AD





Report 2249

# MILITARY PETROLEUM PIPELINE SYSTEMS

by

Wayne E. Studebaker

June 1978

Approved for public release; distribution unlimited.



U.S. ARMY MOBILITY EQUIPMENT
RESEARCH AND DEVELOPMENT COMMAND
FORT BELVOIR, VIRGINIA

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

# REPORT DOCUMENTATION PAGE

READ INSTRUCTIONS BEFORE COMPLETING FORM

| | |
|---|---|
| 1. REPORT NUMBER 2249 | 2. GOVT ACCESSION NO. |
| 3. RECIPIENT'S CATALOG NUMBER | |
| 4. TITLE (and Subtitle) MILITARY PETROLEUM PIPELINE SYSTEMS, | 5. TYPE OF REPORT & PERIOD COVERED Final rept., |
| | 6. PERFORMING ORG. REPORT NUMBER |
| 7. AUTHOR(s) Wayne E. Studebaker | 8. CONTRACT OR GRANT NUMBER(s) ~~DAAG53-76-C-0096~~ |
| 9. PERFORMING ORGANIZATION NAME AND ADDRESS U.S. Army Mobility Equipment Research and Development Command, ATTN: DRDME-GS Fort Belvoir, Virginia 22060 | 10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS 1G763726DK4104 |
| 11. CONTROLLING OFFICE NAME AND ADDRESS | 12. REPORT DATE June 1978 |
| | 13. NUMBER OF PAGES 254 |
| 14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office) MERADCOM-2249 | 15. SECURITY CLASS. (of this report) Unclassified |
| | 15a. DECLASSIFICATION/DOWNGRADING SCHEDULE |

6 9 10 16 17 04 11 14 12 252p.

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release; distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

| | |
|---|---|
| Pipe | Pump Stations |
| Pumps | Pump-Engine Assemblies |
| Pipelines | Pipeline Construction |
| Pipeline Systems | Petroleum Products Pipelines |
| Pipe Joining Techniques | Fuels Handling Equipment |

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The present Army capability to install, operate, and maintain petroleum product pipelines is examined in light of current commercial pipeline technology and projections of fuel consumption for combat units in the event of future hostilities. The objective of this investigation is to provide a measure of effectiveness for and determine the technical feasibility of alternative pipeline systems operating as subsystems in a large logistical system for distribution of fuels in a theater of operations during wartime conditions.

(Continued)



78 11 15 153

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE

UNCLASSIFIED

i

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

403160

78 11 15 15

(Block 20 Cont'd)

A broad array of pipe materials, pipe joining techniques, pumping equipment, ancillary pipeline components, and system designs are evaluated. The findings indicate that substantive improvement in the operational effectiveness of military pipeline can be achieved using aluminum pipe and self-latching mechanical couplings in lieu of the existing military standard lightweight steel pipe joined by grooved-end, split-ring mechanical couplings. High-speed, medium-duty, diesel-engine-driven pump units are recommended for all pipeline pump station applications. Flexible hoselines are not an efficient or cost effective means for transporting large volumes of fuel over long distances.

CONTENTS



Page [illegible] left blank intentionally

## ILLUSTRATIONS

ILLUSTRATIONS (Cont'd)

## ILLUSTRATIONS (Cont'd)

## TABLES

## TABLES (Cont'd)

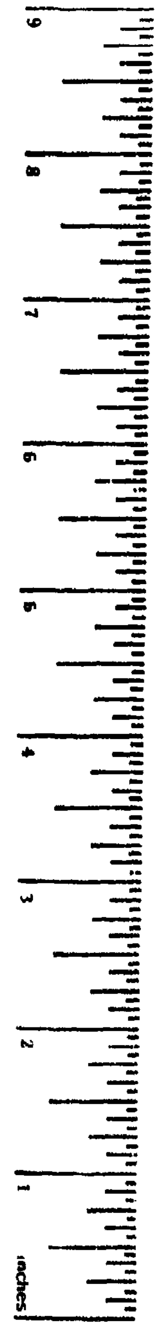
TABLES (Cont'd)

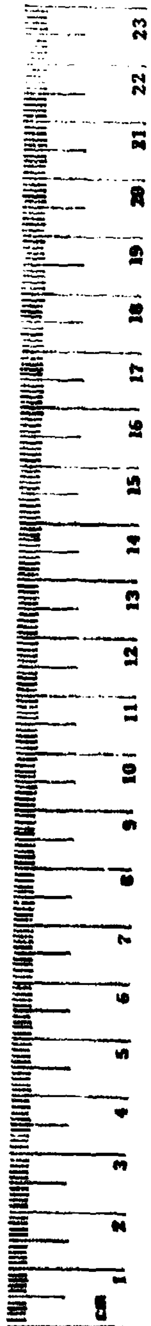
# METRIC CONVERSION FACTORS

## Approximate Conversions to Metric Measures

| Symbol | When You Know | Multiply by | To Find | Symbol |
|---|---|---|---|---|
| | | **LENGTH** | | |
| in | inches | *2.5 | centimeters | cm |
| ft | feet | 30 | centimeters | cm |
| yd | yards | 0.9 | meters | m |
| mi | miles | 1.6 | kilometers | km |
| | | **AREA** | | |
| $in^2$ | square inches | 6.5 | square centimeters | $cm^2$ |
| $ft^2$ | square feet | 0.09 | square meters | $m^2$ |
| $yd^2$ | square yards | 0.8 | square meters | $m^2$ |
| $mi^2$ | square miles | 2.6 | square kilometers | $km^2$ |
| | acres | 0.4 | hectares | ha |
| | | **MASS (weight)** | | |
| oz | ounces | 28 | grams | g |
| lb | pounds | 0.45 | kilograms | kg |
| | short tons (2000 lb) | 0.9 | metric tons | t |
| | | **VOLUME** | | |
| tsp | teaspoons | 5 | milliliters | ml |
| Tbsp | tablespoons | 15 | milliliters | ml |
| fl oz | fluid ounces | 30 | milliliters | ml |
| c | cups | 0.24 | liters | L |
| pt | pints | 0.47 | liters | L |
| qt | quarts | 0.95 | liters | L |
| gal | gallons | 3.8 | liters | L |
| $ft^3$ | cubic feet | 0.03 | cubic meters | $m^3$ |
| $yd^3$ | cubic yards | 0.76 | cubic meters | $m^3$ |
| | | **TEMPERATURE (exact)** | | |
| °F | Fahrenheit temperature | 5/9 (after subtracting 32) | Celsius temperature | °C |

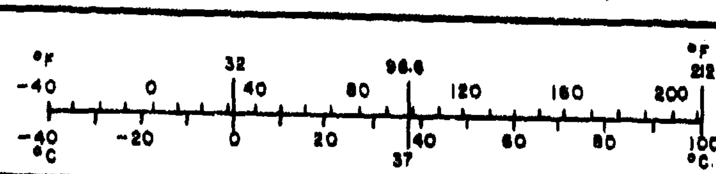
* 1 in = 2.54 cm (exactly)






## Approximate Conversions from Metric Measures

| Symbol | When You Know | Multiply by | To Find | Symbol |
|---|---|---|---|---|
| | | **LENGTH** | | |
| mm | millimeters | 0.04 | inches | in |
| cm | centimeters | 0.4 | inches | in |
| m | meters | 3.3 | feet | ft |
| m | meters | 1.1 | yards | yd |
| km | kilometers | 0.6 | miles | mi |
| | | **AREA** | | |
| $cm^2$ | square centimeters | 0.16 | square inches | $in^2$ |
| $m^2$ | square meters | 1.2 | square yards | $yd^2$ |
| $km^2$ | square kilometers | 0.4 | square miles | $mi^2$ |
| ha | hectares (10 000 $m^2$) | 2.5 | acres | |
| | | **MASS (weight)** | | |
| g | grams | 0.035 | ounces | oz |
| kg | kilograms | 2.2 | pounds | lb |
| t | metric tons (1000 kg) | 1.1 | short tons | |
| | | **VOLUME** | | |
| ml | milliliters | 0.03 | fluid ounces | fl oz |
| L | liters | 2.1 | pints | pt |
| L | liters | 1.06 | quarts | qt |
| L | liters | 0.26 | gallons | gal |
| $m^3$ | cubic meters | 35 | cubic feet | $ft^3$ |
| $m^3$ | cubic meters | 1.3 | cubic yards | $yd^3$ |
| | | **TEMPERATURE (exact)** | | |
| °C | Celsius temperature | 9/5 (then add 32) | Fahrenheit temperature | °F |



## I. SUMMARY

1. **Summary.** Since first employed early in World War II, pipelines have served a vital role in the bulk distribution of fuel during every subsequent conflict involving U.S. combat forces. Pipelines have proven to be the most efficient means for overland transportation of large quantities of liquid hydrocarbon fuels. The present Army capability to install, operate, and maintain petroleum pipelines is examined herein in light of current commercial pipeline technology and projections of fuel consumption for combat units in the event of future hostilities.

The objective of this investigation is to provide a measure of effectiveness for and to determine the technical feasibility of alternative pipeline systems operating as subsystems in a large logistical system for distribution of fuels in a theater of operations during wartime conditions. Desired improvements in the Military pipeline operational capability include:

a. More rapid construction (up to 30 kilometers per day).
b. Greater system reliability.
c. Reduced personnel requirements.
d. Lower life cycle costs.
e. Minimizing potential for fuel losses.

A broad array of pipe materials, pipe joining techniques, pumping equipment, ancillary pipeline components, and system designs have been evaluated. The findings reveal that substantive improvements in the operational effectiveness of Military pipelines can be achieved using aluminum pipe and self-latching mechanical couplings in lieu of the existing Military standard grooved-end steel pipe joined by split-ring mechanical couplings and gaskets. This substitution will achieve the primary goal of increased construction rate with a reduction in manpower requirements. In addition, the change in pipe material and construction methodology will result in improved pipeline operational and maintenance characteristics.

The use of high-speed, medium-duty diesel engines at all pump stations is essential to minimizing total life cycle costs for pipeline systems. As fuel costs have continued to rise, the high efficiency of diesel engines has become the overriding factor in their favor.

Two or more pump units, operating in series, are needed at each booster pump station to realize the maximum pipeline system mission reliability at the lowest

overall cost. Equipment down time has a strong influence on mission reliability. Thus, to reduce logistical support requirements and eliminate excessive administrative down time waiting for repair parts, all pipeline pumps should share engines with other high-density items of equipment.

Except for special applications, flexible hoselines are neither efficient nor cost effective for transporting large quantities of fuel. Hoselines should be considered as a viable means for bulk distribution of fuels only if flexibility, high mobility, rapid deployment and recovery, and frequent relocation are essential mission requirements.

Development of an improved petroleum pipeline system should be accompanied by improvements in tanker mooring and discharge systems and bulk fuel storage facilities. The tactical commander's needs can be satisfied only if a complete bulk fuel distribution system extends from tankers moored off shore to the fuel tanks of the tactical vehicles.

## II. INTRODUCTION

2. **Subject.** This report contains the results of the system definition activities conducted by MERADCOM during the evaluation of alternative techniques for construction of military petroleum pipelines as subsystems of bulk petroleum fuels distribution systems in theaters-of-operation.

An Army reorganization of the echelons above division was approved by the Army Chief of Staff. The new doctrine eliminated the field army and, consequently, the field army support command from the organizational structure. Inherent in this reorganization were changes in responsibilities and changes in territorial organization which may affect bulk petroleum doctrine, organizations, equipment, and management procedures.

The "Special Analysis of Wheeled Vehicles (WHEELS)" study and a follow-on study, "Recommended Vehicle Adjustment Number 9 (REVA-9) (Expanded)" recommended reductions in the number of vehicles, including bulk petroleum vehicles, organic to the armored, infantry, and mechanized (AIM) divisions and nondivision units. The WHEELS study and REVA-9 (Expanded) study covered vehicle requirements by TOE organization but did not address doctrine, organizations, materiel requirements, and management procedures for effective bulk petroleum supply and distribution in the theater-of-operations.

In the event of any military conflict within the foreseeable future involving a significant commitment of combat forces in conventional warfare, immense quantities of liquid hydrocarbon fuels will be required to support combat operations. The

theater army is normally assigned the responsibility to provide and operate the theater petroleum distribution system in support of all U.S. forces and other authorized consumers operating in a theater-of-operations. This includes inland waterway and intraharbor movement of bulk fuel supplies. As a result of the increased consumption of fuels, the demand for transporting fuels has outgrown the capability of existing bulk fuel distribution systems.

The Adjutant General, Department of the Army, by a letter directive dated 6 January 1975, directed the U.S. Army Training and Doctrine Command (USATRADOC) to conduct a study to determine the adequacy of current doctrine, organizations, equipment, and management procedures to provide petroleum storage and distribution within theaters-of-operation and, where appropriate, to recommend necessary changes in doctrine, organization, equipment, and management procedures. By indorsement to the DA letter directive, Headquarters, TRADOC designated the U.S. Army Logistic Center (USALOGC), Fort Lee, Virginia as the activity to perform the study. The responsibility was further delegated to the U.S. Army Quartermaster School, Fort Lee, Virginia. The results of that study are contained in the U.S. Army Quartermaster School Final Report, "Bulk Petroleum Fuels in a Theater of Operations," June 1977 (Volume I, Executive Summary and Main Report, and Volume II, Appendixes). The results of this investigation of alternative pipeline concepts and construction techniques are intended to supplement the findings of the Quartermaster School study.

3. **Background.** Liquid hydrocarbon fuels were initially used by military forces in small quantities. These limited quantities of fuel were shipped and stored in 5-gallon cans and 55-gallon drums employing the same logistical support procedures used for distribution of other packaged products.

The advent of mechanized military forces substantially increased the quantities of fuels consumed in a theater-of-operation. Distribution and storage of fuels as packaged products in sufficient quantities to meet the increasing demand placed an undue burden on the logistical system. Use of tank trucks and railroad tank cars provided some relief in the number of cans and drums that had to be handled. The rapid advances in the mechanization of our Armed Forces, however, resulted in the consumption of liquid hydrocarbon fuels in quantities which exceeded reasonable expectation for distribution of fuels as packaged products using the then existing logistical supply systems. As a result, pipelines were first used by the military for bulk fuels distribution soon after the United States entered World War II.

Prior to the entry of the United States into World War II, the Shell Oil Company submitted to the War Department a proposal for a lightweight grooved-end steel pipe and bolted-coupling pipeline system that was easily assembled by hand.

This proposal received little attention because of a general disinterest in military petroleum pipelines and satisfaction with existing methods of fuel distribution. It was not until 1942 that the War Department established a policy for use of pipelines for distribution of gasoline in support of combat operations. The pipeline concepts and construction techniques adopted during World War II were essentially those proposed by Shell Oil Company and are still in effect today.

Documentation of events surrounding the use of coupled pipelines during World War II, the Korean War, and the Vietnam Conflict indicates a wide range of problems. Despite these problems, the evidence shows pipelines to be an effective mode for overland transportation of large quantities of liquid fuels.

Although the use of plastic, composite, and aluminum pipelines by Industry has increased significantly in recent years, welded steel pipelines still dominate the commercial pipeline industry. The quality of a welded steel pipeline is determined by the quality of the welds. Pipeline welding is a difficult and rigorous task where perfection is required to produce a reliable pipeline. Civilian pipeline welders are usually men of exceptional skill who have achieved a high degree of proficiency through training and extensive experience. They maintain their high-level proficiency through continuous field practice. Lacking continuing requirements for construction of welded steel pipelines, it is impossible for the Army to develop and maintain an adequate crew of qualified welders. Even if an adequate number of qualified welders were available, the maximum possible rate of construction using manual pipeline welding techniques would be too slow to support the tactical operations of today's highly mobile military forces.

In 1957, the Army initiated action on a development program for an automatic pipeline girth welding machine. A laboratory model of a high-frequency, induction-pressure welding machine developed under this program achieved limited success. On 4 January 1960, however, the Office, Chief of Engineers directed that work on the automatic girth welder be terminated on the basis that studies revealed no requirement for welded pipelines for overland transportation of fuels. A subsequent study conducted by the Combat Development Group of the Engineer School, at the direction of the Chief of Engineers, recommended accelerated development of an automatic pipeline welder for high-pressure pipeline of 8- and 12-inch diameters.

In late 1961, an experimental mobile pipe mill developed by Industry was used to construct 30 miles of 8-inch product pipeline. This mill fabricated high-pressure, longitudinally welded steel pipe. The pipe was produced in long lengths as the self-propelled, self-contained mill moved along the pipeline right-of-way. Army observers were impressed with the potential construction capability of the mobile pipe mill concept. After projected construction capabilities were compared, the mobile pipe mill

was considered to have greater military potential than the automatic girth welder. As a result, all work on the automatic girth welder was terminated and a development program for a mobile pipe mill was initiated on 17 August 1962.

An extensive investigation of the mobile pipe mill was conducted while monitoring of the operation of the prototype mobile pipe mill developed by Industry continued. An Engineering Feasibility Study revealed major problems with production rate, operability, reliability, maintainability, maneuverability, transportability, and safety. The ability to produce good longitudinal welds was considered critical to the success of a mobile pipe mill. A detailed welding study recommended the addition of a weld-normalizing process. The additional power and equipment required for normalizing would increase the size and complexity of the mill making it improbable that the desired performance could be achieved. On this basis, MERADCOM recommended termination of the mobile pipe mill development task and was directed to initiate a study program to determine the most advantageous military POL pipeline construction technique.

Following termination of the mobile pipe mill development program, MERADCOM began investigating alternative methods and materials for pipeline construction. From this investigation, field fabrication of composite pipe emerged as a concept meriting further examination. A feasibility study conducted for MERADCOM by the Materials Engineering Division, Feltman Research Laboratory, Picatinny Arsenal concluded field fabrication of composite pipe could be accomplished by wrapping multiple plies of resin-impregnated fiberglass, woven cloth tape and curing the resin with high-intensity ultraviolet light. Subsequent research in this area has indicated that improved resin cure mechanisms and a mandrel for a continuous wrapping process must be developed before field fabrication of composite pipe can be considered a viable approach for military pipeline construction. A critical factor in demonstrating the military suitability of field-fabricated composite pipe, or any other method of pipeline construction, is the ability to achieve an acceptable rate of construction.

During this same time period, the Combat Operations Research Group (CORG) of Technical Operations, Inc. was conducting a study for the U.S. Army Combat Development Command Engineer Agency to identify bulk petroleum distribution systems that would be effective in all levels of warfare. The CORG study, Bulk Petroleum Facilities and Systems (BPFS), involved an extensive analysis of a large number of candidate pipe materials, joining methods, pumping units, storage tanks, and mooring equipment, resulting in a recommended Army bulk petroleum system for

the 1975 time frame.[1-9] However, materiel development requirement documents authorizing development of recommended new items were never approved.

This investigation includes reassessing many of the pipeline components and system concepts evaluated by CORG. Data from the BPFS study are utilized herein to the extent possible. New developments in technology are incorporated where applicable. The rapid rise of costs since 1967-1969 when the BPFS study was conducted has required substantial updating of the cost data. In addition, this analysis is based on different operational scenarios reflecting current projections of future military fuel requirements.

4. **Statement of Problem.** The objective of this investigation is to provide a measure of effectiveness and determine the technical feasibility of alternative pipeline systems as a subsystem of a logistical system for overland transportation of bulk liquid hydrocarbon fuels by military troops in a theater-of-operations under wartime conditions. The results of this study are intended to identify a pipeline systems concept that, to the extent possible, will:

a. Maximize the system reliability where system reliability is defined as the probability that a quantity of fuel equal to the minimum daily consumption can be transferred from a port of entry to the bulk distribution breakdown point.

---

[1] R. Stanley LaValee et al; *Bulk Petroleum Facilities and Systems (BPFS) - 1970-1985, Phase I: 1970-1975, Main Report.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1968.

[2] Edward W. King; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase I: 1970-1975, Annex A, Historical and Doctrinal Review.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

[3] R. Dean George et al; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase I: 1970-1985, Annex B, Part I: Military Equipment Survey.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

[4] R. Dean George et al; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Annex B, Part II, Industry Equipment Survey.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

[5] Ray A. Anderson; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase I: 1970-1975, Annex C, Pipeline Simulation Model.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

[6] Ray A. Anderson et al; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase I: 1970-1975, Annex E, Cost Effectiveness Analysis.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

[7] Gordon B. Page and Richard A. Tarker; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase I: 1970-1975, Annex F, Engineer Organization and Equipment.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

[8] R. Stanley LaValee and Kenneth R. Simmons; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase I: 1970-1975, Annex G, Synthesized Engineer Bulk Petroleum Facilities System.* Combat Operations Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

[9] John M. McCreary et al; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase II: 1975-1985.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

b. Maximize the rate of construction to provide the capability to advance the pipehead as rapidly as possible, at rates up to 30 kilometers (18.6 miles) per day.

c. Minimize the number of personnel, skill levels, and training required for pipeline construction, operation, and maintenance of military petroleum pipelines.

d. Minimize the total life cycle cost for a complete pipeline system.

e. Minimize the potential for fuel losses due to natural disasters, hostile action, pilferage, contamination, and administrative handling errors.

## III. INVESTIGATION

5. **Methodology.** This section describes the procedures, assumptions, constraints, and scenarios established as a basis for comparison of candidate pipeline components and synthesized systems. The first step in the analysis process, illustrated in Figure 1, is evaluation of the major components included in an integrated pipeline system. These components analyses provide the basis for selection of components during the synthesis of pipeline systems for systems evaluation. The results of the reliability and technological risk assessments are considered in evaluating the cost and operational effectiveness of the candidate systems.

a. **Assumptions.** For the purpose of this investigation, the following assumptions are applicable unless otherwise stated herein.

(1) All performance characteristics shall be based on standard atmosphere conditions.

(2) All pipelines shall be used to handle multiple products using conventional batching procedures. The product mix shall consist of 20 percent motor gasoline, 30 percent diesel fuel, and 50 percent jet fuel (JP-4). All flow characteristics shall be based on the heaviest fuel which is diesel having a specific gravity (SP GR) of 0.8448.[10]

(3) Each candidate pump station will include a manifold of the same basic design used in the Army Facilities Components System (AFCS). Changes to the standard manifold designs will be made to adapt the pressure rating of the manifold to the requirements of each particular pump station concept.

---

10 *Military Petroleum Pipeline System*, Department of the Army Technical Manual, TM 5-343; February 1969; p. 6-2.

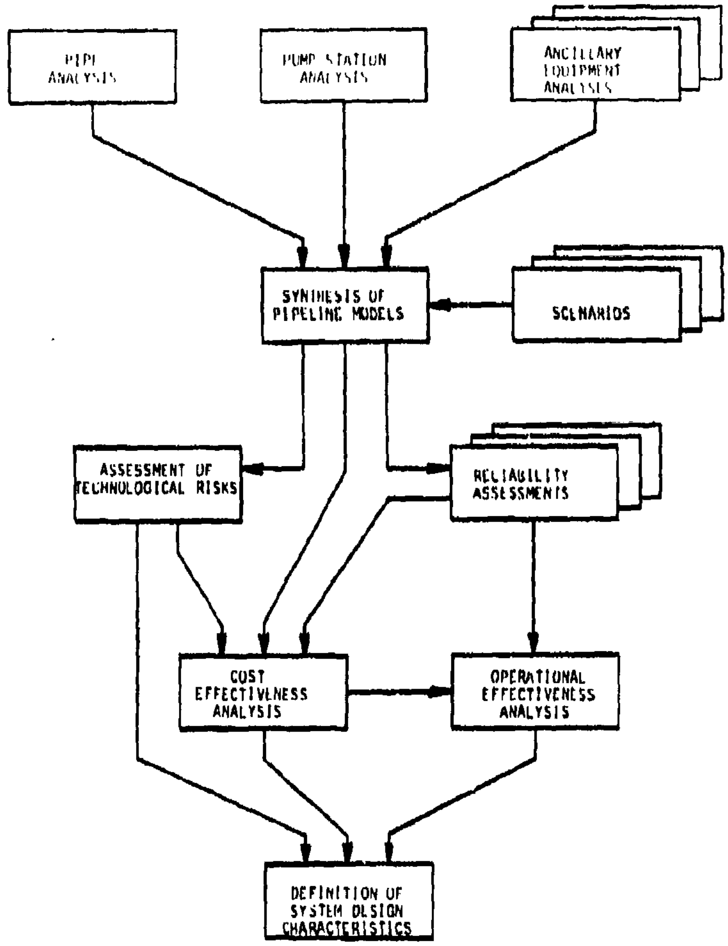



**Figure 1. Schematic diagram of analysis procedure.**

b. **Constraints.** Unless otherwise stated herein, the following constraints are applicable throughout this investigation:

(1) Construction, operation, and maintenance of all pipeline candidates must be possible under environmental conditions specified in AR 70-38 for climatic categories 1 through 7.

(2) The nominal diameter of all candidate pipelines shall be 4, 6, or 8 inches. Use of multiple parallel lines to obtain the required throughput capability is permissible.

(3) All pipeline components, all materials, and any special items of equipment required for pipeline construction and/or maintenance shall be air-transportable by C-130 aircraft.

c. **Scenarios.** Two hypothetical missions are defined in the following paragraphs to provide a common basis for comparison of candidate components and alternative systems. These scenarios are general in nature reflecting various operational requirements that could occur in numerous locations throughout the world. No attempt has been made to develop mission profiles representative of specific threats.

(1) **Scenario I – Ninety-day conflict.** U.S. troops are deployed by air into a foreign objective area 100 miles inland from an available port of entry. Deployment of additional personnel and equipment into the same area continues until day +40.

The initial elements deployed arrive with sufficient supplies, including fuel, to sustain operations for 3 days. Beyond day +3, all fuel is brought forward by airlift and/or 5,000-gallon tank trucks from an existing commercial marine terminal at the port-of-entry 100 miles away until a pipeline can be installed.

To expedite installation, the pipeline is laid along the most direct route possible utilizing road ditches, railroad right-of-ways, stream beds, etc., through areas where grading would otherwise be required. The resulting pipeline profile is defined in Table 1 and Figure 2. This pipeline profile is intended to reflect the major changes in elevation which impact on pipeline system design. No attempt has been made to include minor undulations in elevation which have little effect on pipeline design or performance.

Table 1. Pipeline Profile for Scenario I

| Distance from Marine Terminal (miles) | Elevation Above* Marine Terminal (feet) |
|---|---|
| 0 | 0 |
| 10 | 100 |
| 20 | 400 |
| 30 | 800 |
| 40 | 1300 |
| 50 | 2000 |
| 60 | 3000 |
| 70 | 1500 |
| 80 | 500 |
| 90 | 400 |
| 100 | 400 |

* Profile is assumed to have a constant slope between elevations shown.

