12) are shown in the following order:

1. The design numbers 1001 through 1003 were assigned for the Taylor Forge lightweight analyses, where the design number 1001 in Table A-I represents the stress data for a connector equipped with an Allpax 500 gasket, design number 1002 in Table A-II represents a connector equipped with a Butyl gasket, and design number 1003 in Table A-III represents a connector equipped with a steel gasket.

2. Numbers 2001 through 2003 were assigned to the low profile flange stress analyses. Design number 2001 in Table A-IV gives the analysis data for the connector equipped with an Allpax 500 gasket, design number 2002 in Table A-V represents the connector equipped with a Butyl gasket, and design number 2003 in Table A-VI represents the connector equipped with a steel gasket.

The flange stresses were calculated in five equally spaced points which were distributed along the inner flange wall between the flange facing and the flange-pipe interface.

The stress ratios represent the quotient of the allowable stress divided by the calculated stress, which means that all values above 1.0 exceed the allowable value. To compare the two flange configurations, a new ratio was established to measure the differences between these two flange configurations, which is the quotient of the low profile stress ratio divided by the Taylor Forge stress ratio. Stress calculations were performed for each different gasket and each hubstation, including the flange ring:

$$R = \sigma_{LP}/\sigma_{TF} \quad ,$$

where

LP = Low Profile

TF = Taylor Forge.

The values obtained are shown below.

Hubstation	Allpax 500	Narmco	Steel
1	0.31	0.33	0.472
2	0.48	0.49	0.64
3	0.40	0.42	0.67
4	0.38	0.40	0.78
5	0.31	0.34	0.89
Flange	0.28	0.29	0.39

These ratios clearly demonstrate the better performance characteristics of the low profile flange. Unfortunately the low profile flange was developed after all Saturn IB stages were built and therefore are not included in this optimization study. However, the low profile flange stress values were used to interprete the curves in Figures 17, 18, and 19, which show the changing dimension ratios plotted over the different flange inner diameters of typical applications.

TABLE A-I

TAYLOR FORGE LIGHTWEIGHT FLANGE STRESS DATA (ALLPAX GASKET)

```
          ****************************************************************
                    FLANGED CONNECTOR DESIGN PROGRAM
          ****************************************************************

DESIGN NUMBER  1001

          INTEGRAL FLANGED CONNECTOR WITH NO CONTACT OUTSIDE BOLT CIRCLE
          ****************************************************************


          * INPUT *
          *********


   DI     =         8.0000000     INSIDE PIPE DIAMETER (INCHES)

   DG     =         8.5000000     GASKET CIRCLE DIAMETER (INCHES)

   TP(1)  =          .0620000     LEFT PIPE THICKNESS (INCHES)
   TP(2)  =          .0620000     RIGHT PIPE THICKNESS (INCHES)

   GFI    =      1850.0000        GASKET FORCE REQUIRED TO SEAL (LBS/IN)

   GW     =          .6000000     GASKET WIDTH (INCHES)
   HG     =          .0620000     GASKET THICKNESS (INCHES)

   MU(1)  =          .6000000     COEFFICIENT OF FRICTION BETWEEN LEFT FLANGE AND GASKET
   MU(2)  =          .5000000     COEFFICIENT OF FRICTION BETWEEN RIGHT FLANGE AND GASKET

   V(1)   =          .2800000     RATIO OF POISSON FOR THE LEFT FLANGE MATERIAL
   V(2)   =          .2800000     RATIO OF POISSON FOR THE RIGHT FLANGE MATERIAL

   NCOND  =  5            TOTAL NUMBER OF CONDITIONS ,INITIAL AND OPERATING

   OPT    =  2            CALCULATE STRESSES FOR GIVEN DESIGN

   LIMIT  =   1           MAXIMUM NUMBER OF ITERATIONS TO BE DONE
```

TABLE A-I

(Concluded)

DESIGN NUMBER 1001

INTEGRAL FLANGED CONNECTOR WITH NO CONTACT OUTSIDE BOLT CIRCLE
****

MAXIMUM STRESS RATIO AND ASSOCIATED CONDITION

	LEFT SIDE		RIGHT SIDE	
	STRESS RATIO	CONDITION	STRESS RATIO	CONDITION
HUB STATION 1	1.1070175	5	1.1070158	5
HUB STATION 2	.6863136	5	.6863128	5
HUB STATION 3	.5447113	5	.5447107	5
HUB STATION 4	.4446284	5	.4446279	5
HUB STATION 5	.4162414	5	.4162408	5
FLANGE	.5947808	5	.5947795	5
BOLT	5.852982	5		