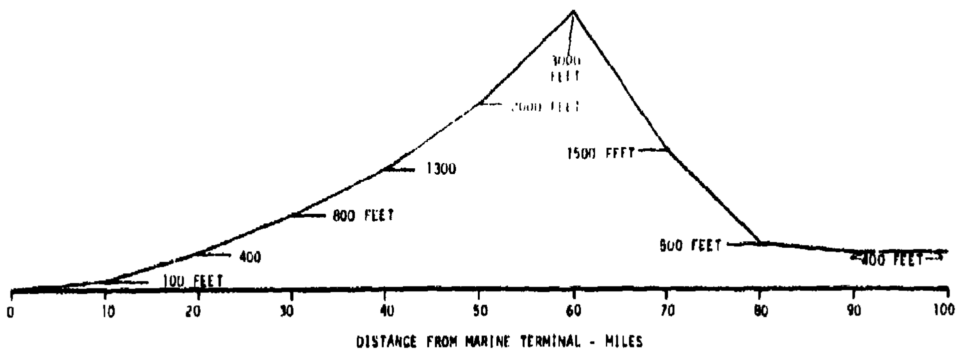



**Figure 2. Pipeline profile for Scenario I.**

The pipeline runs through neutral territory. Sabotage by guerrilla action and pilferage are constant problems.

The daily fuel consumption in the objective area increases at a relatively steady rate from day +1 to day +40. The daily fuel requirements are constant from day +40 through day +90. The available commercial marine terminal at the port of entry has adequate mooring facilities and storage capacity to assure a constant supply of fuel to the pipeline. Actual daily fuel requirements are shown in Table 2 and Figure 3.

A political settlement is reached 90 days after the initial deployment of troops and a cease-fire goes into effect. All U.S. forces are withdrawn; however, the pipeline is left in place to be maintained by indigenous forces pending the potential outbreak of further hostilities.

**(2) Scenario II – Established Theater-of-Operations.** Forces are operating in an established theater-of-operations. The primary port-of-entry for fuel has been destroyed by enemy action creating a need to construct a pipeline to supply fuel from an alternate port-of-entry 100 miles from an intermediate storage terminal.

All pipes, pumps, and ancillary items required for installation of the pipeline are available from resources stockpiled in-country. The new pipeline route is over gentle rolling terrain which can be cleared adequately for pipeline construction by not more than two passes with a bulldozer. The change in elevation is assumed to be a constant gradient rising 500 feet (5 feet per mile) from the port-of-entry to the intermediate storage terminal.

Table 2. Daily Fuel Consumption Scenario I

| Day | Daily Consumption | | Cumulative Total |
|---|---|---|---|
| | Gallons/Day | Barrels/Day | (Barrels) |
| 1 thru 3 | 0 | 0 | 0 |
| 4 | 4,200 | 100 | 100 |
| 5 | 4,200 | 100 | 200 |
| 6 | 192,990 | 4,595 | 4,795 |
| 7 thru 9 | 226,170 | 5,385 | 20,950 |
| 10 | 298,200 | 7,100 | 28,050 |
| 11 | 313,320 | 7,460 | 35,510 |
| 12 | 435,750 | 10,375 | 45,885 |
| 13 | 435,750 | 10,375 | 56,260 |
| 14 thru 19 | 477,330 | 11,365 | 124,450 |
| 20 thru 24 | 605,220 | 14,410 | 196,500 |
| 25 thru 29 | 708,330 | 16,865 | 280,825 |
| 30 | 786,450 | 18,725 | 299,550 |
| 31 thru 33 | 869,820 | 20,710 | 361,680 |
| 34 | 942,060 | 22,430 | 384,110 |
| 35 | 934,710 | 22,255 | 406,365 |
| 36 thru 38 | 1,070,370 | 25,485 | 482,820 |
| 39 thru 54 | 1,128,540 | 26,870 | 916,740 |
| 55 thru End | 1,160,040 | 27,620 | 2,183,260 |

The alternative pipeline systems are designed to deliver an average of 35,000 barrels of fuel daily when operating 23 hours per day.

The pipeline will be used to support military operations for a period of 3 years.

6. **Pipeline Operation.** A military bulk petroleum fuels distribution system in a theater-of-operations consists of an array of equipment and facilities. When U.S. forces are first deployed into an objective area, the distribution system will be very simple and will grow as the campaign develops. Figure 4 illustrates, in schematic form, the type of facilities which might be found in a theater of operations which has developed sufficiently to provide stability in rear areas. The complexity of the distribution system and the amount of equipment involved will vary with the size of the combat force being supported.

A ship-to-shore facility is required to transfer fuel from tankers moored offshore to the onshore facilities. In protected waters, the ship-to-shore facility may be a pipeline laid out onto a jetty where the tanker is berthed alongside. In unprotected

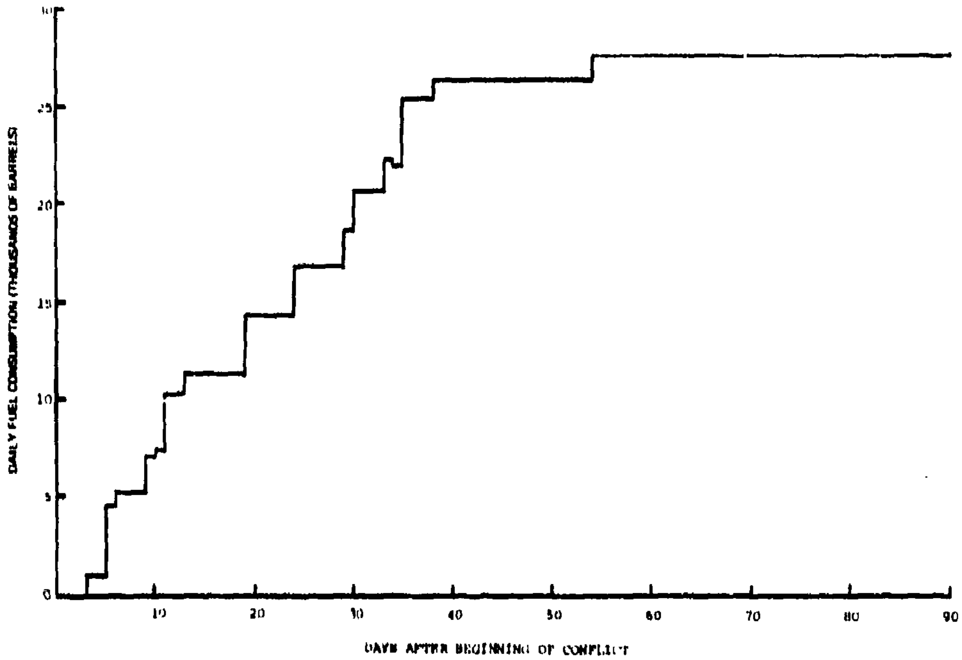


Figure 3. Daily fuel consumption -- Scenario I.

waters, the ship must be moored some distance off the beach using a multileg or single-point mooring facility. In such cases, the fuel must be transferred ashore through a floating hoseline, a submarine hoseline, or a bottom-laid pipeline. Pumps aboard the tanker provide the pressure to push the fuel to the shoreline. Although this study does not directly address the technical aspects of construction, operation, and maintenance of ship-to-shore facilities, much of the information relating to pipe and pipe-joining techniques may be applicable to offshore systems. It must be recognized, however, the criteria for selecting the best technical approach for offshore pipelines are significantly different from that for onshore pipelines. The need for offshore facilities is discussed in Appendix A to this report.

The fuel is delivered from the ship-to-shore facility to a marine terminal storage facility or base terminal. The ship-to-shore pipeline will be connected to the marine terminal manifold. All storage tanks within the marine terminal will be interconnected to this manifold by pipelines so that fuel may be transferred from the tanker directly to any of the storage tanks. This switching manifold also provides the capability to transfer fuel between tanks within the marine terminal and to deliver fuel to pipelines extending inland. Flood and transfer pumps are installed in the switching manifold for this purpose.

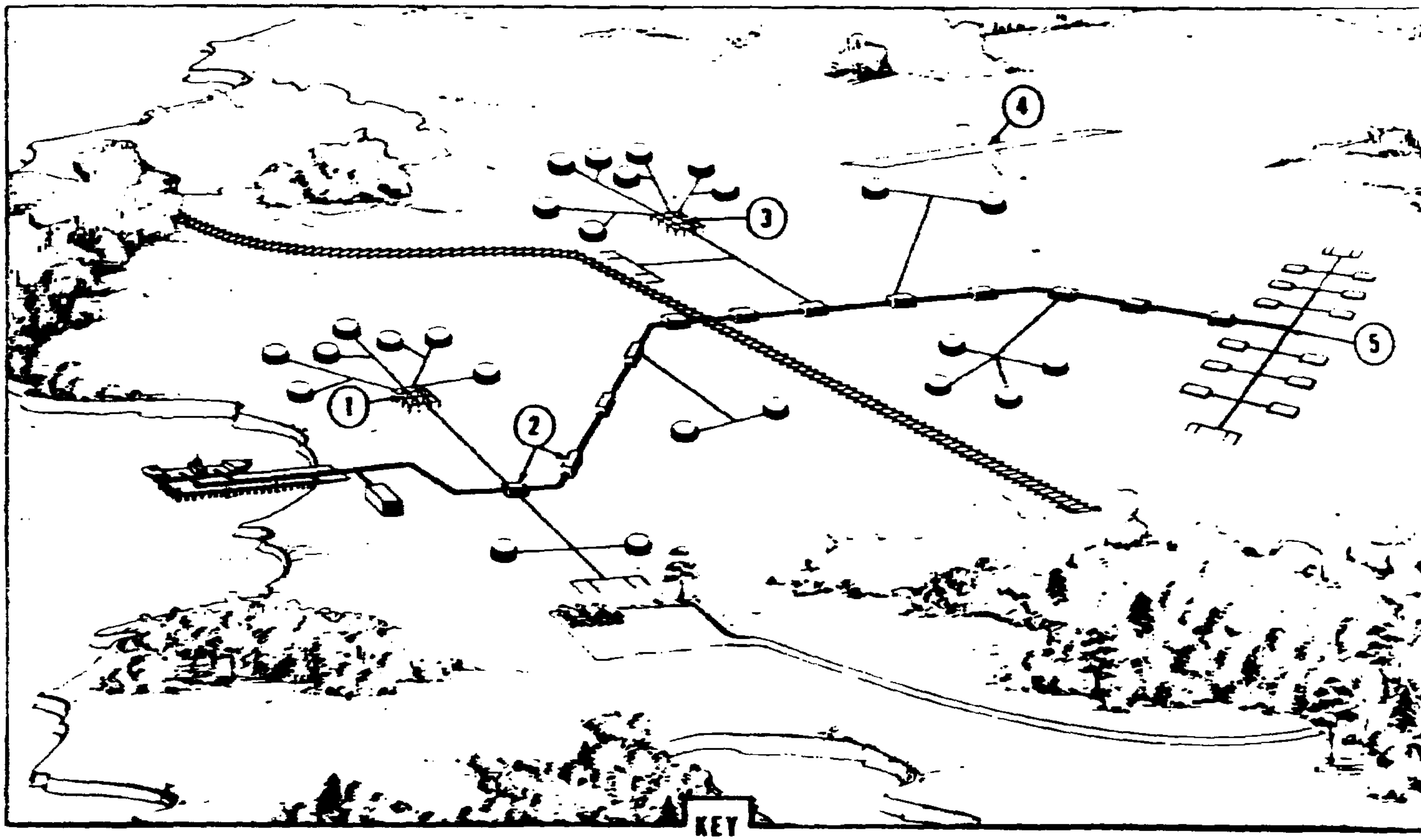



Figure 4. Schematic diagram of bulk fuels distribution system.

The total storage capacity required at a marine terminal depends on the total amount of fuel required to support operations within the theater as well as a number of other factors. The required storage capacity may be obtained using existing commercial storage facilities, bolted or welded steel tanks, collapsible self-supporting fabric tanks, or hasty storage reservoirs. For the purpose of this study it is assumed that adequate storage can be provided. This study does consider the pipe required for the switching manifold and for interconnecting the tanks.

Fuel is transferred inland from a marine terminal by pipeline to intermediate or pipehead storage terminals. Functionally, these storage facilities are essentially the same as marine terminals. Intermediate terminals are generally located where branch pipelines leave the main line and where fuel must be distributed to users. The pipehead storage terminal is located at the end of the pipeline. A pipehead terminal will become an intermediate terminal if the pipeline is extended. In all cases, the storage capacity at a pipeline terminal is a function of combat support requirements.

Typically, military pipelines have been classified according to use as three general types: assault, tactical, and logistical.

An assault system is installed rapidly to provide fuel to advancing combat forces during fast moving assault operations. Used as an expedient means to satisfy rapidly changing situations, assault systems are temporary facilities. The Hoseline Outfit, 4-inch, FSN 3835-892-5157, consisting of 13,000 feet of 4-inch collapsible hose, a booster pump, and ancillary items including valves, fittings, a repair kit, a packing kit, etc., is the only system the Army has standardized for this purpose.

A tactical pipeline system may be temporary or semipermanent and is emplaced rapidly to maintain the pipe head as close as possible to advancing forces. In general, a tactical system requires more effort and time to emplace than an assault system but provides the capability to handle larger quantities of fuel. Employing current Army doctrine, mechanically coupled lightweight steel tubing would be used for tactical pipelines.

Logistical systems are more permanent pipelines designed to transfer large quantities of fuels within stabilized areas. At present, a logistical system would be a welded steel pipeline installed by a civilian pipeline construction company.

This study reviews a wide variety of pipe materials and construction techniques seeking improved means for satisfying the Army's needs for assault, tactical, and logistical systems. Hoselines and pipelines of various diameters and pumps of varying capacities are considered in relation to the range of fuel quantities that may be required.

a. **Classes of Pumps.** Pumping units are critical elements in all fuel distribution systems providing the power to move the fuel through the pipelines, to transfer fuel between tanks, and for dispensing fuels. A bulk distribution system of the type illustrated in Figure 4 will include a variety of pumping requirements depending on many factors. For the purpose of this study, all pumping units are considered to fall within two general classes: flood-and-transfer pumps and booster pumps.

(1) **Flood-and-Transfer Pumps.** Included in this class of pumps are those pumping units frequently referred to as flood pumps, transfer pumps, feeder pumps, or supply pumps. Flood-and-transfer pumps are normally installed in storage terminal switching manifolds where they serve a variety of functions. In general, they are used to transfer fuel into, within, and out of storage terminals. In support of a pipeline operation, a flood-and-transfer pump draws fuel from storage tanks and delivers the fuel to the first pipeline booster pumping station providing the required manifold suction pressure. Typically, flood-and-transfer pumps are high-capacity, low-head pumps designed to operate without a positive pressure at the pump inlet (suction). In addition, flood-and-transfer pumps are normally self-priming after initial charging of the case with liquid.

(2) **Booster Pumps.** Booster pumps, sometimes referred to as mainline or trunkline pumps, provide the pressure to maintain flow through the pipeline. When more than one booster pump is located at a booster pumping station, the pumps are usually located adjacent to each other in a manifold which may connect the pump suction and discharge lines in parallel or series. Booster pumps are high-capacity, high-head pumps and normally require a positive pressure at the pump inlet (suction).

b. **Method of Operation.** The type and amount of equipment required at a pumping station varies depending on the method of operation. There are three basic methods of pipeline operation: tight-line, float-tankage, and regulation tankage.

(1) **Tightline Operation.** The fuel distribution system illustrated in Figure 4 depicts tightline pipeline operations between the marine terminal and the intermediate storage terminal and between the intermediate storage terminal and the pipehead storage terminal. At each booster pumping station, the receiving pipeline is connected directly to the inlet of the booster station manifold as shown in Figure 5. This is the most complex method of pipeline operation because it requires exact coordination of all pumping units along the pipeline between storage terminals.

(2) **Float-Tankage Operation.** Using the float-tankage method of pipeline operation each pumping station draws fuel directly from storage tanks located at the pumping station site. At the next pumping station the fuel is discharged from the incoming pipeline directly into storage tanks. The float tankage method of operation, illustrated in Figure 6, allows each pipeline segment to operate independently. The

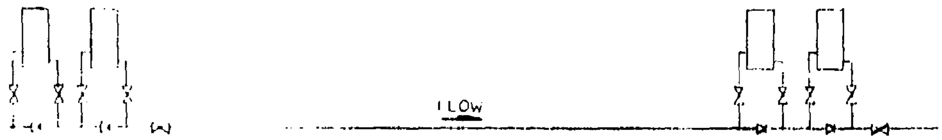



**Figure 5. Tightline pipeline operation.**

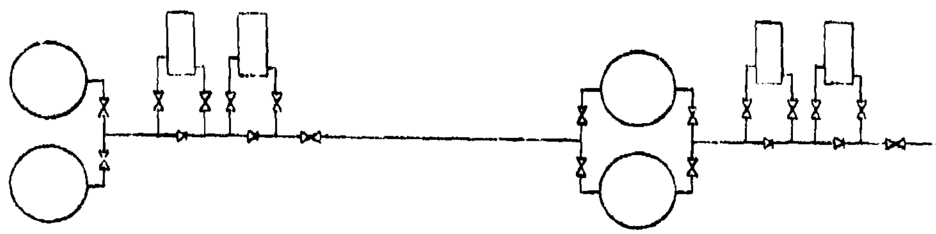


**Figure 6. Float-tankage pipeline operation**

only requirement for coordination between pumping stations is for the receiving station to monitor the incoming flow to insure that the fuel being received is directed into the proper storage tanks.

(3) **Regulation Tankage Operation.** A pipeline operated using the regulation tankage method shares some common characteristics with both the tight-line and float-tankage methods of operation. As in a tight-line operation, the pipeline coming into a regulation tankage pumping station is connected directly to the inlet of the pumping station manifold. However, as illustrated in Figure 7, an open line from the incoming pipeline is also connected to a storage tank as in a float-tankage operation. Using the regulation tankage method of operation, all pumping stations are operated simultaneously at or near the same flow rate. The storage tank at each booster pumping station allows a slight variation in flow rates between adjacent pumping stations. It also allows brief periods of interruption of operation of any pipeline segment without affecting the operation of the other pumping stations. Since the storage capacity of the regulation tankage at each pumping station normally would be small, the average flow rate of each pumping station between two major storage terminals must be approximately the same over any extended period of operation. Otherwise, one pumping station will require all pumping stations to adjust their pumping rate or shutdown.

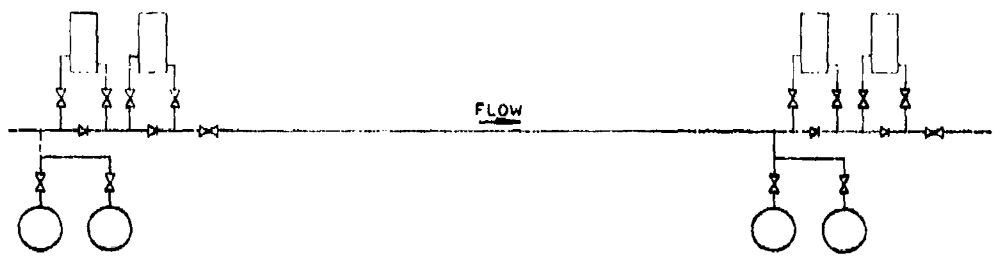



**Figure 7. Regulation tankage pipeline operation.**

Certain advantages and disadvantages are inherent to each method of pipeline operation. However, the influence each of the general characteristics has on the suitability of a method of operation for a particular application is tempered by numerous factors. The National Security Industrial Association (NSIA) trade-off technique is used to compare the suitability of the three methods for military pipeline operations. (A detailed description of the application of the NSIA trade-off technique is contained in Appendix B). The factors considered in this evaluation are shown in Tables 3, 4, and 5 with the relative weighting and rating values assigned for each factor.

Table 3. Evaluation of Tightline Method of Pipeline Operation

| Parameters | Considerations | Relative Weighting | Basic Rating | | Adjusted Values | |
|---|---|---|---|---|---|---|
| | | | Undesirable | Desirable | Undesirable | Desirable |
| Equipment Required | Pump Units | 3 | | +70 | | +210 |
| | Storage Tanks | 3 | | +70 | | +210 |
| Installation | Manpower | 3 | | +90 | | +270 |
| | Skill Levels | 4 | | +70 | | +280 |
| | Equipment | 2 | | +50 | | +100 |
| | Time | 5 | | +100 | | +500 |
| Operation | Manpower | 3 | | +70 | | +210 |
| | Skill Level | 4 | -70 | | -280 | |
| | Throughput Capacity | 5 | -50 | | -250 | |
| | Commingling | 4 | | +70 | | +280 |
| | Fuel Losses | 1 | | +10 | | +10 |
| | Safety | 1 | -30 | | -30 | |
| | Communication | 1 | -50 | | -50 | |
| Maintenance | Manpower | 4 | | +30 | | +30 |
| | Skill Levels | 3 | | +10 | | +10 |
| | Equipment | 2 | | +20 | | +20 |
| Totals | | 48 | | | -610 | +2130 |

Net Value = 2130 - 610 = 1520
Average Net Value = 1520/48 = +31.7

Table 4. Evaluation of Float-Tankage Method of Pipeline Operation

| Parameters | Considerations | Relative Weighting | Basic Rating Undesirable | Basic Rating Desirable | Adjusted Values Undesirable | Adjusted Values Desirable |
|---|---|---|---|---|---|---|
| Equipment Required | Pump Units | 3 | -30 | | -90 | |
| | Storage Tanks | 3 | -30 | | -90 | |
| Installation | Manpower | 3 | -30 | | -90 | |
| | Skill Levels | 4 | -30 | | -120 | |
| | Equipment | 2 | -10 | | -20 | |
| | Time | 5 | -70 | | -350 | |
| Operation | Manpower | 3 | -50 | | -150 | |
| | Skill Level | 4 | | +70 | | +280 |
| | Throughput Capacity | 5 | | +100 | | +500 |
| | Commingling | 4 | -70 | | -280 | |
| | Fuel Losses | 1 | -10 | | -10 | |
| | Safety | 1 | | +50 | | +50 |
| | Communications | 1 | | +70 | | +70 |
| Maintenance | Manpower | 4 | -30 | | -120 | |
| | Skill Levels | 3 | -10 | | -30 | |
| | Equipment | 2 | -10 | | -20 | |
| Totals | | 48 | | | -1370 | +900 |

Net Value = 900 - 1370 = -470
Average Net Value = -470/48 = -9.8

Table 5. Evaluation of Regulation Tankage Method of Pipeline Operation

| Parameters | Considerations | Relative Weighting | Basic Rating Undesirable | Basic Rating Desirable | Adjusted Values Undesirable | Adjusted Values Desirable |
|---|---|---|---|---|---|---|
| Equipment Required | Pump Units | 3 | | +10 | | +30 |
| | Storage Tanks | 3 | | +10 | | +30 |
| Installation | Manpower | 3 | | +10 | | +30 |
| | Skill Levels | 4 | -30 | | -120 | |
| | Equipment | 2 | -10 | | -20 | |
| | Time | 5 | | +30 | | +150 |
| Operation | Manpower | 3 | -30 | | -90 | |
| | Skill Levels | 4 | | +20 | | +80 |
| | Throughput Capacity | 5 | | +50 | | +250 |
| | Commingling | 4 | -50 | | -200 | |
| | Fuel Losses | 1 | -10 | | -10 | |
| | Safety | 1 | | +50 | | +50 |
| | Communications | 1 | -30 | | -30 | |
| Maintenance | Manpower | 4 | -10 | | -40 | |
| | Shell Levels | 3 | -10 | | -30 | |
| | Equipment | 2 | | +10 | | |
| Totals | | 48 | | | -540 | +620 |

Net Value = +620 - 540 = + 80
Average Net Value = +80/48 = +1.7

Equipment requirements at the first pipeline booster pumping station is virtually the same for all methods of pipeline operation. A flood-and-transfer pump is required to draw fuel from the storage terminal and deliver the fuel to the booster pumping station manifold at the required suction pressure. The booster pumping station must include a sufficient number of booster pumps to develop the required pumping station discharge pressure.

In a float-tankage pipeline operation every booster pumping station is essentially a small intermediate storage terminal. Thus, every booster station requires virtually the same amount of equipment as is required at an intermediate storage terminal except the total storage capacity may be less. In some applications, it is possible that the storage capacity available at booster pumping stations may permit some reduction in the storage capacity required at marine, intermediate, and pipehead storage terminals. However, since storage tanks, switching manifolds, flood-and-transfer pumps, and booster pumps are required at every booster pumping station, the float-tankage method of pipeline operation requires the greatest amount of equipment. Thus, from an equipment standpoint, the float-tankage method of operation is the least desirable approach.

By contrast, the tight-line method of pipeline operation is the most desirable approach because it requires the least amount of equipment. Since each booster pumping station manifold receives fuel from the incoming pipeline at the required suction pressure, a tight-line booster pumping station consists of the number of booster pumps necessary to develop the required pumping station discharge pressure plus the interconnecting manifold. Requiring no storage tanks at the booster stations, the total storage capacity in a tight-line pipeline is that required at marine, intermediate, and pipehead storage terminals. Flood-and-transfer pumps are required only at the storage terminals. In some cases it may be desirable to have standby pumps at each booster station to improve the system reliability. Even in this event, the total equipment will be less than for either of the two other methods of operation.

In a regulation tankage pipeline operation, the amount of equipment required is a function of the desired flexibility of operation. If very limited storage capacity is provided at each booster pumping station, the amount of equipment required is minimized at the expense of operational flexibility. Increasing the storage capacity at each booster pumping station provides greater flexibility of operation. When minimum storage is used at each booster pumping station, the float-tankage method of operation is not appreciably different from a tight-line operation. At the other extreme, if the storage capacity at each booster pumping station in a regulation tankage pipeline operation is large, the equipment requirements and operational flexibility approaches that of a float-tankage operation. Because the system can be tailored to the requirements of the individual situation, the advantages of the regulation tankage method of operation are considered to marginally outweigh the disadvantages.

For military applications where rapid rates of construction are required, the tight-line method of pipeline operation is highly desirable because the required construction effort is minimized. Further, the tight-line method of operation requires little, if any, construction equipment for installation. On the same basis, installation of a float-tankage pipeline system will require the greatest amount of construction effort and support equipment since installation of any significant amount of storage facilities requires a substantial construction effort. If self-contained pump-engine units are used, the pumps require low skill levels for installation. Collapsible self-supporting tanks can be installed by relatively untrained personnel. Any other type of storage has the undesirable feature that some special skills and training are required to produce a satisfactory fuel container.

On the basis of installation factors, the tight-line method of operation is highly desirable because pump stations can be installed quickly. Again, the float-tankage method of operation is the least desirable approach because of the time required for installing storage tanks.

Manpower requirements are the greatest for a float-tankage operation since operation of each pump station requires the operation of a tank farm. Since this approach allows the greatest flexibility in operation, the skill levels required are not as stringent. Only a few people are required to operate a pipeline using the tight-line method. However, because every pump station must operate in exact coordination with all other pump stations, operating personnel must be well trained in pipeline operating procedures. A disadvantage of the tight-line method of operation is that each item of equipment must have high reliability to insure an acceptable system availability if the required throughput approaches the design delivery capacity of the system. The most obvious method of reducing reliability requirements for individual items of equipment is to use regulation or float tankage. Another alternative is to increase the maximum throughput capacity of the system so that the total demand can be delivered with the system operating at less than optimum performance and allowing more system downtime. Component and system reliability is examined more fully in subsequent sections of this report.

When batching is used to deliver several products through a single pipeline, commingling of the fuels at the interfaces between batches always occurs. The tight-line method of operation minimizes the commingling problem since, for each batch, only one interface must be cut out along the entire length of the pipeline. When float-tankage is used a new interface is introduced to the pipeline at each pumping station and must be cut out when the fuel is received into tankage at the next pumping station. Thus, the float-tankage method of operation results in substantially more commingling than in a tight-line operation. Also, each pumping station must have separate tankage for each type of fuel being handled.