MINIMUM GASKET LOAD = 1149.3 (LBS/IN) 1

TABLE A-II

TAYLOR FORGE LIGHTWEIGHT FLANGE STRESS DATA (BUTYL GASKET)

```
*************************************************************
          FLANGED CONNECTOR DESIGN PROGRAM
*************************************************************

DESIGN NUMBER  1002

         INTEGRAL FLANGED CONNECTOR WITH NO CONTACT OUTSIDE BOLT CIRCLE
         *************************************************************

         * INPUT *
         *********

   DI     =        8.0000000    INSIDE PIPE DIAMETER (INCHES)

   DG     =        8.5000000    GASKET CIRCLE DIAMETER (INCHES)

   TP(1)  =         .0620000    LEFT PIPE THICKNESS (INCHES)
   TP(2)  =         .0620000    RIGHT PIPE THICKNESS (INCHES)

   GFI    =      1850.0000      GASKET FORCE REQUIRED TO SEAL (LBS/IN)

   GW     =         .5000000    GASKET WIDTH (INCHES)
   HG     =         .0620000    GASKET THICKNESS (INCHES)

   MU(1)  =         .1200000    COEFFICIENT OF FRICTION BETWEEN LEFT FLANGE AND GASKET
   MU(2)  =         .1200000    COEFFICIENT OF FRICTION BETWEEN RIGHT FLANGE AND GASKET

   V(1)   =         .2600000    RATIO OF POISSON FOR THE LEFT FLANGE MATERIAL
   V(2)   =         .2800000    RATIO OF POISSON FOR THE RIGHT FLANGE MATERIAL

   NCOND  =  5          TOTAL NUMBER OF CONDITIONS ,INITIAL AND OPERATING

   OPT    =  2          CALCULATE STRESSES FOR GIVEN DESIGN

   LIMIT  =   1         MAXIMUM NUMBER OF ITERATIONS TO BE DONE
```

TABLE A-II

(Concluded)

DESIGN NUMBER 1002

INTEGRAL FLANGED CONNECTOR WITH NO CONTACT OUTSIDE BOLT CIRCLE
●●●

MAXIMUM STRESS RATIO AND ASSOCIATED CONDITION

	LEFT SIDE		RIGHT SIDE	
	STRESS RATIO	CONDITION	STRESS RATIO	CONDITION
HUB STATION 1	.9652899	5	.9652897	5
HUB STATION 2	.5972980	5	.5972979	5
HUB STATION 3	.4725731	5	.4725730	5
HUB STATION 4	.3828907	5	.3828907	5
HUB STATION 5	.3546338	5	.3546337	5
FLANGE	.5063924	5	.5063922	5
BOLT	5.028572	5		

MINIMUM GASKET LOAD = 1149.3 (LBS/IN) 1

TABLE A-III

TAYLOR FORGE LIGHTWEIGHT FLANGE STRESS DATA (STEEL GASKET)

```
************************************************************
          FLANGED CONNECTOR DESIGN PROGRAM
************************************************************

DESIGN NUMBER  1003

         INTEGRAL FLANGED CONNECTOR WITH NO CONTACT OUTSIDE BOLT CIRCLE
         ************************************************************

         * INPUT *
         *********

   DI     =         8.0000000   INSIDE PIPE DIAMETER (INCHES)

   DG     =         8.5000000   GASKET CIRCLE DIAMETER (INCHES)

   TP(1)  =          .0620000   LEFT PIPE THICKNESS (INCHES)
   TP(2)  =          .0620000   RIGHT PIPE THICKNESS (INCHES)

   GFI    =     13000.0000      GASKET FORCE REQUIRED TO SEAL (LBS/IN)

   GW     =          .5000000   GASKET WIDTH (INCHES)
   HG     =          .0620000   GASKET THICKNESS (INCHES)

   MU(1)  =          .3000000   COEFFICIENT OF FRICTION BETWEEN LEFT FLANGE AND GASKET
   MU(2)  =          .3000000   COEFFICIENT OF FRICTION BETWEEN RIGHT FLANGE AND GASKET

   V(1)   =          .2600000   RATIO OF POISSON FOR THE LEFT FLANGE MATERIAL
   V(2)   =          .2600000   RATIO OF POISSON FOR THE RIGHT FLANGE MATERIAL

   NCOND  =   5         TOTAL NUMBER OF CONDITIONS ,INITIAL AND OPERATING

   OPT    =   2         CALCULATE STRESSES FOR GIVEN DESIGN

   LIMIT  =    1        MAXIMUM NUMBER OF ITERATIONS TO BE DONE
```

TABLE A-III

(Concluded)