When the regulation-tankage method of operation is used only one interface per batch is required between storage terminals. Variations in flow rates between adjacent pipeline segments will result in fuel being discharged into or drawn from the tankage at each booster pumping station. Thus, each pumping station must have separate storage for each type of fuel handled. When an interface passes through a booster pumping station, the valving in the tankage manifold must be switched to direct the flow into a tank containing the same type of fuel as that passing through the pipeline. At best, this method of operation will result in more commingling than a tight-line operation.

The tight-line method of pipeline operation minimizes vaporization losses since the fuel is maintained under a positive pressure at all times when in the pipeline. The float-tankage method of pipeline operation creates losses due to the constant breathing of tanks as the fuel is pumped in and out of the tanks at each pumping station. The loss of fuel is small when regulation tankage is used since only a small part of the fuel enters the tankage at each pump station.

In a tight-line pipeline operation, all pumping units along the pipeline are effectively in series. Thus, the pressure in the pipeline at any point is equal to the sum of the pressure rise across each pump unit upstream less all fluid friction losses occurring upstream from the point in question. If the downstream end of the pipeline is blocked by closing a valve or other means of stopping flow, the fluid friction losses are zero resulting in the pressure in the pipeline being equal to the summation of the pressure rise across all upstream pump units. If a line blockage were to occur while all pump units are operating, the pressure in the downstream end of the pipeline would exceed the burst pressure of the pipeline unless adequate overpressure controls are included in the system. To avoid this problem, each pump unit must be equipped with an automatic pressure control to shut down each pump unit in the event the discharge pressure exceeds a predetermined level. In addition, pressure relief valves should be included in the pipeline to relieve any excess pressure in the event the pump units failed to shut down should an overpressure condition occur.

In the float-tankage method of operation, the pipeline is open to the atmosphere (through a tank) where the pipeline enters each pumping station. Therefore, the maximum pressure to which the pipeline may be subjected, neglecting waterhammer and other transient conditions, is the maximum discharge pressure one pumping station can develop plus any additional pressure resulting from variations in elevation. With the exception of cases where extreme changes in elevation occur, there is little chance of overpressure conditions occurring in a float-tankage pipeline. Problems associated with downhill pressure regulation are examined in subsequent sections of this report.

In a regulation tankage pipeline operation the pipeline is normally open to a tank at the entrance to each booster station. Under normal operating conditions the regulation tankage pipeline, like the float-tankage pipeline, should not be subject to excessive pressures. However, if all the lines to regulation tankage are closed, the system becomes a tight-line operation. Because of this possibility, the regulation tankage pipeline requires overpressure protection similar to that of a tight-line pipeline.

There must be communication between pump stations irrespective of the method of operation. In a tight-line or regulation tankage operation, the entire system is controlled by a dispatcher who must be in constant contact with all pump stations. In the float-tankage method of operation each pump station must be able to communicate with the operator at the next station. In addition, a dispatcher must coordinate the amount and type of fuel to be pumped through the line. However, the total communications requirements are less critical than for tight-line or regulation tankage operation.

Most commercial pipelines are tight-line operations. The high reliability of commercial pipeline equipment and use of automation allowing control of an entire pipeline system from a central location has virtually eliminated the major disadvantages of the tight-line method of operation. The high cost of installation, operation, and maintenance of the additional storage tanks required for either a regulation or float-tankage method of operation cannot be justified for commercial applications.

The nature of military pipeline operations and the necessity to use equipment having lower inherent reliabilities than commercial pipeline equipment tend to increase the attractiveness of float- or regulation-tankage operation. However, any advantages in operational effectiveness offered by the float- or regulation-tankage methods of operation are offset by the lesser amounts of construction time and construction, operation, and maintenance personnel and equipment required for a tight-line pipeline. This is demonstrated by the average net values computed in Tables 3, 4, and 5.

From Table 3, the average net value for the tight-line method of pipeline operation is +31.7. Using the NSIA basic rating scale, this positive value equates to a desirable rating. From Table 5, the regulation tankage method of operation has a small, +1.7, but positive average net value. This small absolute average net value indicates the advantages, and the disadvantages negate each other.

The average net value of -9.8 computed in Table 4 corresponds to a slightly undesirable rating on the NSIA basic rating scale. This rating is undesirable only when compared to the other two alternatives considered. This rating should not be construed to indicate the float-tankage method of operation is unacceptable for military pipelines.

The tight-line method of pipeline operation is the method of operation used throughout the remainder of this study. This approach is taken recognizing conversion of a tight-line operation to a float- or regulation tankage operation is possible by simply adding storage tanks and flood-and-transfer pumps at each booster pumping station.

7. **Pump Stations.** The pump stations are literally the heart of a pipeline system. Design of an integrated pipeline system requires careful matching of pump station performance to pipeline flow characteristics. In the selection of the pump station design best suited for military pipeline application, it is necessary to evaluate the alternative types of pumps and prime movers available and to determine the optimum number of pump units per station.

a. **Types of Pumps.** Pump types are determined by the method used for converting mechanical energy (power) to hydraulic energy (flow and pressure). The broadest division is on the basis of displacement, either positive or variable (non-positive) displacement. Each revolution of a positive displacement pump displaces a fixed quantity of liquid. The amount of liquid displaced by each revolution of a variable displacement pump is a function of numerous operating parameters.

Types of positive displacement pumps include:

Diaphragm or bellows.
Gear (internal and external).
Peristaltic.
Lobed.
Piston (radial, axial, and eccentric).
Plunger (radial, axial, and eccentric).
Screw.
Swash-plate.
Vane (guided, sliding, and swinging).

Only piston- and plunger-type positive displacement pumps have seen any significant application in the pipeline industry. The principal advantage of piston and plunger pumps is their inherent high mechanical efficiency. The principal disadvantages of these pumps are: they are expensive, heavy, and bulky, and they deliver a pulsating flow. Most other types of positive displacement pumps are not well suited to pipeline applications because of limitations on available flow rates and discharge pressure.

To attain their high efficiencies, positive displacement pumps must be designed with close internal working clearances. As a result, any solid contaminants in

the liquid being pumped produces high wear rates and often causes premature pump failure. For this reason, positive displacement pumps are not considered suitable for military pipeline service.

Traditionally, military pipeline pumps have been variable-displacement-type centrifugal designs. In its pure form, a centrifugal pump is an impeller composed of a number of curved vanes radiating off a hub enclosed by a circular pumping chamber. As the liquid enters the center or eye of the impeller, it is swept up by the leading edge of the vanes. The centrifugal force created by the liquid being forced to rotate with the impeller slings the liquid to the outside of the impeller. At the outside diameter of the impeller, the pump case gathers the liquid and directs it toward the pump outlet converting the additional velocity imparted to the liquid by the impeller to hydraulic energy or pressure. Since flow through centrifugal pumps is from the center of the impeller outward from the axis of rotation, they are ofteh referred to as radial-flow pumps.

By locating a propeller or fan-shaped impeller in a fluid passage, energy can be imparted to the fluid by rotation of the impeller. In this case the direction of flow is parallel to the axis of rotation. Hence, this type of pump is referred to as an axial-flow pump. Although the pumping action is not a result of centrifugal force, axial-flow pumps are included in the broad classification of centrifugal pumps. This is a convenient grouping since the pumping action and performance characteristics are somewhat similar to a true centrifugal pump.

Combining some of the design principles of radial-flow and axial-flow pumps produces a pump which generates the pumping action partly by centrifugal force and partly by propeller action. Pumps of this type, known as mixed-flow pumps, can develop higher discharge pressures than straight propeller-type pumps and handle larger volumes more efficiently than true centrifugal pumps.

The term "centrifugal pump" as used herein, unless otherwise specified, includes radial-flow, mixed-flow, and axial-flow pumps. The pump design theory used to determine the propeller/impeller vane shape which will produce the required performance characteristics is well established and documented in detail in the technical literature. Thus, this investigation addresses the performance characteristics of centrifugal pumps only to the extent necessary for cost and operational effectiveness analysis.

As noted earlier, the desired performance characteristics for a pump operating under normal conditions are used to determine the physical design features of a centrifugal pump intended for a specific application. If actual operating conditions are different from the design parameters, a centrifugal pump adjusts its performance to match the existing conditions. Similarly, the performance characteristics

of a centrifugal pump can be altered by changing the pump rotational speed. As discussed in subsequent sections of this report, these two characteristics make centrifugal pumps extremely well suited for military petroleum pipeline systems.

The BPFS study conducted by CORG for the U.S. Army Combat Developments Command includes a detailed analysis of the suitability of centrifugal pumps for military pipeline service. This study[11] concluded:

> Because of the many advantages of the centrifugal impeller pump such as light weight, small size, ability to pump particle-contaminated fuel, and low cost, it is recommended for pipeline pumping applications.

This recommendation is supported by the use of centrifugal pumps almost exclusively by Industry for petroleum product pipelines. There have been no significant changes in pump technology since the BPFS study was completed in 1969. Thus, without further comparative analysis, centrifugal pumps are accepted in this investigation as the best type of pumps for military pipelines.

b. **Prime Movers.** A variety of prime movers suitable for driving pipeline pumps are available. The pipeline industry, dominated for many years by the diesel engine, recently has seen substantial gains in the popularity of electric-motor- and gas-turbine-engine-driven pumps. The selection of prime movers for commercial petroleum product pipeline pumps is usually based on an economic analysis. In each case, the required pump performance characteristics are well defined and all available energy sources can be identified. In contrast, military pipeline pumps are designed to satisfy a broad range of general requirements allowing the pumps to be used in a variety of applications. Since pump station locations are unknown, the alternative energy sources must be limited to those fuels that the military plans to have available.

(1) **Electric Motors.** Electric-motor-driven pumps have a lower initial cost, are smaller in size, weigh less, have a higher reliability, require less maintenance, and are more readily adapted to automated systems than pumps driven by any other type of prime mover. When a continuous supply of low-cost electricity is available from a reliable electrical power distribution system, electric-motor-driven pumps are clearly the best choice. The low probability of an adequate electric power supply line being readily available at the desired pump station locations, particularly in foreign countries, makes electric-motor-driven pumps an impractical consideration for military application.

---

[11] R. Dean George et al, *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Annex B, Part II, Industry Equipment Survey;* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

The alternative to line power is to use engine-generator sets to generate the electrical power. This is an impractical approach because of the unnecessary loss of efficiency introduced to the system and the cost of nonessential equipment. If an engine is to be the primary source of power, the engine can drive the pumps eliminating the need for electric motors and generators.

A disadvantage of electric-motor-driven pumps occurs when it is desirable to vary pump operating speeds to meet changes in operational requirements. Variable-speed electric motor controls are expensive, have low efficiencies, and are complicated. An alternative to variable-speed motors is to equip each pump station with several pumps of varying capacities and discharge heads. Manifolding these pumps together so they can be operated in various parallel and/or series configurations allows choosing the combination of pumps that most nearly meets the desired operating conditions.

A less desirable approach for varying the discharge conditions for a fixed-speed electric-motor-driven pump station is to install a throttle valve in the discharge line. This allows infinitely variable discharge conditions achieved through an inefficient sacrifice of discharge pressure. At best, the need for variable control of the discharge from an electric-powered pumping station will result in increased cost and complexity of operation.

(2) **Gasoline Engines**. In terms of sheer numbers, the reciprocating piston, spark-ignition internal combustion engine is the prime mover most widely used today for mobile or portable equipment applications. More commonly referred to as gasoline engines, they are produced by numerous manufacturers in a wide variety of sizes with power ratings up to 200 brake horsepower (bhp). Industrial models of gasoline engines are available from a few manufacturers with power ratings up to 300 bhp.

In the power range below 30 bhp, nearly all engines in use today are spark-ignition, burning gasoline. Many of these small engines are air cooled for simplicity and to reduce their weight, size, and cost. The relative low initial cost of gasoline engines make it extremely difficult for other types of engines to be competitive for applications where power requirements are low and the cost of the fuel consumed is not an overriding factor in the total life cycle cost.

Gasoline engines are comparatively simple to operate. Although regular maintenance is required to keep them operating properly, they are relatively easy to maintain. The recent introduction of electronic ignition systems on some gasoline engines has greatly reduced the required maintenance but adds to the complexity of the ignition system. Electronic fuel injection and super charging can be added to gasoline engines, but the gains in performance do not justify the associated increase in cost and complexity.

Gasoline engines operate well over a broad range of speeds and under varying load conditions. They operate most efficiently at rated speed and power setting but do not suffer excessive losses in efficiency at low speeds and under partial loads.

Although there is no definitive Army policy in this matter, the trend today is away from using gasoline engines on new items of military equipment. The goal is greater fuel economy and reduction of the number of types of fuels used. Informal guidance furnished by TRADOC indicates that no future pipeline pumps should be powered by gasoline engines. On this basis, gasoline-engine-driven pumps are not considered in this study even though some data are presented for comparison purposes.

**(3) Diesel Engines.** Reciprocating piston, compression-ignition, internal-combustion engines, better known as diesel engines, are rapidly replacing gasoline engines for most applications other than for very low power service, passenger cars, and other light vehicles. In the power range above 30 bhp, diesel engines are competitive with gasoline engines, particularly for operations requiring long life.

Medium-duty, high- and medium-speed industrial engines ranging in horsepower output from the very small engines to in excess of 1200 bhp are available from the major diesel engine manufacturers. In general, these diesel engines weigh from 1.5 to 2.5 times as much as comparable gasoline engines and may cost 3 times as much. However, diesel engines are much more rugged and reliable than gasoline engines; they have the ability to operate for long periods with little or no maintenance and require fewer overhauls during a substantially longer service life than do gasoline engines.

Standard models of heavy-duty, low-speed diesel engines are available with power ratings exceeding 10,000 bhp. Special marine and stationary versions may exceed 40,000 bhp. Produced by only a few manufacturers, the smaller sizes of these engines may weigh twice as much as comparable medium-duty diesel engines and the cost may be several orders of magnitude higher. The major advantages of these heavy-duty units include their low specific fuel consumption and the ability to operate continuously for periods of 3 to 5 years before requiring overhaul. Designed to withstand repeated overhauls, these large engines virtually have an infinite service life.

Because of their immense size and weight, it is frequently necessary to ship the large heavy-duty diesel engines to the installation site partially disassembled. Thus, these units are impractical for use other than at permanent installations.

Current production of diesel engines offers a wide choice of models and designs for every power requirement throughout the range from a few horsepower to tens of thousands of horsepower. The unique feature of diesel engines is that over their entire power range, they have a comparatively high efficiency. This high efficiency is a major advantage, since each diesel engine may be operated at any power setting and at varying speeds with little or no loss of efficiency. No other type of prime mover can claim this capability. As a result, the diesel engine is superior where fuel economy is important, particularly if operating conditions include varying loads and speeds.

Most research and development effort over recent years has beer devoted to increasing the power output from a given displacement, improving fuel consumption, and increasing engine reliability and service time between overhauls. The demand for higher specific power outputs and increased efficiencies have been met by design changes which have increased the mechanical and thermal stresses on the engine structures. However, the availability of improved materials and lubricants has allowed advances in these areas, accompanied by improvements in engine reliability and longer periods of operation before overhaul.

Predictions for future improvements of diesel engines reflect increases of 20 to 25 percent in power output for a given displacement with no significant increase in weight or loss in economy, reliability, or service life. Drastic changes in diesel engine designs and performance characteristics are not foreseen in the near Future. However, breakthroughs may be realized through the use of concepts such as variable-compression ratios or free-piston engines.

(4) **Gas-Turbine Engines.** The gas-turbine engine gained its first real success as a prime mover late in World War II as it rapidly took over the aircraft propulsion field. Characteristics of the gas-turbine giving it ready acceptance by the aircraft industry were a high power-to-weight ratios, good reliability, and low maintenance. These advantages were considered to override the reputation of gas-turbines for high fuel consumption, particularly at less than full-load conditions.

The pipeline industry became the first significant user of gas-turbine engines outside the aircraft industry when El Paso Nautral Gas Co. installed 28 gas-turbine-driven compressors on a gas-transmission pipeline early in the 1950's. Low-cost fuel available from the pipeline, low initial cost, ease of installation, reduced maintenance, and suitability for remote control made the gas-turbine engines competitive with other types of prime movers where power requirements were high.

Since that time, gas-turbine engines have gained wide acceptance in gas-transmission pipelines with 62.9 percent of the installed horsepower being gas-

turbine power by 1971.[12] Gas-transmission pipelines are an ideal application for gas-turbine engines because large centrifugal and axial-flow compressors can be driven at the turbine shaft speed. The high output shaft speed of the gas-turbine must be reduced substantially to be compatible with liquid pipeline pump rotating speeds. Despite the need for gear speed reducers, the gas-turbine has acquired a part of the liquid pipeline pump market, although acceptance as a pump drive has been substantially slower than for driving gas compressors.

The greatest acceptance of the gas-turbine by the pipeline industry has been for offshore applications where the high power-to-weight ratio, low vibration, high reliability, low maintenance requirements, and suitability for remote operating techniques including control, condition-monitoring and failure-detection systems are important. The rapid rise in fuel costs since 1973 has made the high fuel consumption rates for gas-turbines a more significant factor in the total life cycle cost of a unit and has tended to reduce the rate of acceptance. However, during this same period a substantial increase in the use of gas-turbines for electric-power generation has resulted in production economics reducing the initial acquisition cost.

The use of turbine engines for electric-power generation began to grow rapidly following the Northeast blackout in November 1965. By using standard gas-turbine mechanical-drive packages to achieve production cost savings, gas-turbine engines became cost competitive with other types of prime movers for generating electrical power during periods of peak power consumption and to meet temporary and emergency power generation requirements. As power demands have grown faster than new fossil fuel and nuclear power plants have been completed, many gas-turbine peaking units have been forced into service for longer periods than anticipated for normal peaking services. This experience has shown the large gas-turbine engines to be cost competitive in many applications.

The gas-turbine engines being marketed today as standard models include a mixture of heavy-duty units designed specifically for industrial applications and of aircraft engines modified for industrial use. Most of these engines have continuous power ratings in excess of 1,000 bhp. There are few commercial models of gas-turbine engines available with continuous power ratings below 500 bhp, and the selection is only slightly better in the power range from 500 to 1000 bhp.

The efficiency of gas-turbine engines drops rapidly when they are being operated under loads less than 85 percent of the maximum continuous power ratings. Thus, it behooves the equipment designer to select a gas-turbine engine having a continuous power rating matched closely to the normal operating load. This is

12 "Cost of Pipeline and Compressor Station Construction under Non-Budget Type Certificate Authorizations as reported by Pipeline Companies in Fiscal Year 1971 (July 1970 through June 1971)," report of Federal Power Commission Bureau of Natural Gas.

frequently precluded by the lack of standard models in many power ranges. The high cost of engine development makes it impractical to develop a new gas-turbine engine for a specific application unless a high utilization of the engine can be foreseen. Both gasoline and diesel engines enjoy a decided advantage in this respect because (a) their efficiency is not reduced significantly when operating under partial load and (b) there are numerous standard models available with virtually any continuous power rating that may be required.

**(5) Engine Derating Factors.** The performance of an internal-combustion engine varies with altitude and temperature. Engine manufacturers typically provide brake horsepower rating data corrected to SAE test code J-816A standard conditions of 500 feet altitude and 85°F ambient. Sea level and 60°F ambient is another set of conditions frequently used for rating engines. In either case, the engines must be derated to account for the loss in power which results if the engine is operated at a higher temperature and/or altitude than rating conditions.

The recommended derating factors vary between types of engines as well as between manufacturers of the same type of engines. The derating factors for gasoline engines and naturally aspirated diesel engines follow the same general pattern. The most conservative procedures recommended for these engines require derating 3 percent for each 1,000 feet above sea level and 1 percent for each 10 degrees above 60°F. A few manufacturers indicate that no derating is required up to 5,000 feet altitude and 85°F. The most common practice for derating gasoline engines and naturally aspirated diesel engines is to reduce the standard values by 3 percent for each 1,000 feet above 500 feet altitude and 1 percent for each 10 degrees above 85°F.

Turbocharging a diesel engine overcomes much of the effect of higher altitudes. As a result, the recommended derating factors are less than for naturally aspirated engines. A significant number of manufacturers do not require derating at altitudes and temperatures up to 5000 feet and 85°F. The largest derating factors identified for turbocharged engines are 2 percent for each 1000 feet of altitude above sea level and 1 percent for each 10 degrees above 60°F.

Gas-turbine engines suffer the greatest loss in power output due to increased temperatures and higher altitudes. In addition, air filters which produce inlet losses and exhaust gas silencers causing back pressure add significantly to the required derating factors. An altitude correction factor of 0.75 is representative of most manufacturers recommended derating at 5000 feet altitude and 85°F. In addition, the ratings taken at atmospheric pressure must be further reduced by approximately 0.5 percent for each inch of water pressure loss at the turbine inlet or back pressure at the turbine exhaust. It is unlikely that these pressure losses can be kept below 6 inches of water representing an additional power reduction of approximately 3 percent. Thus, a

power correction factor of 0.72 is representative of the required derating for a gas-turbine capable of operating at altitudes up to 5000 feet.

c. **Performance Characteristics of Centrifugal Pumps.** Although a variety of factors determine the performance characteristics of a centrifugal pump, pump performance is usually defined by four parameters common to all pump designs:

(1) Capacity is the rate of low, usually expressed in gallons per minute.

(2) Head or Total Dynamic Head (TDH) is the total pressure increase produced by the pump, most conveniently expressed as the height, in feet, to which the pump can lift the fluid being pumped.

(3) Specific speed is a dimensionless number that describes the internal geometry of the pump.

(4) Net positive suction head (NPSH) is a measure of the energy conditions of the fluid as it enters the suction side of the pump, usually expressed in feet of fluid, absolute.

Variation of the head with capacity at a constant speed is called the pump characteristics. In addition, the characteristics of a pump include the relationship of pump efficiency and the brake horsepower required to drive the pump. Any change in the flow characteristics of the system in which a centrifugal pump is operating will be accompanied by a change in capacity, head, brake horsepower, and efficiency. When the pump speed is changed, the pump performance characteristics change. These phenomena are illustrated in Figure 8 by the performance curves for the military standard 4-inch, four-stage pump when operating at 1800 and 2000 rpm.

The variation of head, capacity, and brake horsepower follow definite affinity laws. These rules are:

(1) The capacity of a pump changes in direct proportion to the speed of the pump. Doubling the pump speed doubles the capacity.

(2) The head developed by a pump changes directly as the square of the speed. Doubling the pump speed increase the head by a factor of four ($2^2$).

(3) The brake horsepower required to drive a pump increases in direct proportion with the cube of the speed. Doubling the speed increases the brake horsepower by a factor of eight ($2^3$).

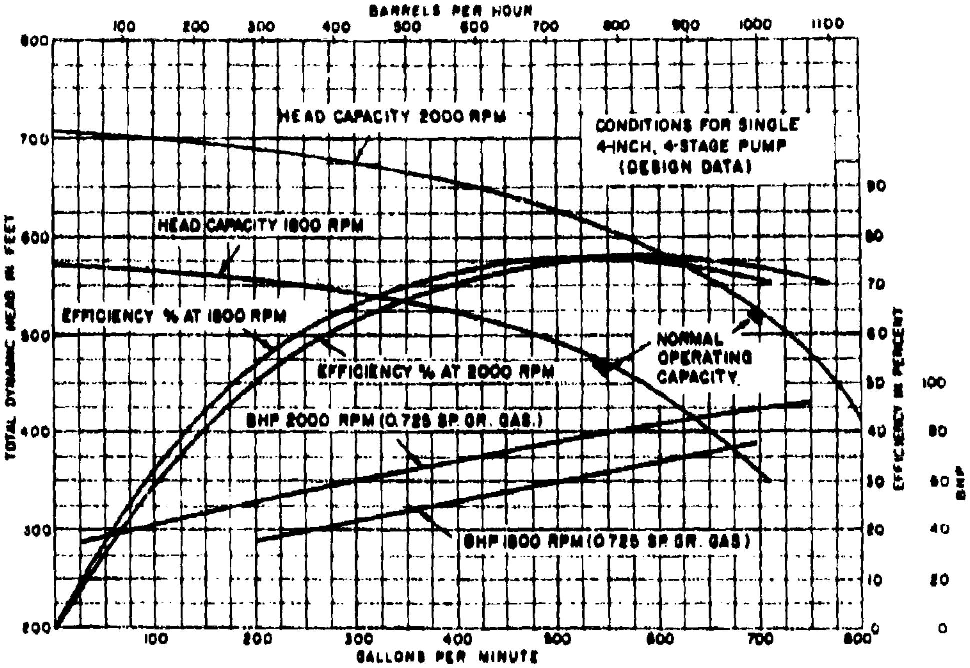



**Figure 8. Performance characteristics of a 4-inch, four-stage pump.**

For example, consider a pump which would deliver 1,000 gpm at 500 feet of head and require 160 bhp when operating at 1,800 rpm. The performance of this pump if driven at 2,000 rpm would be determined as follows:

(1) Since capacity varies directly with speed, the rate of flow at 2000 rpm would equal 1,200 gpm x (2,000/1,800), or 1,111 gpm.

(2) Increasing the head in proportion to the square of the speed, the head at 2,000 rpm would be 500 x $(2,000/1,800)^2$ or 617 feet.

(3) Increasing the brake horsepower by a ratio equal to the cube of the change in speed finds the new required brake horsepower equals 160 x $(2,000/1,800)^3$, or 219 bhp.

The first step in the selection of a centrifugal pump for a particular application is to determine the required rate of flow and pressure rise. These performance requirements are usually established by the system requirements. The next step is to establish the pump operating speed. If the pump is to be close-coupled to the prime mover, the speed will be determined by the prime mover selected to drive the pump. Generally a gear-type speed increaser or decreaser can be used between the prime mover and the pump to obtain a more desirable pump speed.

The pump performance characteristics under normal operating conditions (i.e., capacity, head, and impeller rotating speed) are used to determine the basic physical design features of the pump. As defined previously, specific speed is a dimensionless number that describes the internal geometry of a centrifugal pump. The specific speed value is the same for all centrifugal pumps of the same geometric shape, regardless of size. The formula for computing specific speed ($N_s$) is:

$$N_s = \frac{NQ^{1/2}}{H^{3/4}}$$

where:

N = impeller rotational speed, rpm.
Q = flow rate, gpm.
H = pressure developed by the pump, feet of fluid.

The impeller vane shapes for various specific speeds illustrated in Figure 9 provide the most efficient pump performance. In general, the higher the impeller rotational speed, the smaller the pump can be and still develop the required discharge pressures. The cost and weight of a pump are a direct function of the size. Therefore, it is desirable to design centrifugal pumps using the highest specific speed consistent with other system performance requirements.

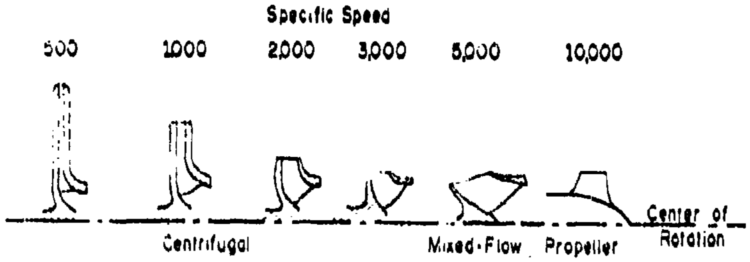



**Figure 9. Impeller vane shape versus specific speed.**

The upper limit for specific speed is usually a function of net positive suction head (NPSH). By definition, NPSH is the pressure at the inlet of the pump (read in feet of liquid and corrected to the pump centerline), minus the vapor pressure of the liquid at the pumping temperature, plus the velocity head of the liquid at the pump inlet. Two NPSH values, required and available, must be considered.

The available NPSH is a characteristic of the system in which the pumps are located and is the difference between the absolute pressure at the pump inlet and the vapor pressure of the liquid. Available NPSH may include the effects of atmo-

spheric pressure, static head due to difference in elevations in the suction manifold, and pressure from other pumps located upstream in the pipeline system.

A liquid must be pushed into the impeller of a centrifugal pump. The required NPSH is the pressure, in feet of liquid, at the pump inlet required to push the liquid into the impeller at the required rate of flow. The required NPSH is a function of pump design and must be determined by testing. Although there is no simplified method for determining the required NPSH, there are some known relationships. For a given geometric shape and size, required NPSH varies in direct relation with specific speed.

Unless the available NPSH exceeds the required NPSH, cavitation will occur in the inlet of the pump. When this happens, small vapor bubbles form in the liquid in the low-pressure area of the pump suction. As the liquid passes through the pump, the increasing pressure causes these vapor bubbles to collapse. The results are usually a drop in pump capacity, discharge pressure, and efficiency accompanied by severe pitting and erosion of the impeller vanes. Therefore, it is imperative the suction pressure at the inlet to each pump be equal to or greater than the required NPSH.

The most efficient pipeline design is based on operating at the highest pump discharge pressure possible within the safe limits of the pipeline and the lowest possible pressure at the inlet to pump stations. The difference between these two pressures determines the allowable pressure loss between pump stations. Since pressure loss through a pipeline at a given flow rate is a function of line length, the maximum allowable pressure drop between pump stations provides the maximum spacing between pump stations.

From the preceeding discussion it becomes evident there is conflict between pump and pipeline design goals regarding NPSH. In pump design, the goal is to use the highest possible specific speed to minimize pump size, weight, and cost. Since the required NPSH increases with specific speed, the incentive in pump design is toward high NPSH values. In contrast, efficient pipeline design demands minimizing the required NPSH. As a result, a tradeoff of NPSH requirements must be made to determine the most effective system design.

In the simplest form, a centrifugal pump would be a single impeller with the capability to develop the total pressure rise desired across the pump. Many single-impeller pumps are used for low-pressure applications. However, high-pressure pipeline operational requirements normally exceed the performance capability of single-impeller pumps. The desired performance is then achieved by using two or more impellers operating in series or parallel. This may be accomplished by including more than one impeller in the same case as a multistage pump or by using more than one pump unit at a pump station. Normally pipeline throughput requirements impose per-

formance, maintenance, and reliability requirements that cannot be satisfied by a single pump-engine assembly.

The use of redundant components is an effective means of improving system reliability. The effects the use of multiple-unit pump stations have on the reliability and maintainability of a complete pipeline system are examined in paragraph 10 of this report.

When two or more pumps of equal rating are operated in parallel, the combined capacity of the pumps at any discharge pressure is a multiple, equal to the number of pumps on line, of the capacity of a single pump operating at the same discharge pressure. Similarly, when two or more pumps of equal rating are operating in series, the combined discharge pressure at any capacity is an equal multiple of the discharge pressure of a single pump operating at the same capacity. Figure 10 shows the head-capacity curves for a single pump, two pumps operating in parallel, and two pumps operating in series.

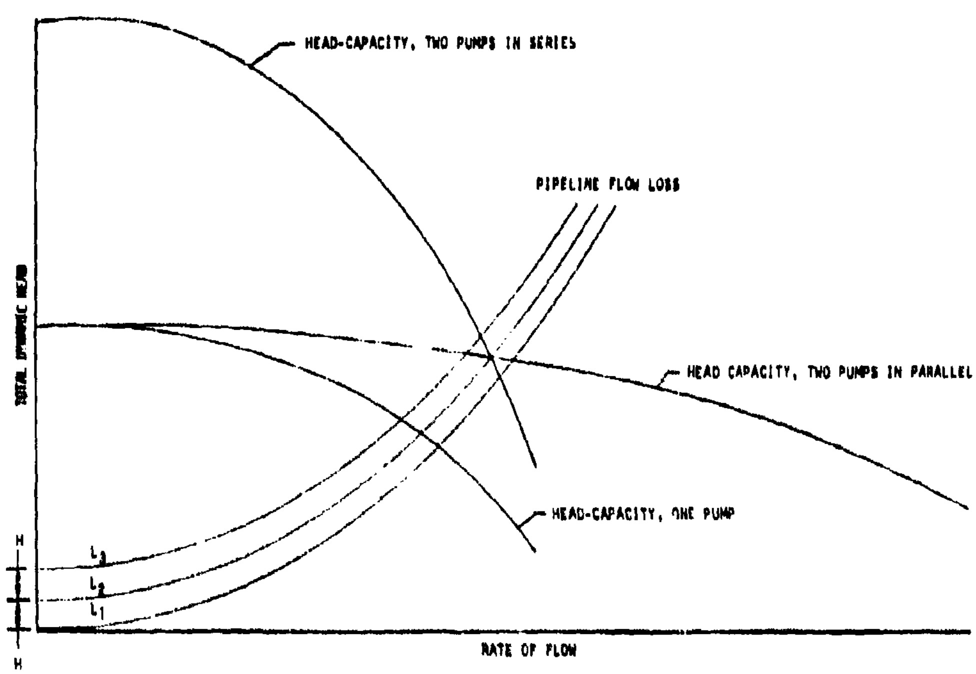



Figure 10. Head-capacity curves for pumps operating in parallel and series.

To determine whether the maximum flow rate from two pumps will be obtained with the pumps in series or parallel, the pump head-capacity curves and pipeline flow loss curve must be plotted. For example, if curve $L_1$ in Figure 10 represents the dynamic flow loss through a pipeline, the maximum flow rate will be obtained if the pumps are operated in parallel at the intersection of curve $L_1$ with the head-capacity curve for two pumps in parallel. If the discharge end of the pipeline is elevated to a height H above the pump station, the total dynamic head loss through the pipeline is represented by curve $L_2$. Under these conditions, the flow rate would be the same for both parallel or series pump operation since the flow loss curve intersects the pump head-capacity curves at their intersection point. Elevating the discharge end of the pipeline to a height of 2H above the pump station shifts the total dynamic head loss through the pipeline up to curve $L_3$. The highest flow rate would then occur if the two pumps were operated in series at the intersection of curve $L_3$ and the head-capacity curve for two pumps operating in series. Thus, if the normal rate of flow lies to the left of the intersection of the parallel and series head-capacity curves, the highest flow rate will result if the pumps are in series. If the normal rate of flow lies to the right of the intersection of the parallel and series head-capacity curves, parallel operations will result in the highest rate of flow.

In Figure 9, a shift in operating conditions from curve $L_1$ to $L_3$ would result in a smaller change in flow rate if the pumps were operating in series rather than parallel. At the same time, the change in operating pressure would be the greatest if the pumps were in series. Thus, under variable flow conditions, series pump operation provides the most stable rate of flow, while parallel operation achieves a more stable operating pressure. From this comparison it is apparent that the stability of the rate of flow and operating pressure is a function of the slope of the pump head-capacity curve. As a general rule, the most stable and efficient pipeline operation is achieved with all booster pumps operating in series.

d. **Comparison of Pump-Engine Assemblies.** Existing military pipeline pump units consist of pumps which are coupled to engines and mounted on rugged skid-type bases. Mounted on the same skid are all the accessories, including the radiator, starting system, controls, etc., necessary for each unit to be entirely self-contained. This provides portable units which are ready for operation as soon as they can be moved into position and connected into the pipeline pump station manifold. Site preparation is limited to the grading required to provide a relatively flat area for the pump station.

The pump industry has available numerous standard models of pumps suitable for use with all types of prime movers. Most of the major pump manufacturers include in their standard product lines a series of gasoline-engine-driven and diesel-engine-driven pump assemblies, skid- or trailer-mounted as self-contained units. This includes all sizes of units from low-capacity units to units capable of flow rates exceed-

ing 5,000 gal/min. Typically, these units are designed for low-pressure application with the maximum operating pressure seldom exceeding 125 $lb/in^2$ . These standard model pump-engine assemblies offer reliable performance at a relatively low cost.

Although standard models are available, pipeline pump-engine assemblies are almost always designed for each application to provide the desired flow rate and operating pressures, automatic controls, and other special features. Most pump manufacturers modify existing designs as necessary to tailor the pump performance to a specific application. The cost of a pump-engine assembly increases rapidly as special design requirements are imposed. Because of the need to tailor pumps to specific needs, most pumps are produced to order. Only a few makes and models sell in sufficient quantities to justify the manufacturer maintaining pumps in stock for off-the-shelf delivery.

Design of highly efficient pump units requires careful matching of engine performance to pump power requirements. Ideally, the power required to drive a pump operating under a fixed set of conditions would be equal to the maximum continuous horsepower rating of the engine. This is seldom possible because few pump units actually operate under fixed conditions at all times. For military applications, the variables in operating conditions include capacity, head, fluid, elevation, and environmental conditions, all of which affect pump or engine performance.

For the purpose of the analyses herein, the following criteria apply to all pump-engine assemblies: Although the pump units are primarily for petroleum pipeline service, all engines must have continuous power ratings, when derated for operation at 5000 feet altitude and 85°F ambient, which equal or exceed the brake horsepower required if pumping water; all comparisons of pump unit cost, weight, and size versus brake horsepower are based on the power required when pumping water at the head and capacity corresponding to the best efficiency point of the pump.

(1) **Procurement Costs.** The cost of pump engine assemblies is highly dependent on design requirements. With application of the method of least squares for linear regression analysis to the list price of 30 standard models of gasoline-engine-driven pumps, the relationship between cost and brake horsepower is determined to be represented by the equation:

$$\text{Cost} = 1{,}770 + 43.9\ \text{BHP} \qquad \text{(Eq. 1)}$$

where

Cost = average list price in dollars.
BHP = derated continuous brake horsepower rating of the engine.

As noted previously, the standard model pump engine assemblies are typically for low-pressure applications. Further, they are generally single-stage pumps with few, if any, automatic controls. Thus, Equation 1 represents the cost of what might be considered basic flood-and-transfer pumps. Because of the low-pressure ratings of the pump cases, however, these pump units would be suitable for use only within a tank farm where there is no chance of their being over pressured.

The cost of similar skid-mounted, gasoline-engine-driven, multi-stage pumps suitable for pipelines operating at pressures from 300 to 500 lb/in² can be found using the equation:

$$\text{Cost} = 2{,}353 + 57.6\ \text{BHP} \qquad \text{(Eq. 2)}$$

The estimated maximum cost for complex high-pressure, gasoline-engine-driven, multistage pipeline pumps is expressed by the equation:

$$\text{Cost} = 16{,}075 + 70.4\ \text{BHP} \qquad \text{(Eq. 3)}$$

The wide variation in the potential cost of gasoline-engine-driven pumps is illustrated graphically in Figure 11. The most likely cost of gasoline-engine-driven pumps suitable for military pipeline operations would fall somewhere between the cost of the commercial low-pressure unit (Eq. 2) and the projected upper cost limit (Eq. 3). Therefore, for the purpose of cost comparisons herein, the cost of gasoline-engine-driven pipeline pumps is considered to be the mean between Equations 2 and 3 or:

$$\text{Cost} = \frac{(2{,}353 + 57.6\ \text{BHP}) + (16{,}075 + 70.4\ \text{BHP})}{2}$$

$$\text{Cost} = 9214 + 64.0\ \text{BHP} \qquad \text{(Eq. 4)}$$

Using a similar cost analysis process, it is determined that a reasonable estimate of cost for military pipeline pumps powered by medium- or high-speed, medium-duty diesel engines can be computed using the equation:

$$\text{Cost} = 13{,}500 + 100.7\ \text{BHP} \qquad \text{(Eq. 5)}$$

There are no standard model pump-engine assemblies using low-speed, heavy-duty diesel engines. Because of the high cost, it is essential that each unit be designed for the specific application where it will be used. Pump units will normally utilize a standard model basic engine and the pump may be a modification of a standard design. However, the integration of controls, accessories, and other special design requirements results in highly individualized designs. The cost of low-speed, heavy-

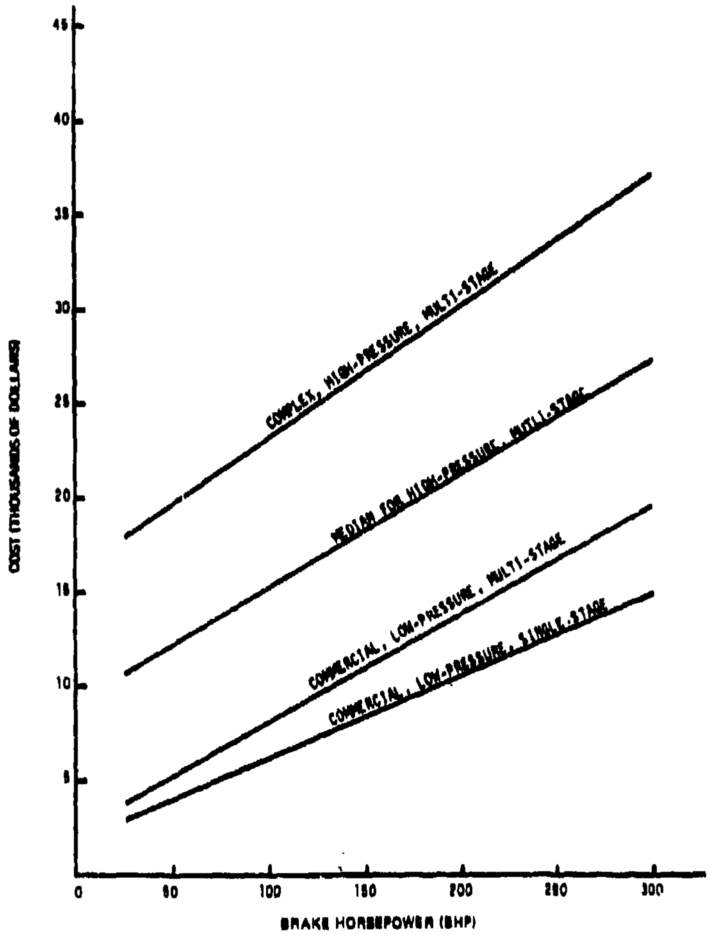



**Figure 11. Cost of gasoline-engine-driven pump units.**

duty diesel-engine-driven pump units, based on the costs of the various engines, pumps, and accessories, is projected to be in the range represented by the equation:

$$\text{Cost} = 36{,}100 + 109.7\ \text{BHP} \qquad \text{(Eq. 6)}$$

Gas-turbine-engine-driven pumps, like low-speed, heavy-duty diesel-engine-driven units, are not produced as standard models. Some engine manufacturers market standard engine modules or mechanical-drive packages which are easily adapted to drive pump units. Based on the cost for these units, the cost of a gas-turbine-engine-driven pump unit can be approximated using the equation:

$$\text{Cost} = 57{,}500 + 169.4\ \text{BHP} \qquad \text{(Eq. 7)}$$

Equations 4, 5, 6, and 7 are graphed in Figure 12. These data allow a general comparison of pump costs over the range of brake horsepower shown. A degree of caution must be exercised, however, if Figure 11 is to be used to estimate the cost of pump units of a given power rating. First, because the number of standard model gas-turbine engines available (especially below 1,000 bhp) is limited, the curve for gas-turbine engines is valid only for brake horsepower values where standard model engines are available. Second, a large demand for a gas-turbine engine of a particular size would result in a substantial cost reduction through production economics. And finally, the curves represent, *at best*, a general guide to the cost of pump-engine assemblies and should not be construed to reflect the exact cost of any specific unit.

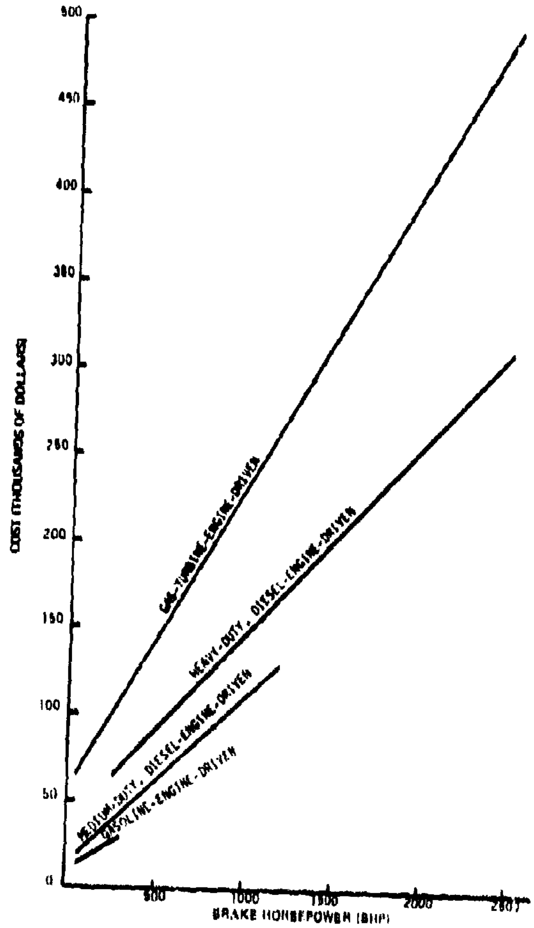



**Figure 12. Cost of engine-driven pump units.**

The curves for gasoline and medium-duty diesel engines can be considered continuous since, among the various makes and models, it is possible to select an engine of virtually any conceivable brake horsepower desired. There are substantially fewer makes and models of heavy-duty, low-speed diesel engines available. However, most heavy-duty engine manufacturers offer several engine models with the number of cylinders per engine optional and ranging from as few as 3 to as many as 18. As a result, standard model heavy-duty diesel engines are available across the entire power range.

(2) **Transportability**. Guidance and procedures for use by materiel and combat developers during the materiel acquisition process is contained in AR 70-47, "Engineering For Transportability," dated 28 January 1976. The transportability criteria imposed by AR 70-47 is dependent upon the mode of transportation to be used. To minimize the possibility that a pump unit may be denied movement or unacceptably delayed, this analysis assumes the requirements for water, rail, truck, and air transportability apply. Further, it is assumed the weight and size of pump units shall be limited to the following requirements for transportation in U.S. Air Force aircraft as specified in Appendix F of AR 70-47: Length – 20 feet; width – 8 feet; height – 8 feet; weight – 20,000 pounds.

These dimensions are consistent with other modes of transportation conforming to the cross-section for International Organizations for Standardization (ISO) and American National Standards Institute (ANSI) Series I containers. The 20,000-pound weight limit is also consistent with the maximum boom lift capability for many lighters and barges.

The approximate weight of a pump unit can be determined using Figure 13. Applying the criteria that all pump units must weigh less than 20,000 pounds eliminates all sizes of heavy-duty, diesel-engine-driven pump units from consideration. The curve for the weight of medium- or high-speed, medium-duty diesel engine is represented by the equation:

$$WT = 9{,}000 + 36.2\ BHP. \qquad \text{(Eq. 8)}$$

Substituting the maximum allowable weight of 20,000 pounds, the upper limit of brake horsepower using medium-duty, high- or medium speed diesel engines is computed as follows:

$$20{,}000 = 9{,}000 + 36.2\ BHP,$$

or

$$BHP = 304 \text{ brake horsepower.}$$

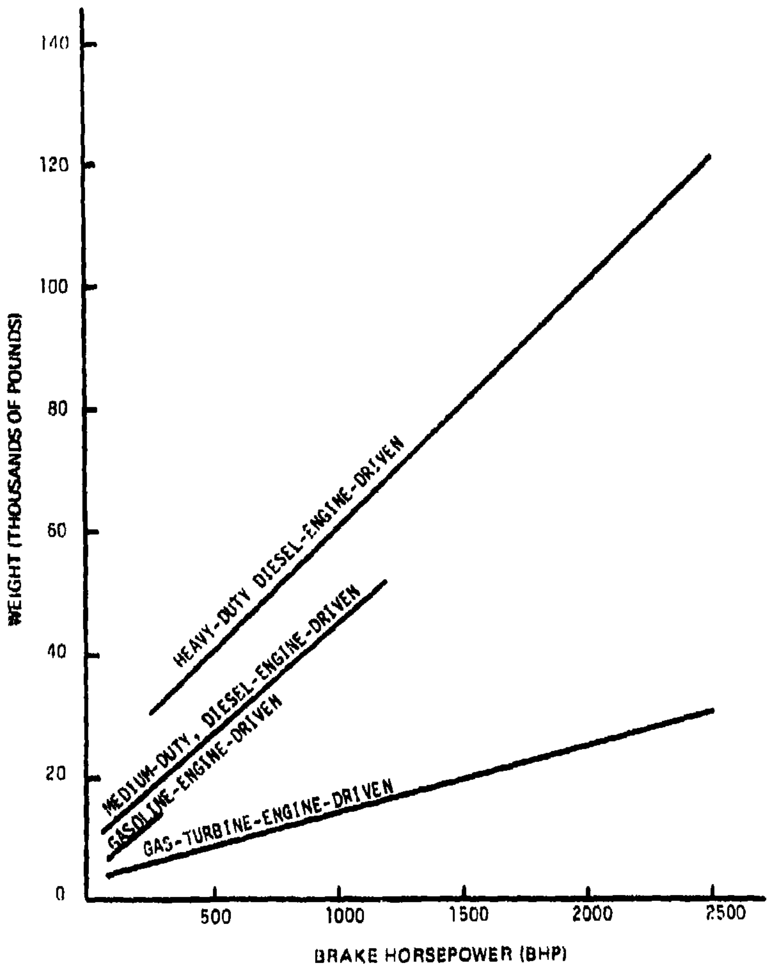



**Figure 13. Weight of engine-driven pump units.**

The curve in Figure 13 for the weight of gas-turbine-engine-driven pump units can be expressed mathematically as

$$WT = 3590 + 11\ BHP. \qquad \text{(Eq. 9)}$$

Substituting 20,000 pounds as the maximum allowable weight, the maximum available brake horsepower is calculated to be:

$$20{,}000 = 3{,}590 + 11\ BHP,$$

or

$$BHP = 1{,}492 \text{ brake horsepower.}$$

The weight of gasoline-engine driven pump units is represented by the equation:

$$WT = 4170 + 36\ BHP. \qquad \text{(Eq. 10)}$$

Using this equation, the maximum brake horsepower for a gasoline-engine-driven pump unit would be

$$20{,}000 = 4{,}170 + 36\ BHP.$$

or

$$BHP = 440 \text{ brake horsepower.}$$

This value is greater than the power rating of the largest gasoline engine in production. Thus the upper limit on the size of gasoline-engine-driven pump units is restricted by the availability of large engines, not by weight.

The relationship between the size of various pump units is shown in Figure 14. Gas-turbine engines are typically considered to be extremely compact, delivering a large amount of power from small units. In contrast to this popular view, Figure 14 shows that for units requiring less than 300 brake horsepower, both gasoline-engine-driven pumps and medium-duty diesel-engine-driven pumps are smaller than gas-turbine-engine-driven pump units. This results from the fact that bulky air inlet filters and exhaust gas silencers are required by gas-turbine engines.

Applying the dimensional limits from AR 70-47, the largest acceptable unit is 8 feet by 8 feet by 20 feet, or 1280 cubic feet. The relationship between volume and brake horsepower for gasoline-engine-driven pump units is expressed by the equation:

$$VOL = 42 + 0.83\ BHP. \qquad \text{(Eq. 11)}$$

A volume of 1280 cubic feet equates to 1492 brake horsepower. Thus, as with weight, the largest acceptable gasoline-engine-driven pump unit is a function of the availability of large engines, not volume.

The volume of medium-duty diesel engines can be expressed mathematically as:

$$VOL = 56 + 1.7\ BHP. \qquad \text{(Eq. 12)}$$

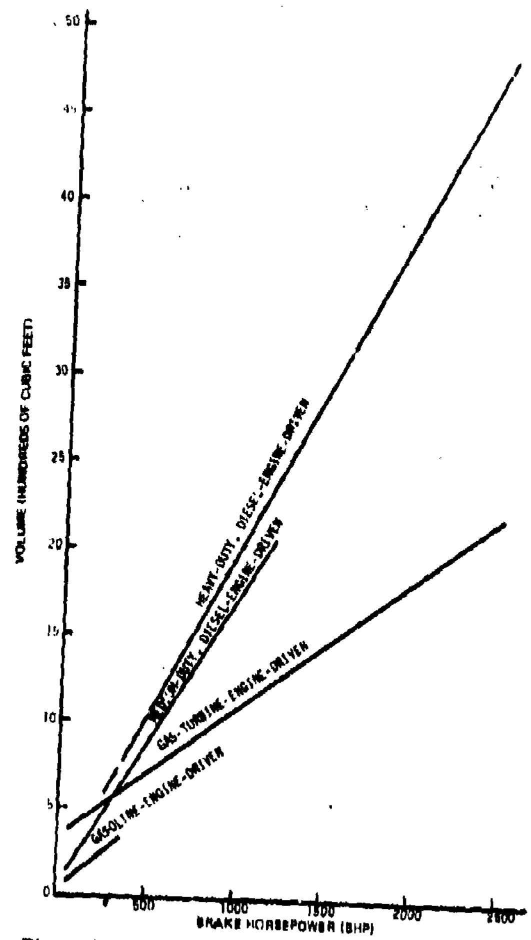



**Figure 14. Volume of engine-driven pump units.**

Using this equation to convert volume to power, 1280 cubic feet is equivalent to 720 brake horsepower. The upper limit of 304 brake horsepower computed on the basis of weight is substantially lower and is the controlling factor for size of medium-duty diesel-engine-driven pumps.

The volume versus brake horsepower curve for heavy-duty diesel engines represents the mathematical expression:

$$VOL = 115 + 1.9\ BHP. \qquad \text{(Eq. 13)}$$

Using this equation, 613 brake horsepower is equal to 1280 cubic feet. Considering only size, heavy-duty diesel-engine-driven pump units up to 613 brake horsepower could be used. However, the reader is reminded that these units were eliminated on the basis of excessive weight.

The equation for volume versus brake horsepower for gas-turbine engine-driven pump units is:

$$VOL = 350 + 0.75\ BHP. \qquad \text{(Eq. 14)}$$

A volume of 1280 cubic feet converts to 1240 brake horsepower using this equation. The power limit computed on the basis of weight is 1,492 brake horsepower resulting in volume being the factor controlling the size of gas-turbine-engine-driven pump units. This limit on unit size could become even more restrictive if the unit cannot be configured to prevent one dimension, width, length, or height, from exceeding the acceptable limit before the other two dimensions reach their respective limits.

The foregoing calculations are summarized in Table 6, showing the largest pump unit of each type that will conform to the transportability requirements of AR 70-47 without approval of amended transportability characteristics. These horsepower ratings are based on projected weights and dimensions of average pump units and, therefore, should not be considered absolute maximum values. Through tradeoff of various design characteristics it may be possible to develop slightly larger units within the weight and size limits.