DESIGN NUMBER 1003

INTEGRAL FLANGED CONNECTOR WITH NO CONTACT OUTSIDE BOLT CIRCLE
****

MAXIMUM STRESS RATIO AND ASSOCIATED CONDITION

	LEFT SIDE		RIGHT SIDE	
	STRESS RATIO	CONDITION	STRESS RATIO	CONDITION
HUB STATION 1	.3992776	3	.3992776	3
HUB STATION 2	.2405245	3	.2405245	3
HUB STATION 3	.1830164	3	.1830164	3
HUB STATION 4	.1308459	3	.1308459	3
HUB STATION 5	.0973798	5	.0973798	5
FLANGE	.1454508	5	.1454508	5
BOLT	1.392632	1		

MINIMUM GASKET LOAD = 1149.3 (LBS/IN) 1

TABLE A-IV

LOW PROFILE FLANGE STRESS DATA (ALLPAX GASKET)

```
************************************************************
          FLANGED CONNECTOR DESIGN PROGRAM
************************************************************

DESIGN NUMBER  2001

       INTEGRAL FLANGED CONNECTOR WITH NO CONTACT OUTSIDE BOLT CIRCLE
       ************************************************************


       * INPUT *
       *********


   DI     =        8.0000000   INSIDE PIPE DIAMETER (INCHES)

   DG     =        8.2810000   GASKET CIRCLE DIAMETER (INCHES)

   TP(1)  =         .0620000   LEFT PIPE THICKNESS (INCHES)
   TP(2)  =         .0620000   RIGHT PIPE THICKNESS (INCHES)

   GF1    =      921.3000      GASKET FORCE REQUIRED TO SEAL (LBS/IN)

   GW     =         .2490000   GASKET WIDTH (INCHES)
   HG     =         .0620000   GASKET THICKNESS (INCHES)

   MU(1)  =         .5000000   COEFFICIENT OF FRICTION BETWEEN LEFT FLANGE AND GASKET
   MU(2)  =         .5000000   COEFFICIENT OF FRICTION BETWEEN RIGHT FLANGE AND GASKET

   V(1)   =         .2800000   RATIO OF POISSON FOR THE LEFT FLANGE MATERIAL
   V(2)   =         .2800000   RATIO OF POISSON FOR THE RIGHT FLANGE MATERIAL

   NCOND  =  5         TOTAL NUMBER OF CONDITIONS ,INITIAL AND OPERATING

   OPT    =  2         CALCULATE STRESSES FOR GIVEN DESIGN

   LIMIT  =  1         MAXIMUM NUMBER OF ITERATIONS TO BE DONE
```

TABLE A-IV

(Concluded)

DESIGN NUMBER 2001

INTEGRAL FLANGED CONNECTOR WITH NO CONTACT OUTSIDE BOLT CIRCLE
****

MAXIMUM STRESS RATIO AND ASSOCIATED CONDITION

	LEFT SIDE		RIGHT SIDE	
	STRESS RATIO	CONDITION	STRESS RATIO	CONDITION
HUB STATION 1	.3457204	5	.3457180	5
HUB STATION 2	.3272729	5	.3272704	5
HUB STATION 3	.2193662	5	.2193648	5
HUB STATION 4	.1680179	5	.1680172	5
HUB STATION 5	.1300242	5	.1300238	5
FLANGE	.1650827	5	.1650804	5
BOLT	6.411703	5		

MINIMUM GASKET LOAD = 977.9 (LBS/IN)

TABLE A-V

LOW PROFILE FLANGE STRESS DATA (BUTYL GASKET)

```
        ************************************************************
                  FLANGED CONNECTOR DESIGN PROGRAM
        ************************************************************

DESIGN NUMBER  2002

        INTEGRAL FLANGED CONNECTOR WITH NO CONTACT OUTSIDE BOLT CIRCLE
        ************************************************************


        * INPUT *
        *********


   DI     =          8.0000000    INSIDE PIPE DIAMETER (INCHES)


   DG     =          8.2810000    GASKET CIRCLE DIAMETER (INCHES)


   TP(1)  =           .0620000    LEFT PIPE THICKNESS (INCHES)
   TP(2)  =           .0620000    RIGHT PIPE THICKNESS (INCHES)


   GFI    =        921.3000       GASKET FORCE REQUIRED TO SEAL (LBS/IN)


   GW     =           .2490000    GASKET WIDTH (INCHES)
   HG     =           .0620000    GASKET THICKNESS (INCHES)


   MU(1)  =           .1200000    COEFFICIENT OF FRICTION BETWEEN LEFT FLANGE AND GASKET
   MU(2)  =           .1200000    COEFFICIENT OF FRICTION BETWEEN RIGHT FLANGE AND GASKET


   V(1)   =           .2800000    RATIO OF POISSON FOR THE LEFT FLANGE MATERIAL
   V(2)   =           .2800000    RATIO OF POISSON FOR THE RIGHT FLANGE MATERIAL


   NCOND  =  5            TOTAL NUMBER OF CONDITIONS ,INITIAL AND OPERATING


   OPT    =  2            CALCULATE STRESSES FOR GIVEN DESIGN


   LIMIT  =   1           MAXIMUM NUMBER OF ITERATIONS TO BE DONE
```