Table 6. Transportability Limits on Pump Units

| Type Engine | Maximum Brake Horsepower, Derated | Limiting Factor |
|---|---|---|
| Gasoline | 440* | Weight |
| Medium-Duty Diesel | 304 | Weight |
| Heavy-Duty Diesel | 0 | Weight |
| Gas-Turbine | 1240 | Volume |

* Standard commercial models of gasoline engines this large are not readily available.

(3) **Fuel and Lube Oil Consumption.** The cost of the fuel consumed by an engine-driven pump unit represents a major portion of the total cost of operation and maintenance. Figure 15 shows the average specific fuel consumption for gasoline, diesel, and gas-turbine engines. These data are representative of the average fuel consumption for each type of engine based on the assumption the engines are operated at a power output equal to the maximum continuous brake horsepower, rating derated for operation at 5000 feet altitude and 85°F ambient temperature.

The fuel consumption data for gas-turbine engines in Figure 15 are for simple, or non-regenerative, cycle engines. By adding a regenerator to recover heat from the exhaust gases, the efficiency of a gas-turbine engine can be improved. A regenerative gas-turbine engine will cost approximately 20 percent more, is larger and heavier, and requires more maintenance than a simple-cycle gas-turbine of an equivalent power output. Thus, a regenerative engine would be preferable only for applications where fuel consumption is of primary importance.

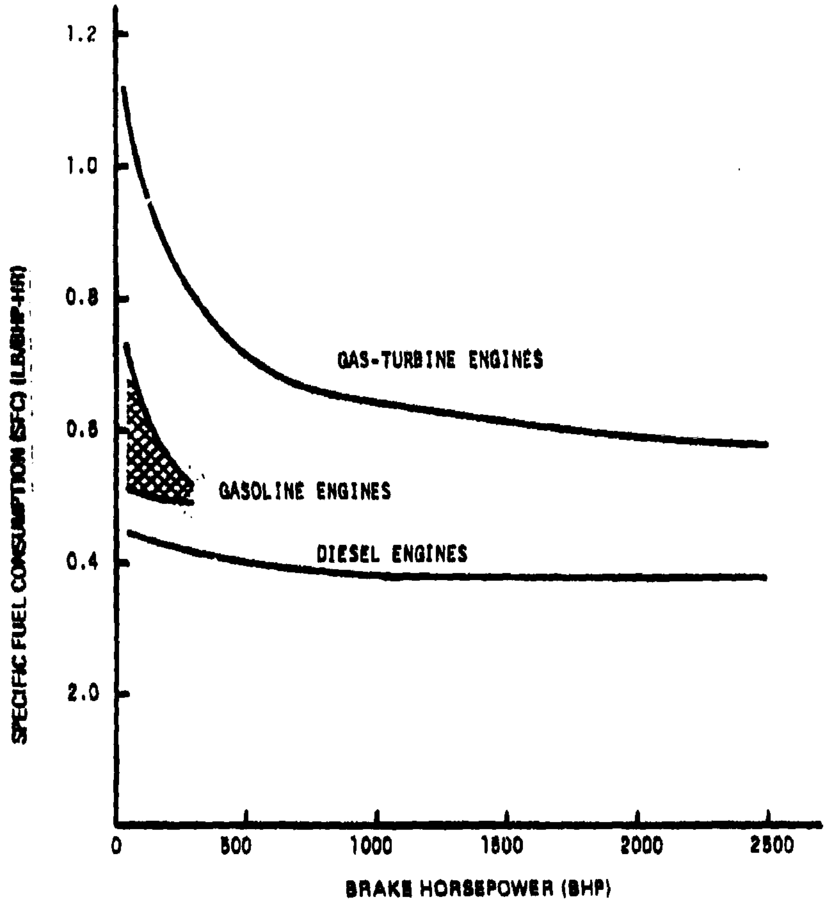



**Figure 15. Specific fuel consumption of engines.**

Most gas-turbine engines in use today are of the simple-cycle type. If regenerative gas turbines are developed and produced commercially for vehicle applications, the production base may be large enough for them to be cost competitive with simple-cycle engines. This broad commercial application is not foreseen in the immediate future. Thus, this study considers only simple-cycle gas-turbine engines.

In addition to their lower fuel consumption, diesel engines have a cost advantage in the price of fuel consumed. The Military standard prices of fuel, as of January 1976, were:

| Type Fuel | Cost (Dollars/Gallon) |
|---|---|
| Motor gasoline | \$0.381 |
| Diesel | \$0.339 |
| JP-4 | \$0.373 |
| JP-5 | \$0.355 |

Using these prices, the cost of fuel, in dollars per brake horsepower hour are shown in Figure 16. The advantages of small diesel engines in applications where fuel costs are a significant part of total life cycle costs is readily apparent in Figure 16. This cost difference becomes even greater when the engines are employed overseas adding fuel transportation costs to the basic fuel costs.

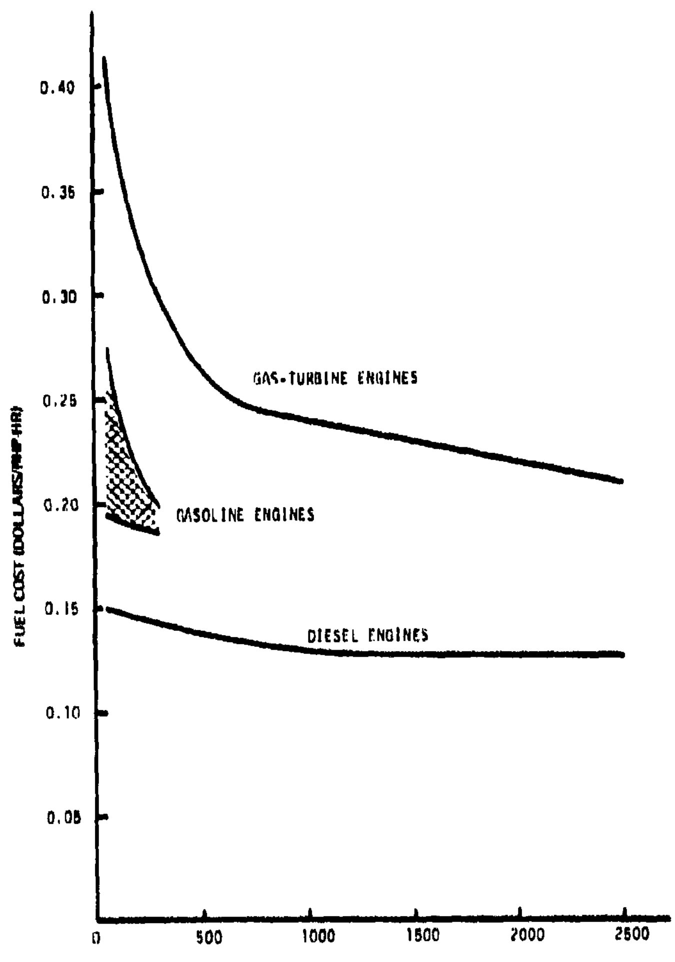



Figure 16. Fuel cost, dollars per brake horsepower-hour.

Lube oil consumption rates for gasoline and diesel engines are approximately equal and increase with engine size. Oil consumption in reciprocating piston engines generally increases as wear occurs during normal operation; however, this does not represent a significant change under normal operating conditions. Extremely high lube oil consumption is usually indicative of a serious problem meriting immediate attention to prevent serious damage to the engine. By contrast, turbine engines consume little lube oil because the lubrication systems are totally separated from the combustion process. Average rates of lube oil consumption are shown in Figure 17.

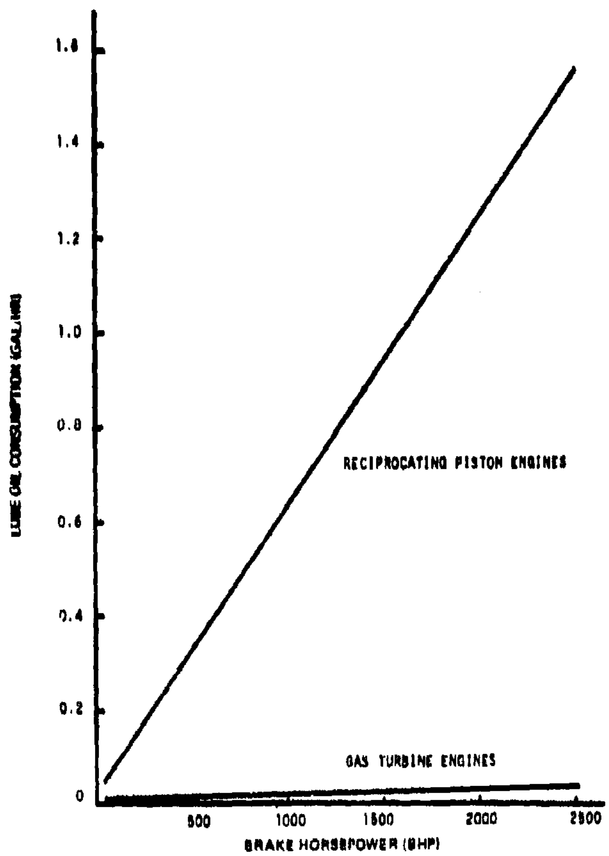



Figure 17. Engine lube oil consumption.

(4) **Maintenance.** The maintenance characteristics of a pump-engine assembly weigh heavily on the suitability for pipeline service. Operational requirements frequently demand an entire pipeline system be maintained in continuous operation for extended periods of time. Thus, the frequency of shutdowns required for maintenance may be as important as the amount of maintenence required.

Centrifugal pumps are relatively simple machines generally considered to provide highly reliable service with little maintenance. Assuming the pump is properly designed, balanced, aligned, and free from excessive stresses from piping connections, only limited maintenance will be required except for periodic replacement of bearings and shaft seals. Limited data available for the chemical processing industry indicate it is reasonable to expect centrifugal pumps to have a Mean Time Between Failure (MTBF) of at least 10,000 hours.

Because of their complexity, engines require substantially more maintenance than the pumps. As with pumps, limited data are available applicable to the maintenance requirements for engines used to drive pipeline pumps. A survey of pipeline operating companies yielded data on 1387 gasoline-, diesel-, and gas-turbine-engine-driven pumps ranging in size from less than 50 horsepower to more than 3500 horsepower. Because of the wide range of sizes and varied applications, these data do not reflect well defined pump maintenance characteristics. Therefore, the survey data have been used to develop an estimated range of values for pump unit maintenance characteristics. These data are shown in Table 7 and Figures 18, 19, 20, and 21.

Table 7. Projected Maintenance Characteristics for Military Pipeline Pump Units

| Characteristic | | Gasoline-Engine-Driven Pumps | Diesel-Engine-Driven Pumps | Gas-Turbine-Engine-Driven Pumps |
|---|---|---|---|---|
| Maintenance Ratio—expressed as ratio of maintenance manhours to operating hours | MIN | 0.20 | 0.01 | 0.01 |
| | MAX | 0.70 | 0.06 | 0.05 |
| Mean Time Between Overhaul—expressed in operating hours | MIN | 2,500 | 5,000 | 4,000 |
| | MAX | 8,000 | 12,000 | 10,000 |
| Overhaul Cost—expressed as percentage of procurement cost | MIN | 28 | 18 | 7.5 |
| | MAX | 35 | 32 | 25 |
| Expected Service Life—expressed in operating hours | MIN | 8,000 | 20,000 | 20,000 |
| | MAX | 35,000 | 50,000 | 120,000 |

Average hourly maintenance costs per operating hour are shown in Figure 18. These costs increase with the size of the pump unit; however, the maintenance costs do not increase proportionally with brake horsepower. The manhours required to perform most routine maintenance tasks are not significantly different for large or small engines. As a result, maintenance costs per brake horsepower hour decrease with engine size.

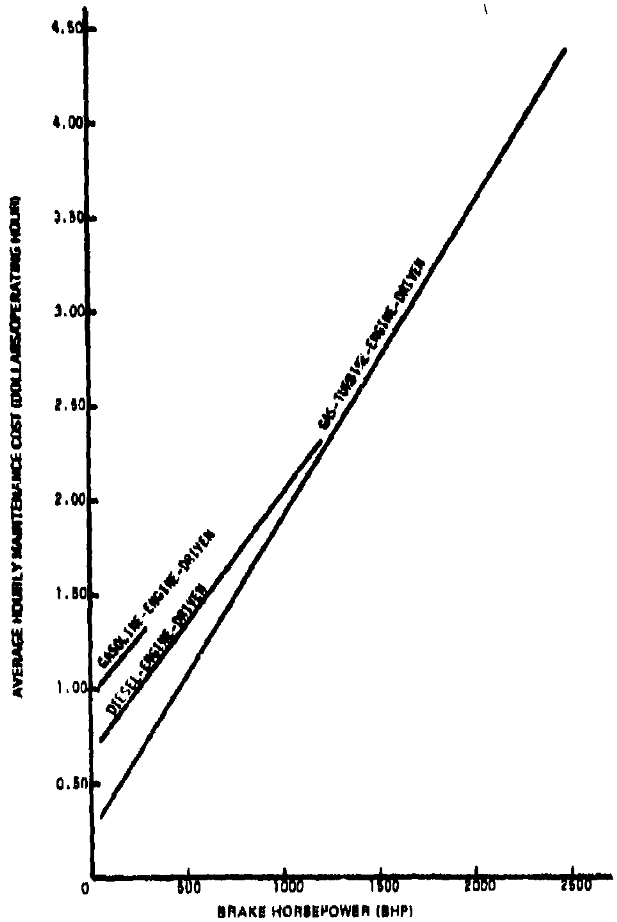



Figure 18. Hourly maintenance cost for pump units.

The average costs for overhaul of pump units are shown in Figure 19 as percentage of the intital procurement cost. Although the overhaul cost as a percentage of procurement cost is highest for gasoline-engine-driven units and lowest for gas-turbine-engine-driven units, a gas-turbine-engine-driven pump unit is still more expensive to overhaul than a gasoline-engine-driven unit of equivalent size. For example, the overhaul cost for 100-brake horsepower units would be:

| | Type of Engine | | |
|---|---|---|---|
| | Gasoline | Diesel | Gas-Turbine |
| Procurement Cost | $15,600 | $23,500 | $74,400 |
| Overhaul Cost as Percentage of Procurement Cost | 32.7% | 29.5% | 24% |
| Overhaul Cost | $5,101 | $6,933 | $17,856 |

Some gas-turbine engines are designed as modular units. These manufacturers recommend replacement of the modules "on condition" in lieu of complete engine overhaul. This has a distinct advantage in reducing overhaul costs. Some modules may operate successively for periods far in excess of the average time for overhaul of complete engines. The modular concept does not eliminate overhaul costs, since modules removed must be repaired.

Representative Mean Time Between Overhaul (MTBO) and expected service life data are shown in Figures 20 and 21, respectively. These data represent conservative expections compared to MTBO and service life data obtained from the pipeline industry. However, it is not reasonable to expect equipment operating in a military combat environment to be as durable as pumping equipment operating in the less severe and demanding environment of commercial pipelines. New materials and design of gas-turbine engines specifically for industry applications may produce significant increases in the MTBO. However, because the availability of these engines remains doubtful, this report reflects the expection for existing gas-turbine engines which are predominately aircraft engines adapted to provide output shaft power.

8. **Pipe.** Pipe selection is generally the most important decision made during the design of a pipeline. The size, material, wall thickness, and other physical properties of the pipe determine many other factors concerning the construction, operation, and maintenance of a pipeline. Because a wide range of economic and technological considerations impact on the design of every pipeline, there are no definitive guidelines to be followed in pipe selection.

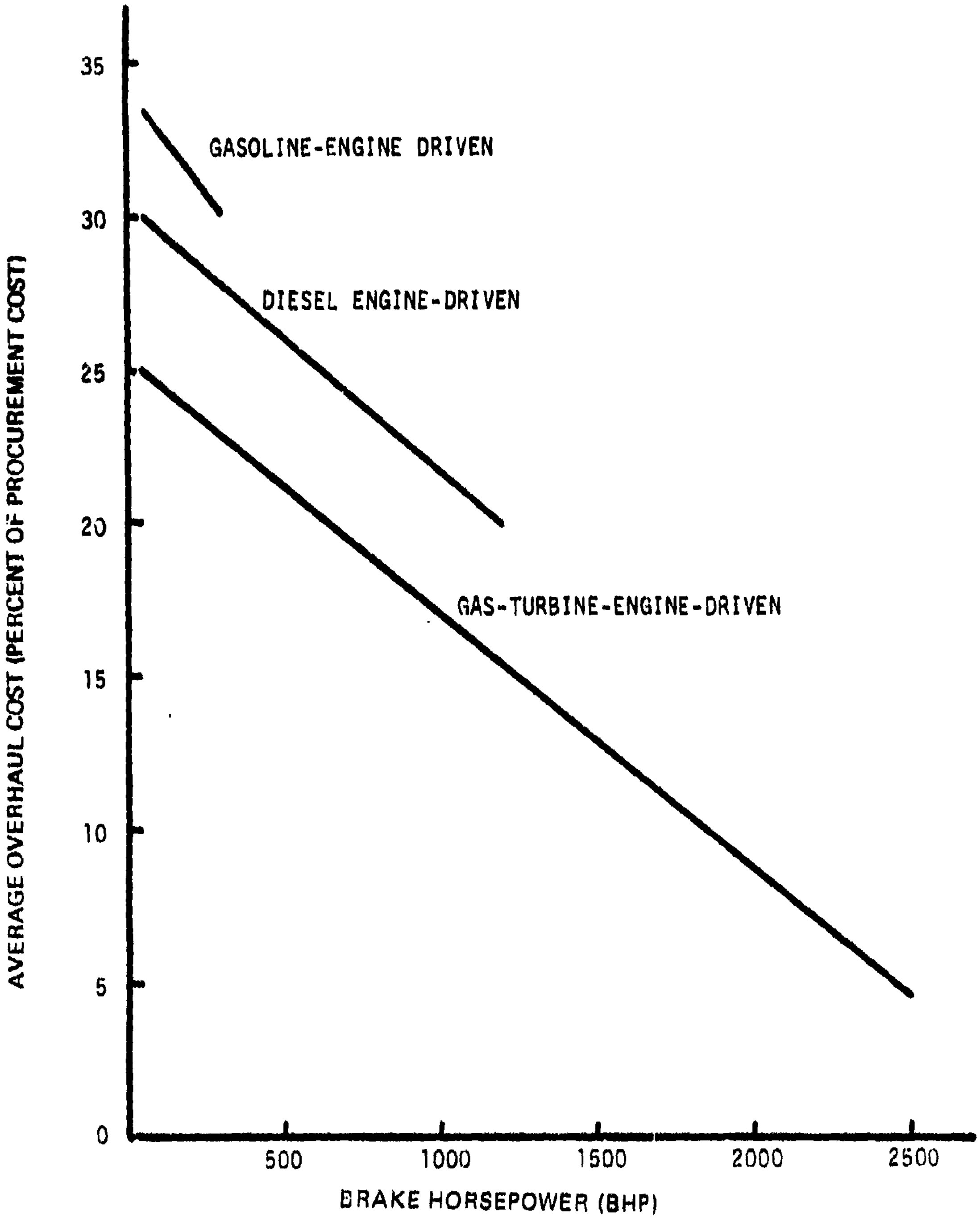



**Figure 19. Average pump unit overhaul cost as percentage of procurement cost.**

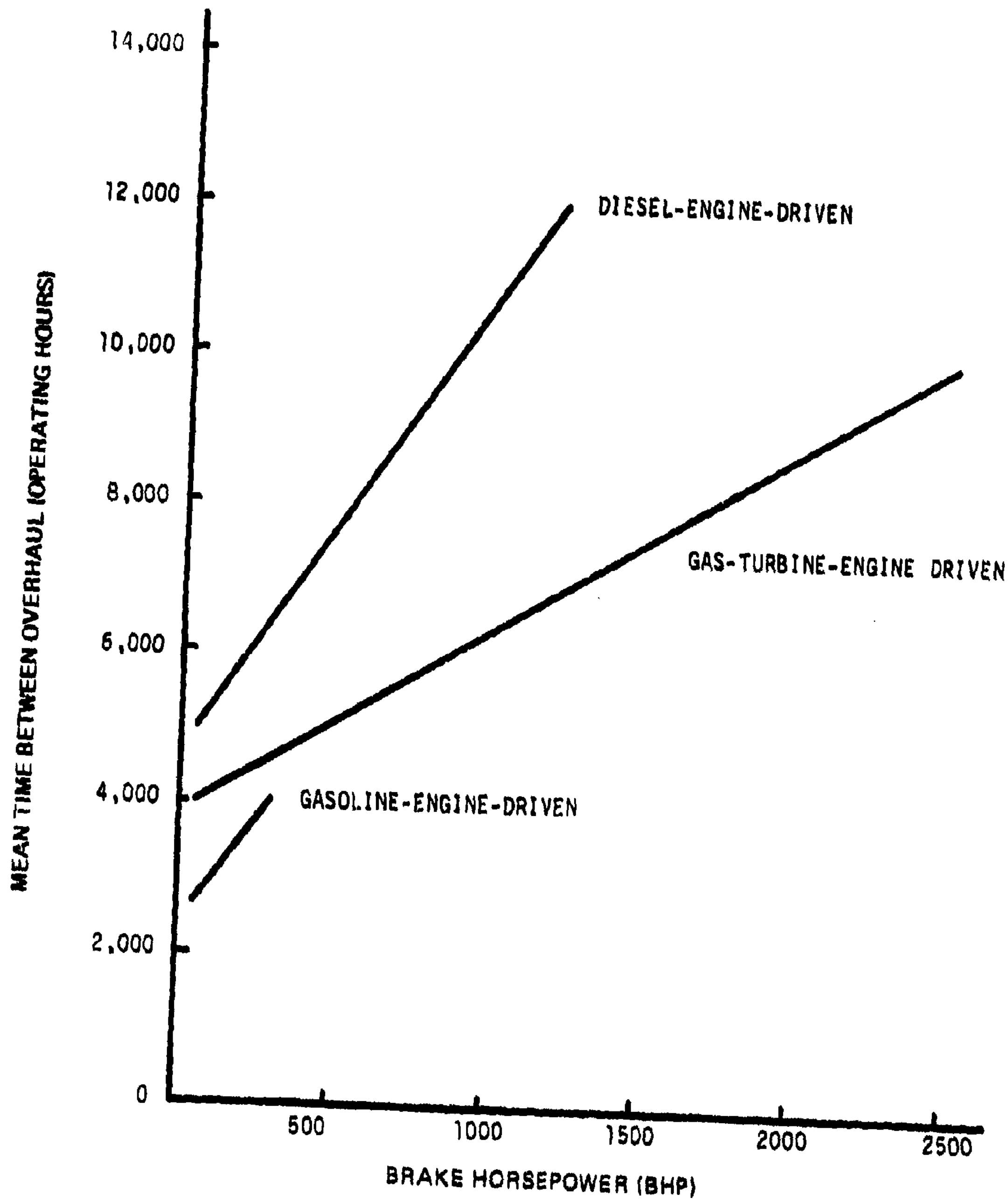



Figure 20. Mean time between overhauls for pump units.

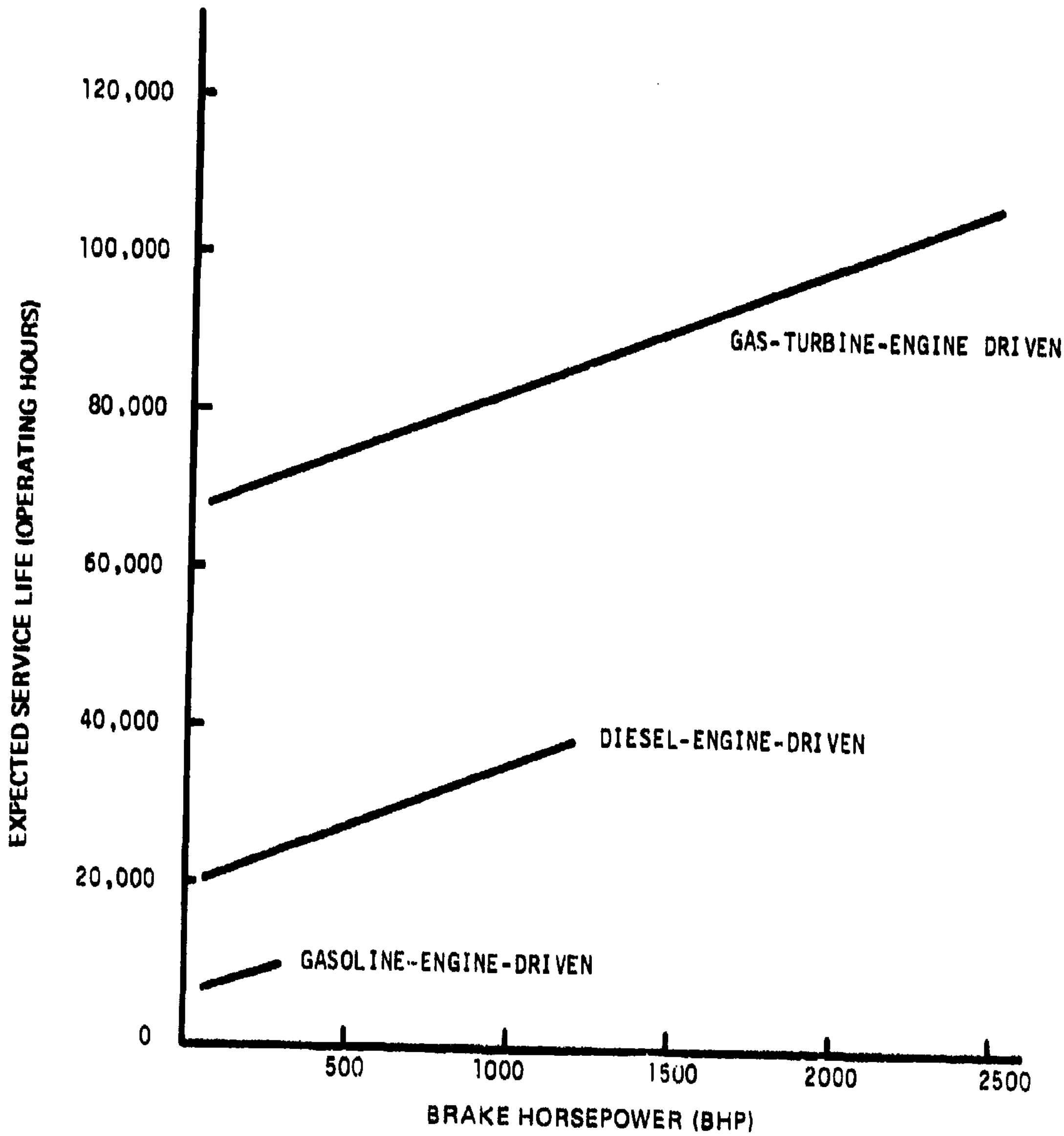



Figure 21. Expected service life of pump units.

The first step in the pipe selection process must be identification of available alternatives. On 7 January 1976, MERADCOM Contract No. DAAG53-76-C-0096 was awarded to Value Engineering Company (VECO), Alexandria, Virginia, to investigate various pipeline concepts considering various materials, joining techniques, and construction procedures. This investigation was to be broad in nature considering innovative pipeline concepts as well as conventional materials and construction techniques. Each pipeline concept was to be evaluated to determine its suitability as an element of a system for military overland transportation of bulk liquid hydrocarbon fuels in a theater-of-operations under wartime conditions.