TABLE A-V

(Concluded)

DESIGN NUMBER 2002

INTEGRAL FLANGED CONNECTOR WITH NO CONTACT OUTSIDE BOLT CIRCLE

****

MAXIMUM STRESS RATIO AND ASSOCIATED CONDITION

	LEFT SIDE		RIGHT SIDE	
	STRESS RATIO	CONDITION	STRESS RATIO	CONDITION
HUB STATION 1	.3145494	5	.3145482	5
HUB STATION 2	.2934878	5	.2934866	5
HUB STATION 3	.1984392	5	.1984385	5
HUB STATION 4	.1553340	5	.1553336	5
HUB STATION 5	.1208106	5	.1208104	5
FLANGE	.1438625	5	.1438613	5
BOLT	5.776687	5		

MINIMUM GASKET LOAD = 977.9 (LBS/IN)

TABLE A-VI

LOW PROFILE FLANGE STRESS DATA (STEEL GASKET)

```
        ************************************************************
                  FLANGED CONNECTOR DESIGN PROGRAM
        ************************************************************

DESIGN NUMBER  2003

        INTEGRAL FLANGED CONNECTOR WITH NO CONTACT OUTSIDE BOLT CIRCLE
        ************************************************************


        * INPUT *
        *********


    DI     =          8.0000000    INSIDE PIPE DIAMETER (INCHES)


    DG     =          8.2810000    GASKET CIRCLE DIAMETER (INCHES)


    TP(1)  =           .0620000    LEFT PIPE THICKNESS (INCHES)
    TP(2)  =           .0620000    RIGHT PIPE THICKNESS (INCHES)


    GFI    =       6474.0000       GASKET FORCE REQUIRED TO SEAL (LBS/IN)


    GW     =           .2490000    GASKET WIDTH (INCHES)
    HG     =           .0620000    GASKET THICKNESS (INCHES)


    MU(1)  =           .3000000    COEFFICIENT OF FRICTION BETWEEN LEFT FLANGE AND GASKET
    MU(2)  =           .3000000    COEFFICIENT OF FRICTION BETWEEN RIGHT FLANGE AND GASKET


    V(1)   =           .2800000    RATIO OF POISSON FOR THE LEFT FLANGE MATERIAL
    V(2)   =           .2800000    RATIO OF POISSON FOR THE RIGHT FLANGE MATERIAL


    NCOND  =  5            TOTAL NUMBER OF CONDITIONS ,INITIAL AND OPERATING


    OPT    =  2            CALCULATE STRESSES FOR GIVEN DESIGN


    LIMIT  =   1           MAXIMUM NUMBER OF ITERATIONS TO BE DONE
```

TABLE A-VI

(Concluded)

DESIGN NUMBER 2003

INTEGRAL FLANGED CONNECTOR WITH NO CONTACT OUTSIDE BOLT CIRCLE
****

MAXIMUM STRESS RATIO AND ASSOCIATED CONDITION

	LEFT SIDE		RIGHT SIDE	
	STRESS RATIO	CONDITION	STRESS RATIO	CONDITION
HUB STATION 1	.1894543	3	.1894543	3
HUB STATION 2	.1530604	3	.1530604	3
HUB STATION 3	.1224729	3	.1224729	3
HUB STATION 4	.1016451	3	.1016451	3
HUB STATION 5	.0807773	3	.0807773	3
FLANGE	.0543903	5	.0543903	5
BOLT	2.056902	5		

MINIMUM GASKET LOAD = 977.9 (LBS/IN)

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APPROVAL

AN ASSESSMENT OF SEPARABLE FLUID CONNECTOR SYSTEM PARAMETERS TO PERFORM A CONNECTOR SYSTEM DESIGN OPTIMIZATION STUDY

By Willibald Peter Prasthofer

The information in this report has been reviewed for security classification. Review of any information concerning Department of Defense or Atomic Energy Commission programs has been made by the MSFC Security Classification Officer. This report, in its entirety, has been determined to be unclassified.

This document has also been reviewed and approved for technical accuracy.

James E. Kingsbury
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