Defined in the broadest sense, the term "pipeline" may include the pipe, valves, fittings, pumps, storage tanks, and all other facilities required to transport a fluid, under pressure, from one point to another. In a narrow sense, a pipeline may be considered to be only the pipe through which the fluid flows. For the purpose of the investigation conducted by VECO, a "pipeline" was defined as any conduit through which fuel can be pumped regardless of the materials used to form/fabricate the pipeline including metals, plastics, composites, elastomers, and/or combination thereof. VECO was to consider the pipeline (conduit) exclusive of design details for pump stations, storage facilities, and ancillary equipment essential to the operation of an integrated pipeline, except consideration was to be given to the relative contributions of these items to total system cost, personnel required for installation, operation, maintenance, system reliability, etc.

a. **Objectives and Criteria for Pipe Evaluation Program.** The objective of this investigation was to provide some measures of effectiveness and technical feasibility for various candidate pipeline concepts and construction techniques which will:

(1) Maximize the system reliability. System reliability was defined as the probability that a minimum daily throughput requirement can be delivered from a port-of-entry to a bulk distribution breakdown point.

(2) Maximize the rate of pipeline construction, providing the capability to advance a pipe head at a rate sufficient to keep pace with fast-moving combat and combat-support units advancing at rates up to 30 kilometers (18.6 miles) per day.

(3) Minimize the number of personnel, the skill levels, and the amount of equipment required for pipeline construction, operation, and maintenance.

(4) Minimize the total life cycle cost for a complete pipeline system.

(5) Minimize the potential for fuel losses due to natural disasters, hostile action, pilferage, contamination, and administrative handling errors

(6) Minimize repair and maintenance down time.

Evaluation criteria furnished included the following:

(1) The average daily throughput requirement will not be less than 10,000 barrels (420,000 gallons).

(2) The maximum average daily throughput will not exceed 35,000 barrels (1,470,000 gallons).

(3) The average distance from the port-of-entry to the bulk distribution breakdown point will be 100 miles.

(4) Construction, operation, and maintenance of the pipeline shall be possible in climatic categories 1, 2, 5, 6, and 7 as defined in AR 70-38.

(5) The nominal size of each candidate pipeline shall be either 4, 6, or 8 inches. Use of multiple parallel lines to obtain required throughput requirements may be considered as an acceptable concept.

(6) All pipeline system components and each item of required construction equipment shall meet the requirement for water, rail, truck, and air transportation.

The Essential Elements of Analysis were to include, but not necessarily be limited to, the following:

(1) Conducting a thorough survey of Industry to identify as many candidate pipeline concepts as possible.

(2) Identifying, for each candidate pipeline concept, the essential engineering characteristics.

(3) Establishing a measure of cost and operational effectiveness for each feasible pipeline concept.

(4) For each of the feasible candidates, identifying the level of effort in research, development, engineering, and testing required for the pipeline and any ancillary equipment.

(5) Identifying the technological risks associated with each proposed pipeline concept.

(6) Ranking the candidates in order of relative potential, identifying necessary tradeoffs.

The investigation was conducted in two phases. Phase I consisted of four steps intended to reduce the large number of potential concepts down to a few of the most promising ideas which could be anlayzed in detail. The first step consisted of defining the factors and characteristics to be considered and the constraints to be applied in determining the technical feasibility and military suitability of a concept. The second step consisted of establishing the interrelationships between the factors, characteristics, and constraints. During step three, VECO developed a listing of alternative pipeline concepts. Step four of Phase I was the evaluation of all the concepts identified and selection of four concepts offering potential for use in a military bulk fuel distribution system.

Phase II of the investigation involved a more detailed study of four selected concepts.

b. **Interrelationships Between Pipeline Characteristics and Design Criteria.** The following design constraints and pipeline system characteristics were identified by VECO to have a significant affect on the design of military pipelines. Although the listing was not intended to be all-inclusive, it was considered to identify the primary factors to be considered in evaluating alternative pipeline concepts.

Air Transport — The degree of suitability for air transportation via C-130 aircraft.

Bend vs Fittings - The relation with regard to advantage of the use of bent pipe sections as opposed to the use of separate fittings for directional changes in the pipeline.

Climate -- The climatic conditions at the installation location which affect pipeline installation and operation.

Diameter -- Pipe diameter (in inches).

Equipment Required -- The types and quantities of equipment required for installation and construction of the pipeline.

Fluid Temperature -- The average temperature of fuel flowing through the pipeline, determined mainly by the climatic conditions of the pipeline location.

Hostility Duration – The time span of the wartime conditions under which the pipeline must operate.

Inspection/Test – The inspection and testing requirements for all components of the completed pipeline.

Joining Method – The construction techniques and mechanical components required to join pipe section during installation.

Joint Cleanliness – The level of foreign matter present during installation which affects proper joining of pipe sections.

Maintainability – Probability of retaining an item in or restoring an item to operation under a given maintenance policy.

Manhandling – The degree of suitability of pipeline components for repeated physical handling by personnel; the maximum allowable weight of materials per man was assumed to be 30 pounds for repeated lifting.

Material – Pipe material and its properties (i.e., composition, density).

Number of Crews – The total quantity of crew units required to install the pipeline at the specified installation rate.

Number Parallel Lines – The number of parallel pipelines required to maintain a specified rate of flow.

Number of Pump Stations – The total quantity of pumping stations required for the total length to pump fuel at the specified rate through the total length.

Prefab Capability – The possibility of performing some assembly operations prior to stringing the pipe, such as attaching a coupling to one end of each length of pipe, so that only one connection need be made at installation.

Pressure Loss – The overall loss of fluid pressure due primarily to friction as fuel passes through pipeline.

Product Contamination – The degree to which interior surfaces of pipe couplings and fittings affect the quality of the fluid being pumped through the pipeline.

Pump Horsepower - The hydraulic horsepower rating required of the pumps used to propel fuel through the pipeline.

Reliability – Probability that the pipeline will continue in operation for a given period of time.

Reuse Components -- Those pipeline system components which are capable of being reused in new construction.

Right-of-Way Required – The distance (measured in feet) required on either side of the pipeline for equipment and personnel during installation.

Safety – The absence or presence of hazards (to personnel) inherent in a particular construction technique.

Section Length – Average length (in feet) of fabricated pipe sections.

Service Life -- The average expected length of time pipeline components will function before requiring replacement.

Size of Crews -- The number of persons required on each installation (joining) crew to meet the specified installation rate with the method employed.

Skill Level – The level of training and practical experience required of each crew member for proper installation of the pipeline.

Storage Life The maximum period of time materials may be stored under probable storage conditions without deterioration.

Surface vs Buried - The relation with regard to advantage of installed pipeline (below ground) to pipeline installed at ground level.

Terrain The surface features of the installation location which affect pipeline installation and operation.

Time per Joint -- The average elapsed time required by personnel to join two pipe sections during installation and move to the next joint.

Throughput The daily maximum required quantity of fuel to be passed through the pipeline.

Installation Rate - The speed at which pipeline must be installed (miles/day).

Total Length -- The total required length in miles of completed pipeline measured from the port-of-entry to the bulk distribution breakdown point.

Velocity -- The average speed of fuel flow necessary to maintain the required rate of flow through the pipeline.

Vulnerability -- A measurement of the potential for pipeline operation disruption by external forces (i.e., hostile action).

Wall Thickness -- Half the difference between inside and outside pipe diameter dimensions (in inches).

Weight -- The average weight of fabricated pipe sections in pounds per foot of length.

Working Pressure -- Average fluid pressures which fabricated pipe sections must withstand during normal pipeline operation.

Friction Factor -- Hazen-Williams coefficient (usually 140 -- 150).

The interrelationships among these design constraints and pipeline system characteristics were established using the matrix shown in Figure 22. The factors listed on the left side of the matrix were found to be independent; that is, they affect some aspect of the system design but are not affected by the system design.

Listed across the top of the matrix are the dependent factors. These factors all are affected by one or more factors of the system design and, in turn, have some influence on other design considerations.

A dot appears in the matrix at the intersection of each horizontal line and column where the corresponding factors were determined to have a significant interrelationship. For example: The skill level required for installation (tenth column heading) is a function of the equipment required for installation (seventeenth line heading), the pipe-joining method (nineteenth line heading), and the suitability of the joining method for prefabrication of certain assemblies (twenty-fourth line heading). As with the listing of independent and dependent factors, the interactions shown in Figure 22 are not all-inclusive but were selected to provide a reliable tool for comparison of candidate pipeline concepts.

c. **Methodology for Evaluation of Pipeline Concepts.** To use the interrelationships or interactions between the design factors as a tool for comparison of the concepts, a value was assigned to each of the 162 relationships identified in

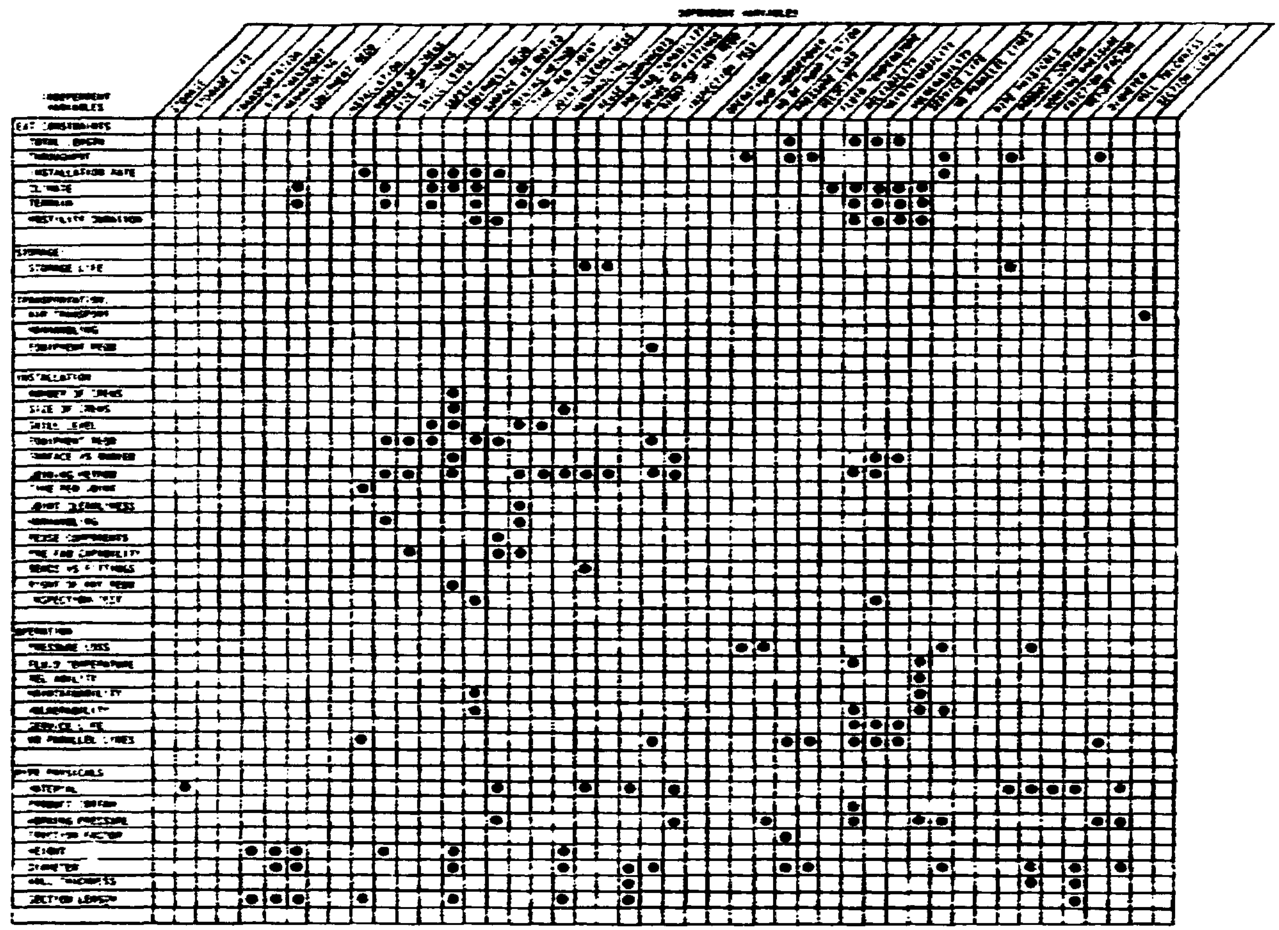


Figure 22. Interrelationships of design constraints and pipeline system characteristics.

Figure 22. The matrix can then be used to compare pairs of concepts on the basis of the interactions.

The values assigned to the interactions were determined as follows:

(1) Each horizontal entry was assigned a value based upon the number of designated interactions in that line. For example, the line labeled "Joining Method" has 12 interactions. The independent variable "Joining Method," therefore, has a value of 12/162 on the basis of the 162 possible interactions.

(2) Each column entry was given a value based upon the number of interactions in that column and the values from step 1 for each of the lines interacting in that column. For example, the column labeled "Size of Crews" has six interactions whose horizontal line values total 47/162. The dependent variable "Size of Crews" then has a value of 6/(47/162).

(3) The value for each individual interaction then was taken as the normalized product (rounded-off) of the line and column values. Using the same example as in steps 1 and 2 above, the product is (12/162) times [6/(47/162)] or 1.532. This value is then normalized based on a value of 2.000, the highest interaction value that appears in the matrix. This value occurs at the interaction of "Service Life" as a function of "Climate." The scoring value for the example is (1.532/2.000) 10 = 7.66 or, rounded-off, 8, as shown in Figure 23 at the interaction of "Size of Crews" as a function of "Joining Method."

After the results were compiled, each interaction value was examined for plausibility. Any anomalies were reconciled through re-examination of the definitions of variables involved.

Comparison of two candidate concepts using the matrix shown in Figure 23 would require a substantial amount of knowledge regarding each concept. Due to the large number of concepts identified and problems encountered in data collection, it was impossible for VECO to acquire this extensive knowledge of each concept. To do so would have required a level of effort in excess of the scope of the contract. Therefore, it was necessary to develop a simplified matrix which, with limited data, would identify the concepts possessing the greatest military potential. For the screening process to be valid, however, it was imperative to consider as many factors as possible.

The abbreviated matrix shown in Figure 24 was developed for this purpose. It requires definition of only four independent design factors (joining method, pipe material, working pressure, and weight), yet those four had a bearing

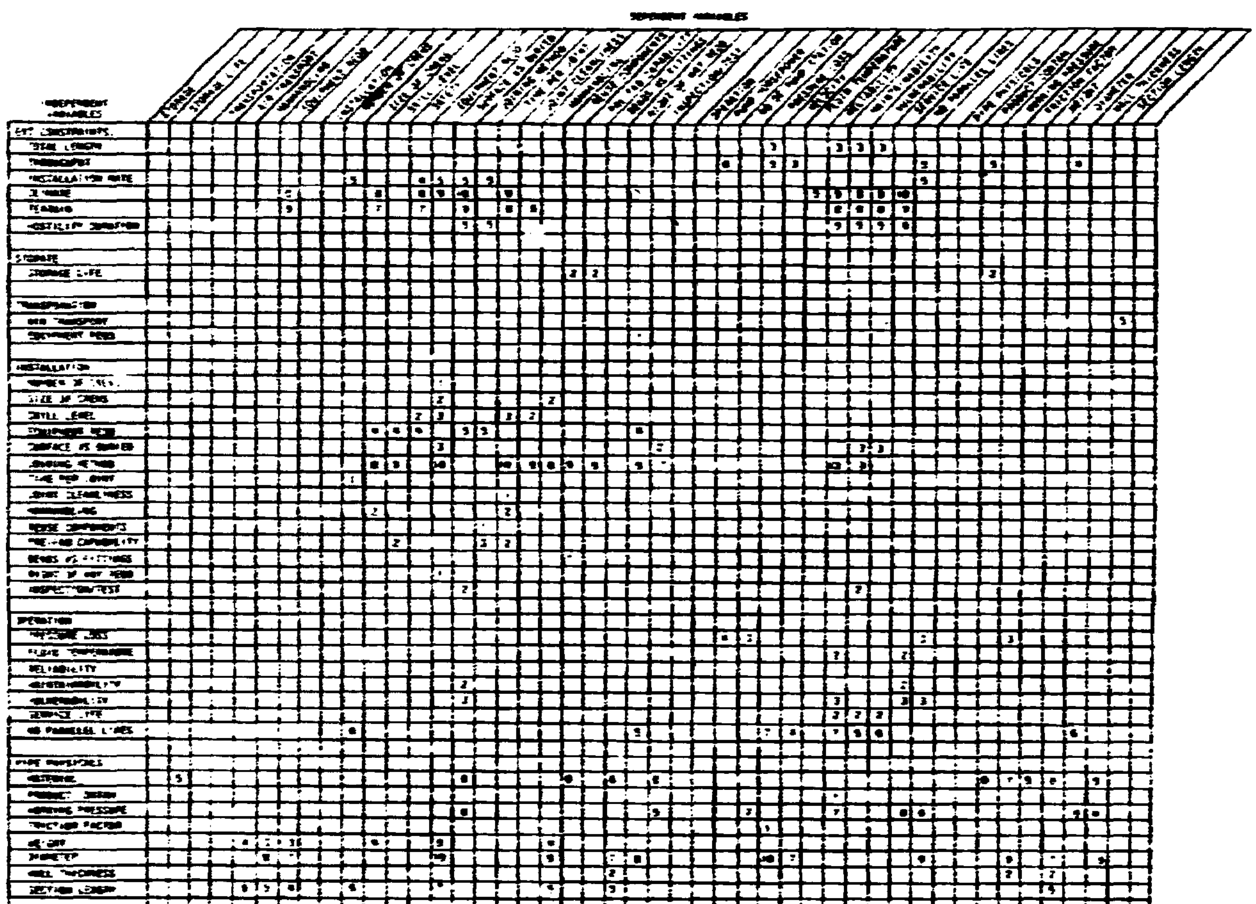


Figure 23. Pipeline scoring matrix.

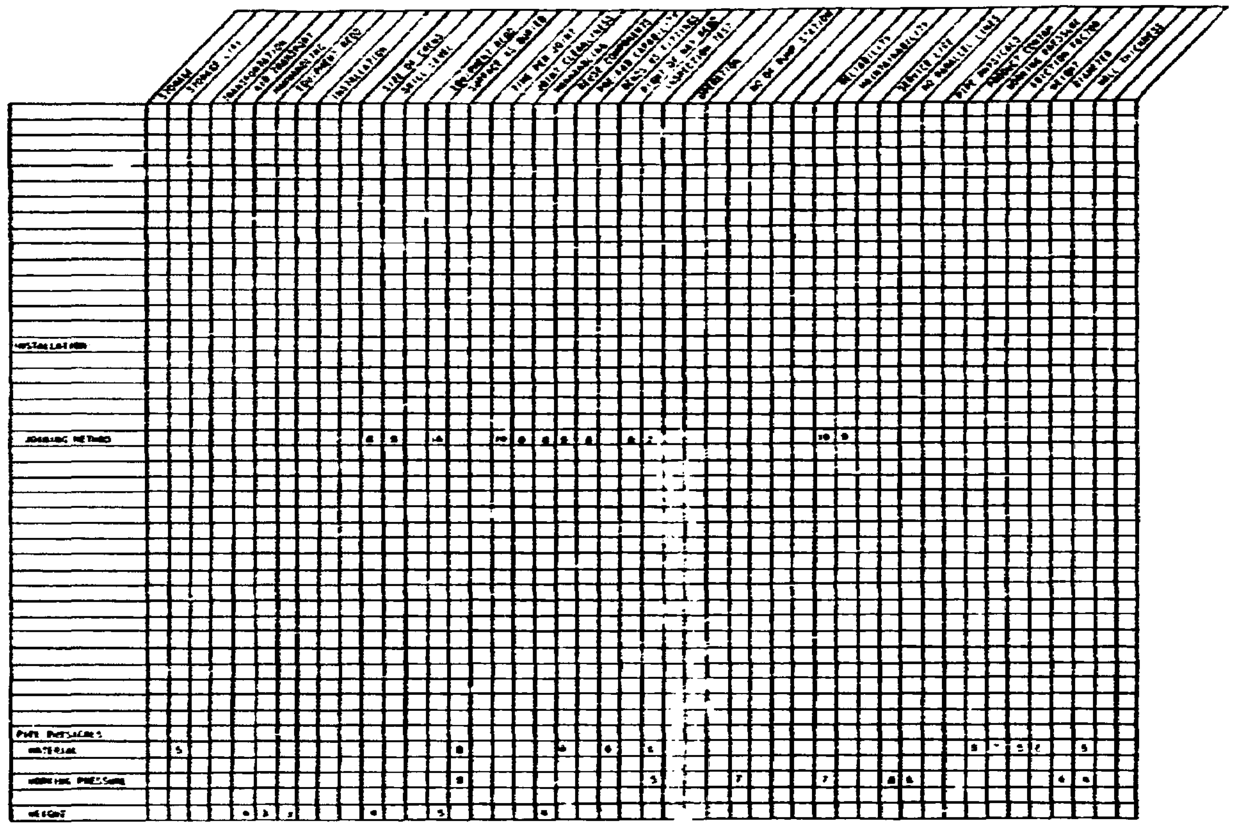


Figure 24. Abbreviated pipeline scoring matrix.

upon 27 of the 36 dependent factors. The values computed for each of the interactions in the full matrix (Figure 23) were retained.

By use of this matrix, the concepts, taken in pairs, were scored by comparison. That is, the attributes of the two concepts were compared in each of the 36 points of consideration. In each instance, the concept having the superior characteristics received the scoring value. In the case of equal qualifications or where sufficient data were not available, both concepts were awarded the value. Thus, the significance of the two scores computed when two concepts are compared is not their magnitudes, but the difference between the scores.

d. **Identification of Pipeline Concepts.** Beginning with the CORG BPFS Study[13-21] as background information, VECO attempted to obtain information on all available pipe materials, joining devices and methods, and high-speed pipeline construction techniques. Using a variety of sources to identify manufacturers, suppliers, and other potential sources of information, VECO sent out 774 solications for data. Replies were received from 264 of the organizations contacted, with 67 of them supplying useful information.

At the outset, an effort was made to contact the 14 companies identified in the CORG BPFS Study to update the findings of that study. These companies are identified in appendix C.

---

[13] R. Stanley LaValee et al; *Bulk Petroleum Facilities and Systems (BPFS) - 1970-1985, Phase I: 1970-1975, Main Report.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1968.

[14] Edward W. King; *Bulk Petroleum Facilities and Systems (BPFS) 1970-1985, Phase I 1970-1975, Annex A, Historical and Doctrinal Review.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

[15] R. Dean George et al; *Bulk Petroleum Facilities and Systems (BPFS) - 1970-1985, Phase I: 1970-1985, Annex B, Part I: Military Equipment Survey,* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

[16] R. Dean George et al; *Bulk Petroleum Facilities and Systems (BPFS) - 1970-1985, Annex B, Part II, Industry Equipment Survey.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

[17] Ray A. Anderson, *Bulk Petroleum Facilities and Systems (BPFS) 1970-1985, Phase I: 1970-1975, Annex C, Pipeline Simulation Model.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

[18] Ray A. Anderson et al; *Bulk Petroleum Facilities and Systems (BPFS) 1970-1985, Phase I: 1970-1975, Annex E, Cost Effectiveness Analysis.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

[19] Gordon B. Page and Richard A. Tarker; *Bulk Petroleum Facilities and Systems (BPFS) - 1970-1985, Phase I: 1970-1975, Annex F, Engineer Organization and Equipment.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

[20] R. Stanley LaValee and Kenneth R. Simmons; *Bulk Petroleum Facilities and Systems (BPFS) 1970-1985, Phase I: 1970-1975, Annex G, Synthesized Engineer Bulk Petroleum Facilities System.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

[21] John M. McCreary et al; *Bulk Petroleum Facilities and Systems (BPFS) - 1970-1985, Phase II: 1975-1985.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

Using data obtained from 34 of the companies contacted, VECO defined 39 pipeline concepts, collectively employing a wide assortment of conduit materials and joining methods. The only pipe materials eliminated from consideration were glass, wood, concrete, and lead. These were judged not suitable for the specified military application.

For the purposes of identification a five-digit alphanumeric code was assigned to each concept. Each digit represented a characteristic or parameter. Figure 25 presents an explanation of the code. For example, the 2 in the identification code 2173D is the concept status (proposed during this study); the 1 indicates the joining method (mechanical coupling); the 7 is the joint geometry (separate fittings); the 3 is the joint description (thermal welding); and the D indicates the conduit material (polypropylene pipe).

On the basis that application of any concept would fall into the near time-frame, VECO considered only those concepts either already commercially available in the form specified or those requiring only adaptation or modification to meet the criteria. Any long-term process development was not deemed feasible; hence, concepts requiring extensive development were not considered.

Listed below are the 39 concept definitions, including five systems currently used by the military: concepts 11112, 12342, 12343, 1234E, and 1240E.

**Concept 11112.** This concept is a pipeline currently used by the military. It employs steel, API 5L pipe, grade A or B, joined by manual welding. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 10.00 lb/ft, 14.97 lb/ft, and 22.34 lb/ft, respectively; corresponding working pressure are 1700 lb/in$^2$, 1200 lb/in$^2$, and 1000 lb/in$^2$, respectively.

**Concept 12342.** This concept is a conventional military pipeline using steel, API 5L pipe, grade A or B, with grooved pipe couplings such as, Victualic style 77 or Gustin-Bacon No. 100 bolted couplings. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 10.00 lb/ft, 14.97 lb/ft, and 22.34 lb/ft, respectively; corresponding working pressures are 1000 lb/in$^2$, 1000 lb/in$^2$, and 800 lb/in$^2$, respectively.

**Concept 12343.** This concept is a conventional military pipeline using lightweight steel tubing with welded-end nipples. The joining method is the same as that used in concept code 12342. Weights of 4-inch-, 6-inch-, and 8-inch-diameter tubing are 3.53 lb/ft, 7.28 lb/ft, and 9.51 lb/ft, respectively; corresponding working pressures are 600 lb/in$^2$, 600 lb/in$^2$, and 500 lb/in$^2$, respectively.

**Concept 1234E.** This is a concept currently used by the military. It uses synthetic rubber hose assemblies conforming to MIL-H-52262, joined by grooved pipe couplings. Weight of 4-inch-diameter hose is 1.65 lb/ft, with a working pressure of 125 lb/in$^2$ (500 lb/in$^2$ burst/225 lb/in$^2$ proof).

**Concept 1240E.** This concept is currently used by the military. It uses synthetic rubber hose assemblies conforming to MIL-H-82127, joined by cam and grooved couplings. Weights of 4-inch- and 6-inch-diameter hoses are 1.25 lb/ft and 2.3 lb/ft, respectively; corresponding working pressure is 100 lb/in$^2$ for diameters (400 lb/in$^2$ burst/200 lb/in$^2$ proof).

**Concept 21111.** This concept proposes a pipeline using aluminum, schedule 40, 6061-T6 pipe, joined by manual welding. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 3.73 lb/ft, 6.56 lb/ft, and 9.88 lb/ft, respectively; corresponding working pressures are 1000 lb/in$^2$, 800 lb/in$^2$, and 650 lb/in$^2$, respectively.

**Concept 21122.** This concept represents a proposed pipeline using steel, API 5L pipe, grade A or B, joined by automatic welding equipment, such as that available from Dimetrics, Astro-Arc, or Sciaky Bros. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 10.00 lb/ft, 14.97 lb/ft, and 22.34 lb/ft, respectively; corresponding working pressures are 1700 lb/in$^2$, 1200 lb/in$^2$, and 1000 lb/in$^2$, respectively.

**Concept 2123C.** This concept proposes using high-density polyethylene (HDPE) pipe, joined by thermal welding, such as Ryerson "Monoline" and M.L. Sheldon "Sclairpipe." Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipe are 2.77 lb/ft, 5.99 lb/ft, and 9.35 lb/ft, respectively; corresponding working pressure is 160 lb/in$^2$ for each diameter.

**Concept 2173D.** This concept uses schedule 40 polypropylene pipe, joined by thermally welded, separate fittings (R & G Sloane "Fuseal"). Weight of 4-inch- and 6-inch-diameter pipes are 1.87 lb/ft and 3.56 lb/ft, respectively; corresponding working pressures are 125 lb/in$^2$ and 100 lb/in$^2$, respectively.

**Concept 220DB.** This proposed pipeline concept uses epoxy resin fiberglass-reinforced plastic pipe, joined by CIBA-GEIGY "Pronto-Lock" and "Pronto-Lock II" male/female integral threaded couplings. Weights for 4-inch-, 6-inch-, and 8-inch-diameter pipes are 0.8 lb/ft, 1.7 lb/ft, and 3.3 lb/ft, respectively; corresponding working pressures are 300 lb/in$^2$, 200 lb/in$^2$, and 150 lb/in$^2$, respectively.

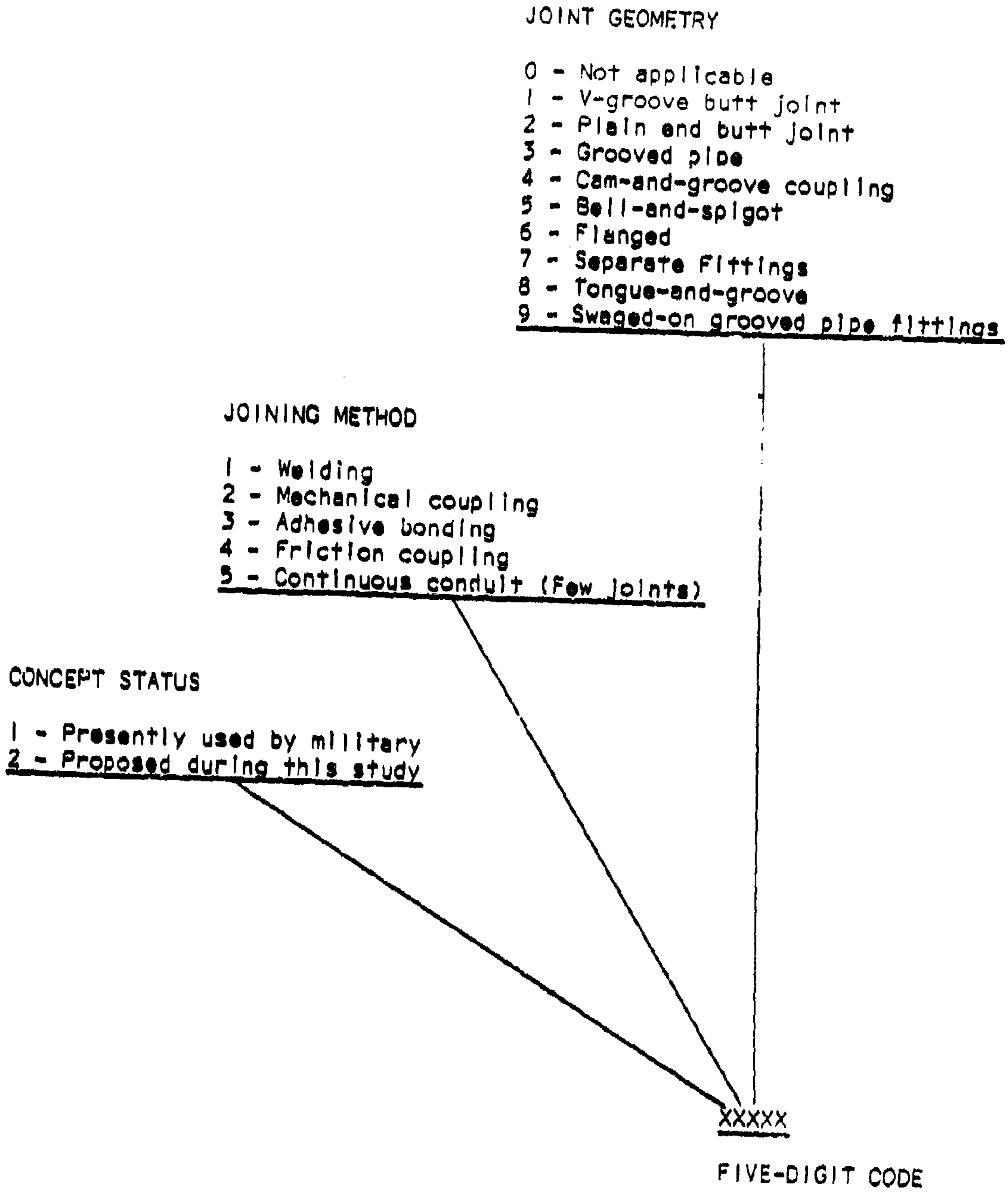



Figure 25. Concept identification codes.

JOINT DESCRIPTION

0 - Not applicable
1 - Manual welding
2 - Automatic welding
3 - Thermal welding
4 - Bolted coupling
5 - Wedge locking coupling
6 - Latching coupling
7 - Bolted gripping coupling
8 - Rubber seal or "O" ring
9 - Flange clamp and "O" ring
A - Locking strip
B - Butt-and-strap hand lay-up
C - Threaded
D - Male/Female threaded integral coupling
E - Swaging
F - Latching lugs

CONDUIT MATERIAL

1 - Aluminum, schedule 40 pipe, 6061-T6 or 6063-T63
2 - Steel, API 5L pipe, grade A or B
3 - Steel, lightweight tubing
4 - Steel, schedule 40 pipe
5 - Steel, high-strength well casing
6 - Steel, spiral welded pipe
7 - Cast iron pipe
8 - Ductile iron pipe
9 - Polyvinyl chloride (PVC) pipe
A - Polyester resin fiberglass reinforced plastic (FRP) pipe
B - Epoxy resin fiberglass reinforced plastic pipe
C - High density polyethylene (HDPE) pipe
D - Polypropylene pipe
E - Synthetic rubber hose

XXXXX

FIVE-DIGIT CODE

Figure 25. Concept identification codes. (Continued)

**Concept 22272.** This concept is a proposed pipeline using steel, API 5L pipe, grade A or B, joined by Gustin-Bacon No. 200 bolted gripping couplings. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 10.00 lb/ft, 14.97 lb/ft, and 22.34 lb/ft, respectively; corresponding working pressures are 1000 lb/in$^2$, 600 lb/in$^2$, and 500 lb/in$^2$, respectively.

**Concept 22273.** This concept proposes a pipeline using lightweight steel tubing, joined by the same mechanical coupling as that used in Concept 22272. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 3.53 lb/ft, 7.28 lb/ft, and 9.51 lb/ft, respectively; corresponding working pressures are 600 lb/in$^2$, 600 lb/in$^2$, and 500 lb/in$^2$, respectively.

**Concept 22341.** This concept proposes using aluminum, schedule 40, 6061-T6 pipe. Sections are joined by grooved couplings, such as Gustin-Bacon No. 101 bolted coupling. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 3.73 lb/ft, 6.56 lb/ft, and 9.88 lb/ft, respectively; corresponding working pressures are 1000 lb/in$^2$, 1000 lb/in$^2$, and 800 lb/in$^2$, respectively.

**Concept 22356.** This concept uses spiral-welded steel pipe, joined by Naylor "Wedgelock" wedge locking grooved-pipe couplings. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 3.96 lb/ft, 7.94 lb/ft, and 13.20 lb/ft, respectively; corresponding working pressure is 400 lb/in$^2$ for each diameter.

**Concept 22363.** This proposed pipeline concept consists of lightweight steel tubing with welded-end nipples. Sections are joined by latching grooved pipe couplings, such as Victaulic style 78 or Guston-Bacon No. 115. Weights of 4-inch-, 6-inch-, and 8-inch-diameter tubing are 3.53 lb/ft, 7.28 lb/ft, and 9.51 lb/ft, respectively; corresponding working pressure is 300 lb/in$^2$ for each diameter.

**Concept 22401.** This concept proposes using aluminum, schedule 40, 6061-T6 pipe, joined by cam-and-groove-type couplings, such as Andrews 400A, 400D, 600A, 600D, 800A, 800D, or OPW 633-A, 633-D with NPT female threads (aluminum). Weights of 4-inch-, 6-inch- and 8-inch-diameter pipes are 3.73 lb/ft, 6.56 lb/ft, and 9.88 lb/ft, respectively; corresponding working pressures are 100 lb/in$^2$, 75 lb/in$^2$, and 50 lb/in$^2$, respectively.

**Concept 22404.** This concept proposes using steel, schedule 40 pipe, joined by cam-groove-type couplings, such as Andrews 400A, 400D, 600A, 600D, 800A, 800D, or OPW 633-A, 633-D with NPT female threads (steel). Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 10.79 lb/ft and 28.55 lb/ft, respectively; corresponding working pressure is 100 lb/in$^2$ for the 4- and 6-inch diameters.

**Concept 224AB.** This is a proposed pipeline concept using filament-wound epoxy resin fiberglass-reinforced plastic pipe, joined by bell-and-spigot coupling with locking key strip, such as those available from Brunswick and Fiberglass Resources.

**Concept 225F1.** This proposed pipeline uses aluminum 6063-T63 pipe, joined by Race and Race "Racebilt" bell-and-spigot coupling with an "0" ring seal and latching lugs. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 1.35 lb/ft, 3.06 lb/ft, and 4.64 lb/ft, respectively; corresponding working pressure is 350 lb/in$^2$ for each diameter.

**Concept 2269A.** This proposed pipeline concept uses filament-wound polyester resin fiberglass-reinforced plastic (FRP) pipe, joined by Beetle "Quick-Lock" flange clamp with "0" ring. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 1.5 lb/ft, 2.7 lb/ft, and 4.1 lb/ft, respectively; corresponding working pressures are 200 lb/in$^2$, 200 lb/in$^2$ and 150 lb/in$^2$, respectively.

**Concept 227AB.** This is a proposed pipeline concept using epoxy resin fiberglass-reinforced plastic pipe, joined by Fiberglass Resources' "Kwik-Key" coupling with "0" ring and locking strip. Weights of 4-inch, 6-inch-, and 8-inch-diameter pipes are 0.8 lb/ft, 1.6 lb/ft, and 2.7 lb/ft, respectively; corresponding working pressures are 350 lb/in$^2$, 250 lb/in$^2$, and 260 lb/in$^2$, respectively.

**Concept 22705.** This concept proposes using high-strength well casing steel pipe, joined by Armco "Seal Lock" threaded well casing couplings. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 11.60 lb/ft, 23.00 lb/ft, and 32.00 lb/ft, respectively; corresponding working pressures are 2100 lb/in$^2$, 1700 lb/in$^2$, and 1500 lb/in$^2$, respectively.

**Concept 228A1.** This concept proposes a pipeline using aluminum, schedule 40 pipe, joined by Sandia Labs' male/female tongue-and-groove coupling with locking strips ("Taped Joint"). Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 3.73 lb/ft, 6.56 lb/ft, and 9.88 lb/ft, respectively; corresponding working pressures are 1700 lb/in$^2$, 1200 lb/in$^2$, and 1000 lb/in$^2$, respectively.

**Concept 228A2.** This concept uses steel, API 5L pipe, grade A or B, joined by the same coupling as that used in Concept 228A1. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 10.00 lb/ft, 14.97 lb/ft, and 22.34 lb/ft, respectively; corresponding working pressures are 1700 lb/in$^2$, 1200 lb/in$^2$, and 1000 lb/in$^2$, respectively.

**Concept 2294B.** This concept proposes using epoxy resin fiberglass-reinforced plastic pipe, joined by "Gamagrip" swaged-on grooved pipe couplings. Weights of 4-inch- and 6-inch-diameter pipes are 0.8 lb/ft and 1.7 lb/ft, respectively; corresponding working pressures are 225 $lb/in^2$ and 250 $lb/in^2$, respectively.

**Concept 232BA.** This concept uses filament-wound polyester resin fiberglass-reinforced plastic (FRP) pipe, joined by butt-and-strap hand lay-up of resin and mat, such as that available from Century Fiberglass. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 1.5 lb/ft, 2.7 lb/ft, and 4.1 lb/ft, respectively; corresponding working pressure is 150 $lb/in^2$ for each diameter.

**Concept 23509.** This concept proposes a pipeline using polyvinyl chloride (PVC) pipe, joined by cemented (adhesive-bonded) bell-and-spigot couplings, such as those available from Certain-Teed. Weights of 4-inch- and 6-inch-diameter pipes are 1.822 lb/ft and 3.947 lb/ft, respectively; corresponding working pressure is 200 $lb/in^2$ for both diameters.

**Concept 2350B.** This concept is a pipeline employing epoxy resin fiberglass-reinforced plastic pipe, joined by cemented (adhesive-bonded) bell-and-spigot couplings, such as those available from Fiberglass Resources, Fiber Cast, and Koch. Weights for 4-inch-, 6-inch-, and 8-inch-diameter pipes are 0.8 lb/ft, 1.6 lb/ft, and 2.7 lb/ft, respectively; corresponding working pressures are 350 $lb/in^2$, 250 $lb/in^2$, and 260 $lb/in^2$, respectively.

**Concept 23709.** This concept proposes using polyvinyl chloride (PVC) pipe, joined by cemented (adhesive-bonded) fittings, such as those available from Certain-Teed and Dixie Plastics. Weight of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 1.822 lb/ft, 3.947 lb/ft, and 6.679 lb/ft, respectively; corresponding working pressure is 200 $lb/in^2$ for each diameter.

**Concept 2370B.** This concept proposes a pipeline using epoxy resin fiberglass-reinforced plastic (FRP) pipe, joined by Conley FRP cemented (adhesive-bonded) fittings. Weights for 4-inch-, 6-inch-, and 8-inch-diameter pipes are 0.8 lb/ft, 1.6 lb/ft, and 2.7 lb/ft, respectively; corresponding working pressure is 150 $lb/in^2$ for each diameter.

**Concept 240E1.** This concept uses aluminum schedule 40, 6061-T6 pipe, joined by "ZAP-LOK" swaged bell-and-spigot friction coupling. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 3.73 lb/ft, 6.56 lb/ft, and 9.88 lb/ft, respectively; corresponding working pressures are 1700 $lb/in^2$, 1200 $lb/in^2$, and 1000 $lb/in^2$, respectively.

**Concept 240E2.** This concept proposes using steel, API 5L pipe, grade A or B, joined by the same method as that used in Concept 240E1. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 10.00 lb/ft, 14.97 lb/ft, and 22.34 lb/ft, respectively; corresponding working pressures are 1700 lb/in$^2$, 1200 lb/in$^2$, and 1000 lb/in$^2$, respectively.

**Concept 24587.** This is a pipeline concept using cast iron pipe, joined by a bell-and-spigot-type friction joining mechanism with an "0" ring seal, such as McWave "Tyton" and American "Fastite." Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 15 lb/ft, 23.9 lb/ft, and 34.7 lb/ft, respectively; corresponding working pressure is 350 lb/in$^2$ for each diameter.

**Concept 24588.** This concept proposes using ductile iron pipe, joined by a bell-and-spigot-type friction joining mechanism with an "0" ring seal, such as McWave "Tyton" and American "Fastite." Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 13.4 lb/ft, 21 lb/ft, and 29.7 lb/ft, respectively; corresponding working pressure is 350 lb/in$^2$ for each diameter.

**Concept 24589.** This is a proposed concept using polyvinyl chloride (PVC) pipe, joined by bell-and-spigot coupling with a rubber seal, such as ASC Plastics' "Vulcan" with integral coupler; Certain-Teed "Fluid-Tite;" Clow "Bell-Tite;" Ethy "Bell-Ring;" Johns-Manville "Ring-Tite;" Rehau "Mechan-O-Joint." Weights for 4-inch-, 6-inch-, and 8-inch-diameter pipes are 1.86 lb/ft, 4.05 lb/ft, and 6.91 lb/ft, respectively; corresponding working pressure is 200 lb/in$^2$ for each diameter.

**Concept 2458C.** This concept proposes a pipeline using high-density polyethylene (HDPE) duct, with a bell-and-spigot-type friction joining mechanism with an "0" ring seal, such as Phillips Product "Driscon 3700." Weights for 4-inch and 6-inch-diameter pipes are 0.96 lb/ft and 1.82 lb/ft, respectively; corresponding working pressure is 75 lb/in$^2$ for both diameters.

**Concept 24789.** This concept presents a proposed pipeline using polyvinyl chloride (PVC) pipe, joined by Tridyn "Wedge-Tite" friction coupling with rubber seal. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 1.48 lb/ft, 3.22 lb/ft, and 5.44 lb/ft, respectively; corresponding pressure is 200 lb/in$^2$ for each diameter.

**Concept 247E1.** This concept proposed a pipeline using aluminum schedule 40, 6061-T6 pipe, joined by McDonnel "Duraswage" swaged-friction couplings. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 3.73 lb/ft, 6.56 lb/ft, and 9.88 lb/ft, respectively; corresponding working pressures are 1700 lb/in$^2$, 1200 lb/in$^2$, and 1000 lb/in$^2$, respectively.

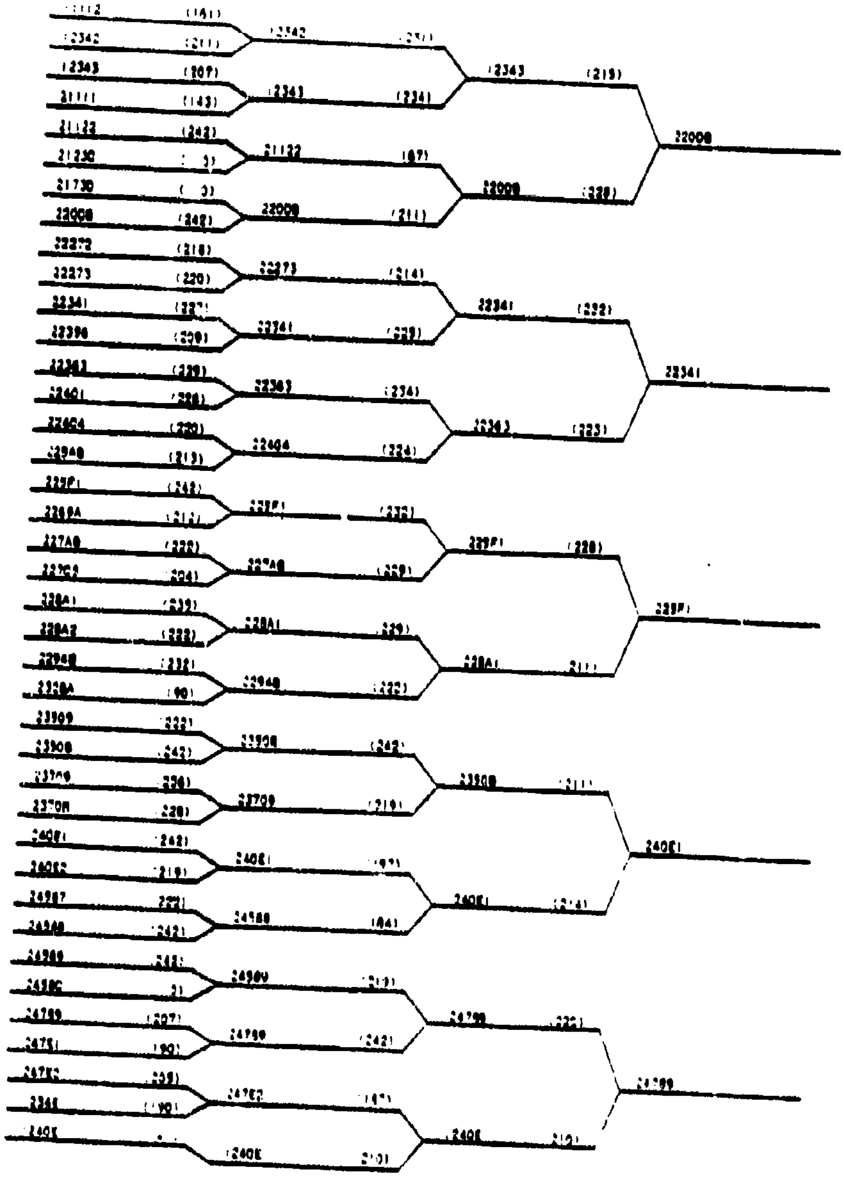


Figure 26. Elimination scoring results.

**Concept 247E2.** This proposed pipeline concept uses steel, API 5L pipe, grade A or B, joined by the same method as that used in Concept 247E1. Weights of 4-inch-, 6-inch-, and 8-inch-diameter pipes are 10.00 lb/ft, 14.97 lb/ft, and 22.34 lb/ft, respectively; corresponding working pressures are 1700 lb/in$^2$, 1200 lb/in$^2$, and 1000 lb/in$^2$, respectively.

e. **Comparison of Proposed Concepts.** The 39 proposed pipeline concepts were paired for comparison as shown in Figure 26. Relative scores for the concepts in each pair were computed using the abbreviated scoring matrix (Figure 24). These scores are shown, inclosed in parentheses in Figure 26. The concept from each pairing receiving the lowest score was eliminated from further consideration. Using sequential pairings of the higher scoring concepts, 34 of the 39 concepts were eliminated from further consideration.

When the scoring matrix technique for comparison of alternatives is used, any alternative found to have an unacceptable characteristic is assigned a value of zero. Three concepts (2123C, 217ED, and 2458C) received scores of zero because the materials, high-density polyethylene and polypropylene, are not compatible with the applicable petroleum products throughout the specified environmental temperature range.

A MERADCOM program review found that the pairing procedure used by VECO will not necessarily select the five best concepts. In Figure 26, concepts 11112, 12342, 12343, 21111, 21122, 2123C, 2173D, and 230DB are compared to each other through the pairing process. It is valid to conclude that concept 220DB is the preferred concept from this group of eight. However, concept 220DB has not been compared, in any way, to the 31 other concepts listed below concept 220DB along the left side of Figure 26. Thus, it is possible that any, or all, of these 31 concepts could be superior to concept 220DB.

Similarly, concept 22341 is superior to concepts 22272, 22273, 22356, 22363, 22401, 22404, and 225AB. However, the relative value of concept 22272 in comparison to the 31 other concepts is not known. Therefore, it is not valid to conclude that concept 22341 is necessarily one of the five best concepts.

Following this rationale to its conclusion, concepts 220DB, 22341, 225F1, 240E1, and 24789 have not been identified positively as the five best alternatives. Actual determination of the five best concepts using paired comparisons would require a large number of comparisons based on a complex decision tree. As an alternative, VECO compared the five proposed concepts to five concepts currently in use by the Military. The results of these comparisons are shown in Figure 27. In every case, when the abbreviated scoring matrix was used, the proposed concepts all scored

| PROPOSED CONCEPTS | PRESENT MILITARY SYSTEMS: 11112 | 12342 | 12343 | 1234E | 1240E |
|---|---|---|---|---|---|
| 220DB (117) | 172 / 222 | 215 / 228 | 215 / 228 | 207 / 223 | 207 / 232 |
| 22341 (117) | 180 / 228 | 227 / 242 | 223 / 242 | 195 / 222 | 215 / 223 |
| 225F1 (130) | 180 / 220 | 204 / 220 | 208 / 228 | 205 / 232 | 215 / 242 |
| 240E1 (135) | 199 / 242 | 173 / 205 | 180 / 205 | 183 / 208 | 200 / 210 |
| 24789 (113) | 183 / 222 | 199 / 222 | 207 / 222 | 194 / 220 | 210 / 220 |

Figure 27. Present systems compared to proposed concepts.

higher than the existing Military systems. The number in parentheses below each proposed concept identification code is the sum of the differences between the concept scores and the respective scores for the five present Military systems.

Further evaluation of concept 24789, PVC pipe joined by Tridyn "Wedge-Tite" friction coupling, could have some seepage at the joints. In the usual application (waterlines) for that type of pipe, some seepage at the joints is allowable. Eliminating the potential for the seepage would require changes in tolerances, manufacturing methods, and/or the geometry of the proprietary seal. Due to this problem and because concept 24789 had the lowest total sum of the differences when compared to the five existing Military systems, the concept was eliminated from further consideration.

f. **Summary of Value Engineering Company Findings.** Given the objectives and criteria specified in the contract (outlined herein in paragraph 8), construction of 100 miles of 8-inch-diameter pipeline was selected as the basis for comparison of the four concepts. For the purposes of this investigation, VECO considered the ability to deliver the maximum anticipated throughput to be the most demanding criterion.

For each concept, a method of construction and sequence of operation considered to be the most efficient and cost-effective means of construction were established. In each case, the construction capability of a single crew was less than the desired 30 kilometers per day. Thus, the evaluation considered the use of multiple crews to achieve the desired rate of construction.

**(1) Concept 220DB – CIBA-GEIGY PRONTO-LOCK Pipe System.** The CIBA-GEIGY fiberglass-reinforced epoxy resin pipe is available in diameters from 2 inches through 16 inches. Designed for continuous operation at a maximum working pressure of 150 $lb/in^2$, the product line includes pipe, fittings, and adapters. Using a bell-and-spigot design, the PRONTO-LOCK mechanical joining system provides a quick, simple method for joining pipe and fittings.

The bell-shaped, female-end fitting or PRONTO-LOCK box end, shown in Figure 28, is threaded internally and contains an O-ring seal in a groove below the threads. The male end of the pipe or PRONTO-LOCK pin is threaded above the smooth tapered end which is stabbed into the box, compressing the O-ring to establish a seal. For pipe diameters from 2 inches through 6 inches, the standard PRONTO-LOCK pin has the tapered sealing surface and thread fabricated as an integral part of the pipe. The joint is tightened by rotating the pipe with a strap wrench.

Because of problems associated with alignment and rotating pipe of diameters from 8 inches through 16 inches, the male threads are on a concentric sleeve which can be rotated to tighten the joint without rotating the pipe. This concentric sleeve seats against a shoulder at the back of the tapered sealing surface. This design feature, designated PRONTO-LOCK II, permits 2 degrees of angular deflection in the joint.

Standard nominal joint length for the CIBA-GEIGY pipe is 40 feet. An 8-inch-diameter, 40-foot section of pipe, with end fittings, weighs approximately 145 pounds. Thus, four men can handle one section of pipe without any special handling equipment. VECO proposes the pipe to be hauled to the construction site using 5-ton truck tractors towing flatbed semitrailers with telescoping bodies. Each joint of pipe would be joined to the end of the pipeline as it is off-loaded from the delivery trucks.

Installation of a joint of pipe begins with two men positioning cribbing to support the pipe during installation. Simultaneously, four men off-load a section of pipe from the delivery truck, carry it to the end of the pipeline, and place the pipe on the cribbing. After removing the end protectors, inspecting the pipe ends for dirt and damage, and lubricating the pin, the four men lift the section of pipe, stab the pin into the box end of the previous section, and run the thread up. Another crew member, using a strap or spanner wrench tightens the joint. The crew then lifts the pipe allowing the cribbing to be removed and advanced to the next joint of pipe.

Concept Code 2200B

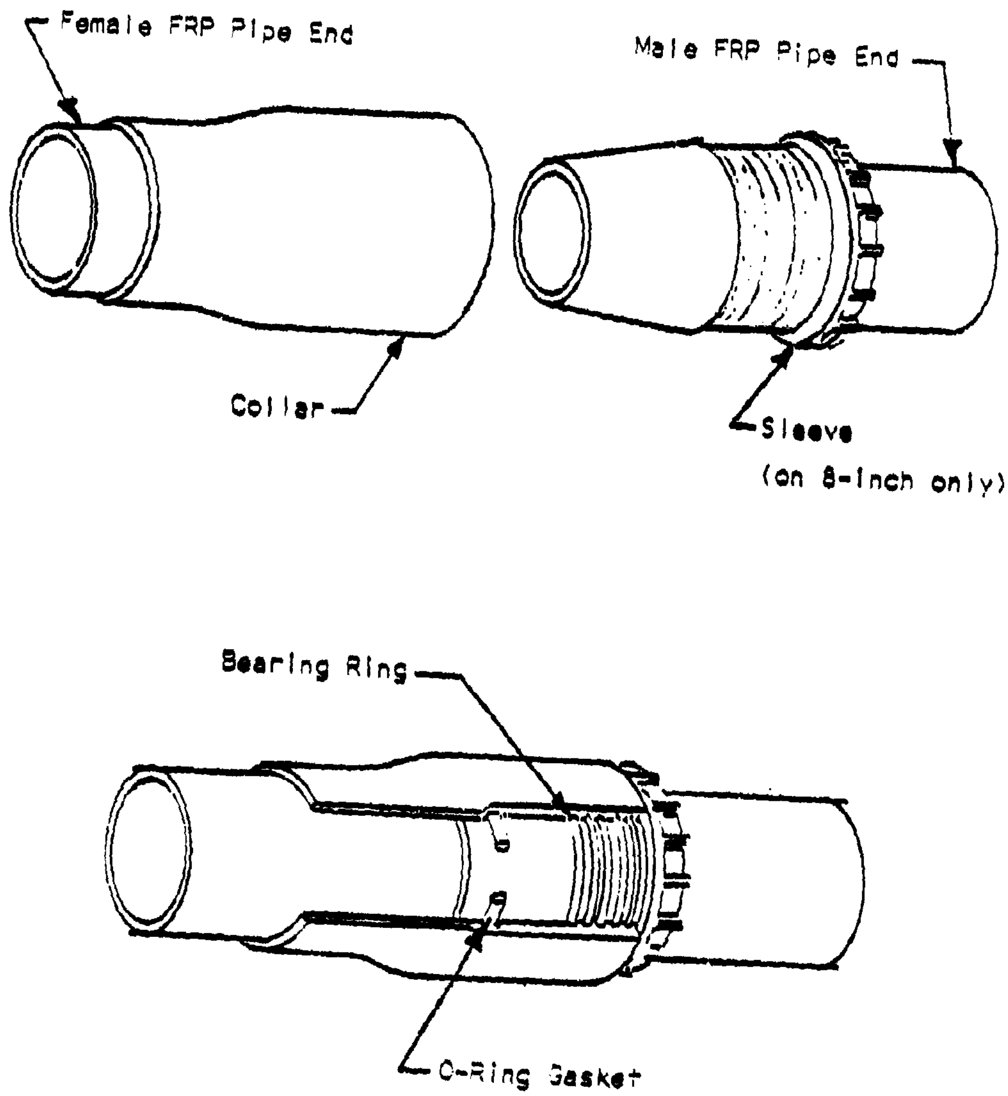



**Figure 28. Ciba-Geigy PRONTO-LOCK Joint.**

Two men are stationed on the delivery truck to assist in off-loading the pipe. Thus, the proposed installation procedure requires at least 9 men, 2 on the truck off-loading the pipe, 2 to carry and position cribbing, 4 to carry and install the pipe, and 1 to tighten the joint. The estimated rate of construction for this crew is one joint every 84 seconds. This equates to a construction rate of 0.32 mile per hour. Assuming a crew works a 10-hour shift, the maximum length of line installed in one day by one crew would be 3.2 miles. Thus, to obtain the desired construction rate of 18.6 miles per day, at least six crews would be required.

Based on further analysis, it was concluded that 8 crews, working 4 crews per shift and two 10-hour shifts per day, can best accomplish the construction of 30 kilometers (18.6 miles) per day. This approach will allow adequate time for the crews to install valves and fitting, make grade crossings, etc.

To support the installation crews, a continuous supply operation is required. It was assumed that each truck can haul 36 lengths of 8-inch-diameter pipe maintaining an average traveling speed of 30 miles per hour and that all the pipe is prepositioned at one end of the 100-mile pipeline. On this basis, it was determined that thirty-eight 5-ton truck tractors with telescoping flatbed semitrailers would be required to maintain a continuous supply of pipe.

Additional equipment requirements include seven 2-½-ton cargo trucks and two ¼-ton utility trucks. Two of the seven 2-½-ton cargo trucks would be outfitted as pipeline construction trucks with winches and A-frames for installing valves and other heavy components.

To deliver 35,000 barrels per day through one 8-inch-diameter CIBA-GEIGY pipeline would require approximately 19 pump stations, assuming no change in elevation along the 100-mile length of the pipeline. Each pump station would operate at a maximum discharge pressure of 150 lb/in$^2$ delivering approximately 100 hydraulic horsepower.

**(2) Concept 22341 – Grooved-End, Mechanically Coupled, Aluminum Pipe System.** Mechanical couplings for joining grooved-end pipe are manufactured by Gustin-Bacon Division, Aeroquip Corporation, Lawrence, Kansas, and Victualic Company of America, Elizabeth, New Jersey. This concept employs the same basic design as the Military standard coupled steel pipelines except it is proposed to use schedule 40, 6061-T6 aluminum pipe and aluminum couplings.

A segmented coupling engages circumferential grooves around the end of the pipe as shown in Figure 29 to provide a positive mechanically locked joint. An elastomeric gasket encased by the coupling seals the joint. When used with appropriate couplings, 8-inch-diameter, grooved-end, schedule 40 aluminum pipe is suitable for operating at pressures up to 800 lb/in$^2$. An 8-inch grooved coupling will allow 1 degree, 41 minutes deflection in the joint.

A 20-foot length of schedule 40, 8-inch-diameter, aluminum pipe weighs approximately 198 pounds. Use of longer lengths would be desirable to reduce the number of joints. However, the weight of longer sections would preclude manhandling the pipe during the stringing-and-laying operations.

Concept Code 22341

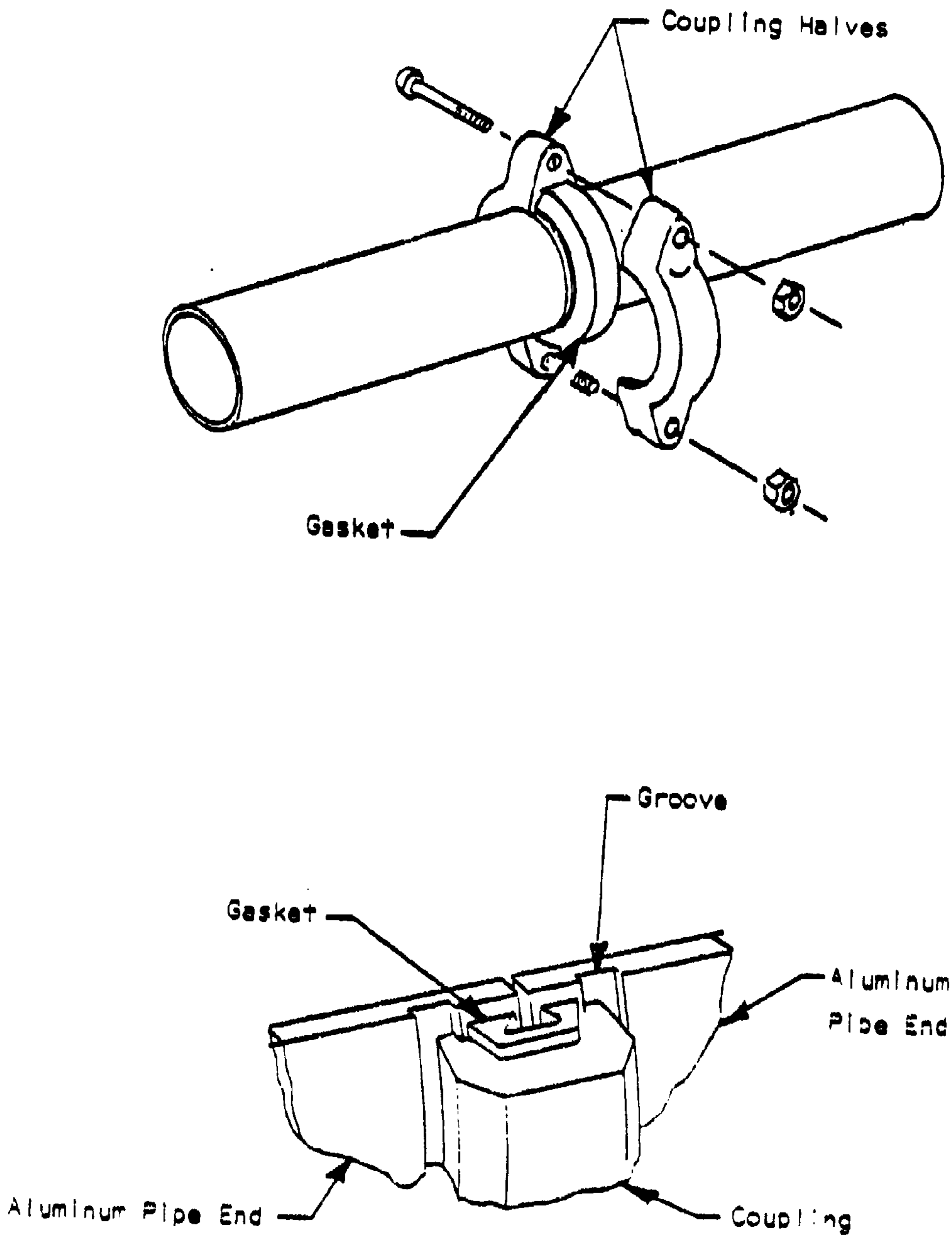



Figure 29. Grooved-end, mechanically coupled pipe.

It is proposed that the pipe stringing and joining be accomplished as a single operation. The procedure to be used would be as outlined in TM 5-343 for construction of coupled steel pipelines. The VECO investigation concluded that one crew can lay 70 sections of 8-inch pipe during a 10-hour shift. On this basis, achieving a 30-kilometer-per-day construction rate would require 70 crews working 35 crews per shift and two 10-hour shifts per day. Construction rates actually achieved during tests at Fort Belvoir using steel tubing indicates this estimate of possible construction rates is extremely pessimistic.

A 2-½-ton truck tractor and a bolster trailer would be required to supply pipe to each of the 35 crews. Additional equipment required would include ten 2-½-ton pipeline construction trucks and ten ¾-ton utility trucks. The 2-½-ton pipeline construction trucks would be equipped with winches and A-frames.

Delivery of 35,000 barrels per day through an 8-inch-diameter, schedule 40, coupled aluminum pipeline would require five pump stations. Each pump station would operate at a maximum discharge pressure of 800 lb/in$^2$ delivering approximately 475 hydraulic horsepower.

(3) **Concept 225F1 – Race and Race Racebilt.** This concept proposes using schedule 10, 6063 aluminum pipe. The pipe is joined by a mechanical coupling manufactured by Race and Race, Inc., Winter Haven, Florida. Marketed under the registered trademark Racebilt, each length of pipe has a female coupling and male fitting permanently attached by welding. Two sections of pipe are joined by inserting the male end into the female coupling as shown in Figure 30. The cast male fitting has two latching lugs. As the male fitting is inserted into the female coupling, spring-loaded latch rings automatically engage the latching lugs, providing a positive

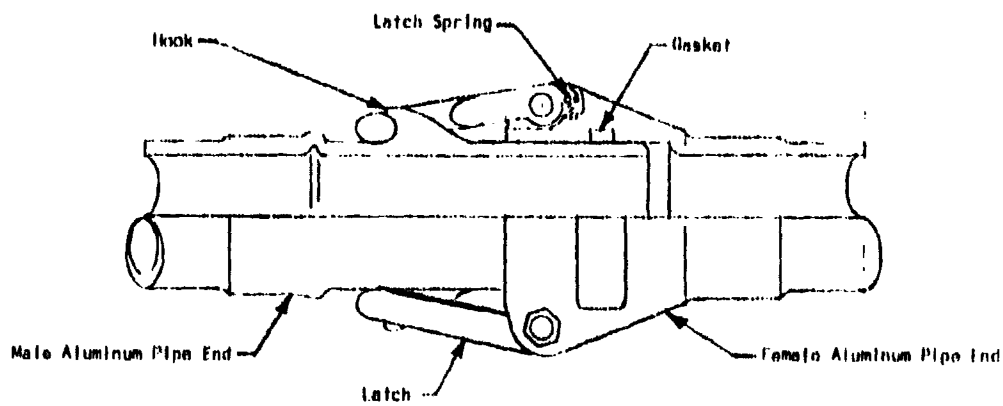



**Figure 30. RACEBILT mechanical coupling**

lock. An elastomeric seal in the bore of the female coupling provides a seal around the outside of the male fitting. An undercut on the latching lugs prevents release of the latches while the coupling is under pressure.

The coupling increases the useful length of a section of pipe by 0.58 foot. With the coupling attached, a 40-foot section of schedule 10, 8-inch-diameter pipe weighs approximately 205 pounds. Although it will be an arduous task, these sections of pipe can be manhandled for stringing and joining.

The stringing-and-laying procedure proposed by VECO is identical to that for the CIBA-GEIGY pipe except the coupling automatically latches itself, eliminating the need for a crew member to secure the joint. The estimated time required to lay one joint of pipe is 54 seconds. Based on this joining rate, VECO projects 6 crews (3 crews working two 10-hour shifts) can lay 30 kilometers of pipe per day and have enough time available to install the necessary valves and fittings, make grade crossings, etc.

Each construction crew would consist of 15 men including a crew chief, 7 men to carry, align, and join the pipe sections, 2 men to install valves and fittings; 2 men to carry and position cribbing; and 3 men working on the delivery truck to assist with off-loading the pipe.

Assuming 5-ton truck tractors towing flatbed semitrailers with telescoping bodies are used to haul 36 lengths of pipe per load, 40 trucks would be needed to support construction of 30 kilometers of pipeline per day. Additional equipment required would include five 2-½-ton cargo trucks and two ¼-ton utility trucks. Two of the 2-½-ton trucks would be equipped with winches and A-frames for handling valves and other heavy items.

With the maximum operating pressure for 8-inch-diameter schedule 10 aluminum pipe limited to 350 $lb/in^2$, a 100-mile-long pipeline would require nine pump stations to deliver 35,000 barrels of fuel per day. Each pump station would produce approximately 215 hydraulic horsepower.

**(4) Concept 240E1 – ZAP-LOK Systems International, Inc. (ZAP-LOK).** The ZAP-LOK pipe joining process, developed by ZAP-LOK Systems International, Inc., Houston, Texas, produces a joint equal in burst strength to the original pipe strength. One end of each section of pipe is expanded or "belled" as shown in Figure 31. The opposite end of each length of pipe is beveled slightly and an annular groove is rolled into the outside diameter. A portable hydraulic press forces the grooved end of one pipe section into the belled end of another pipe section.

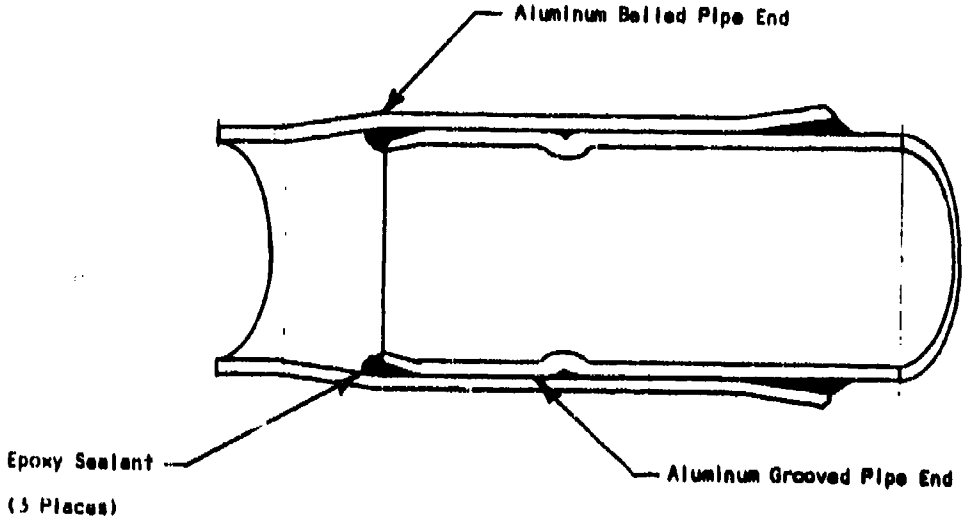



**Figure 31. ZAP-LOK Joint.**

The end preparation, belling and grooving, can be accomplished at the pipe mill, in a storage yard, or on the job site. The controlled-force fit of the joint provides a metal-to-metal seal. An epoxy applied before assembling the joint serves as a lubricant and a secondary seal. The joint reduces the useful length of an 8-inch pipe by approximately 0.88 foot.

The ZAP-LOK process is suitable for joining pipes from 1-inch through 12-inch diameter of various wall thicknesses and materials. VECO has recommended use of 8-inch-diameter, schedule 40, 6061-T6 aluminum pipe with a working pressure of 1,000 lb/in$^2$. A 40-foot length of this pipe weighs approximately 395 pounds.

The proposed installation procedure begins with stringing the pipe using a side-boom tractor to unload the pipe from the truck. The pipe would be placed on cribbing to protect the pipe and facilitate the joining crew.

The hydraulic joining press would be carried by a side-boom tractor and operate using power from the tractor hydraulic system. The joint of pipe being added to the pipeline would be picked up by another side-boom tractor,

the ends inspected, and the epoxy applied to the mating surfaces. After properly aligning the new section of pipe, the hydraulic press would grasp the pipe and force the joint together.

The time required to join a section to the pipeline is estimated to be 90 seconds. At this rate, eight crews of four crews per shift working two 10-hour shifts per day would be required to construct 30 kilometers of pipeline per day.

Two stringinig crews of 5 men each would be required to string the pipe in advance of the joining crews. Each crew would consist of a crew chief, a tractor operator, and 4 men to assist in handling and positioning the pipe. Each joining crew would consist of 7 men: 1 crew chief, 2 tractor operators, 3 men to assist in handling the pipe and applying the epoxy, and 1 man to operate the joining machine.

Five-ton truck tractors towing flatbed semitrailers with telescoping bodies would be used to deliver the pipe to the construction site. Assuming each truck can haul 26 sections of schedule 40, 8-inch-diameter pipe in 40-foot lengths, it is estimated that 35 trucks would be required to string 30 kilometers of pipe per day. Two side-boom tractors would be required for stringing the pipe and another eight side-boom tractors would be required for joining the pipe.

Delivery of 35,000 barrels of fuel per day through 100 miles of 8-inch aluminum pipe at 1000 $lb/in^2$ maximum operating pressure would require four pumping stations. Each pumping station would produce appproximately 590 hydraulic horsepower.

(5) **Results of Concept Comparisons.** Table 8 presents tabulated weight and volume data for the four selected concepts. All equipment dimensions and weights are actual values, unless noted otherwise. An additional 10 percent of total amounts (based on 100-mile pipeline) is included in the calculations, as noted, to compensate for quantities of pipe lost or damaged in transit.

Material and equipment cost data shown in Table 9 are based on 1976 manufacturer's quotations. For the ZAP-LOK system, pipe preparation cost does not include spare or back-up equipment. The cost of using a grooving machine, as an alternative to having the mill perform the grooving operation, is not included in pipe preparation cost for the grooved-pipe coupling system.

Transportation costs are based on MERADCOM "Cost Estimating Guidance Transportation Cost" statement of 9 Sep 1975. Costs include U.S. Line Haul, U.S. Port Handling, Overseas Port Handling, and Overseas Line Haul charges. Figures for U.S. Line Haul and U.S. Port Handling charges for pipe only are computed from volume (for low-density items, charges are based on volume rather than weight).

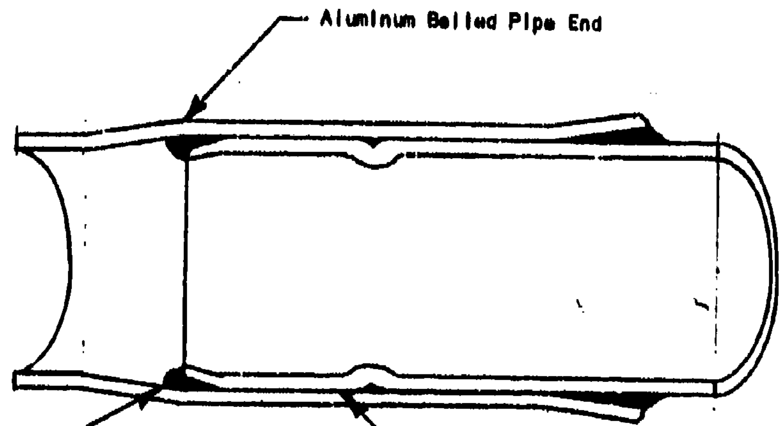



Figure 31. ZAP-LOK Joint.

The end preparation, belling and grooving, can be accomplished at the pipe mill, in a storage yard, or on the job site. The controlled-force fit of the joint provides a metal-to-metal seal. An epoxy applied before assembling the joint serves as a lubricant and a secondary seal. The joint reduces the useful length of an 8-inch pipe by approximately 0.88 foot.

The ZAP-LOK process is suitable for joining pipes from 1-inch through 12-inch diameter of various wall thicknesses and materials. VECO has recommended use of 8-inch-diameter, schedule 40, 6061-T6 aluminum pipe with a working pressure of 1,000 lb/in$^2$. A 40-foot length of this pipe weighs approximately 395 pounds.

The proposed installation procedure begins with stringing the pipe using a side-boom tractor to unload the pipe from the truck. The pipe would be placed on cribbing to protect the pipe and facilitate the joining crew.

The hydraulic joining press would be carried by a side-boom tractor and operate using power from the tractor hydraulic system. The joint of pipe being added to the pipeline would be picked up by another side-boom tractor,

Table 8. Weight and Volume of 8-Inch Pipeline

| | Concept 240E1 Zapata ZAP-LOK | Concept 225F1 Race & Race Racebilt | Concept 22341 Grooved-Pipe Coupling | Concept 220DB CIBA-GEIGY PRONTO-LOCK |
|---|---|---|---|---|
| Pipe Material | Sched 40 Aluminum 6061-T6 | Sched 10 Aluminum 6063-T6 | Sched 40 Aluminum 6061-T6 | Fiberglass-Reinforced Epoxy |
| Pipe Section Length-Actual (ft) | 40 | 41.08 | 20 | 40.67 |
| Pipe Section Length-Usable (ft) | 39.12 | 40.58 | 20.04 | 40 |
| Sections/100 Miles (10% Added) | 14.847 | 14,312 | 28,982 | 14.520 |
| Pipe Section Weight (lb) | 395.2 | 205.4 | 197.6 | 145.2 |
| Section Weight/100 Miles (lb) (10% Added) | 5,868,000 | 2,940,000 | 5,727,000 | 2,108,000 |
| Equipment Weight/100 Miles (lb)* (10% Added) | 203,000 | 118,000 | 439,000 | 68,000 |
| System Weight/100 Miles (lb)* (10% Added) | 6,071,000 | 3,058,000 | 6,166,000 | 2,176,000 |
| Section Volume/100 Miles ($ft^3$)** | 450,000 | 647,000 | 439,000 | 650,000 |

* Equipment includes sealant, lubricant, couplings, valves, anchors, thrust blocks, pipe preparation equipment (ZAP-LOK only).

** Volume for transportation.

Table 9. Pipe and Equipment Costs, 8-Inch Pipeline

| | Concept 240E1 Zapata ZAP-LOK | Concept 225F1 Race & Race Racebilt | Concept 22341 Grooved-Pipe Coupling | Concept 220DB CIBA-GEIGY PRONTO-LOCK |
|---|---|---|---|---|
| Unprepared Pipe (10% Added) | $5,108,000 | N/A | $4,986,000 | N/A |
| Pipe Preparation | 204,000* | N/A | 280,000** | N/A |
| Prepared Pipe | $5,312,000 | $3,596,000 | $5,266,000 | $2,701,000 |
| Couplings | N/A | N/A | 704,000*** | N/A |
| Joining | 1,745,000* | — | — | — |
| Consumable Materials**** | 3,000 | — | 3,000 | 3,000 |
| Subtotal | $7,060,000 | $3,596,000 | $5,973,000 | $2,704,000 |
| Transportation | 631,000 | 653,000 | 641,000 | 617,000 |
| Subtotal | $7,691,000 | $4,249,000 | $6,614,000 | $3,321,000 |
| Valves | 396,000 | 154,000 | 392,000 | 92,000 |
| Total | $8,087,000 | $4,403,000 | $7,006,000 | $3,413,000 |

* Represents total purchase price of manufacturer's equipment used, equipment also can be leased.

** Mill estimate (grooving operation).

*** Victaulic coupling under development, existing Aeroquip coupling priced at $782,000.

**** Includes sealant, lubricant.

The prices of gate valves for the systems are included because of the differences in cost for the different sizes required to handle the varying working pressures. A 600-pound class gate valve selected for the ZAP-LOK and grooved-pipe systems costs approximately $4100; a 300-pound class gate valve selected for the Racebilt system costs about $1700; and a 150-pound gate valve selected for the PRONTO-LOCK system costs approximately $1200. Valve quantities required for each system will also vary.

Table 10 provides a comparison of the basic pipe costs for 4-, 6-, and 8-inch nominal sizes.

Considering calculated total costs, exclusive of relative performance or manpower required for installation, the PRONTO-LOCK system costs the least of the four systems ($3,413,000); ZAP-LOK costs 2.4 times as much ($8,087,000); grooved-pipe 2.1 times as much ($7,006,000); and Racebilt 1.3 times as much ($4,403,000). The PRONTO-LOCK system costs represent costs for prepared pipe (ready to install), purchased directly from the manufacturer. There is no separate charge for preparing the pipe and no equipment required for the joining process.

The higher ZAP-LOK cost is attributed to a considerably higher price for schedule 40 aluminum pipe (versus FRP), the expense of preparing the pipe (belling and grooving), and a large expense for equipment to perform the joining operation. The cost of purchasing four joining presses ($1,745,400), of course, represents an initial cost only and a more accurate representation may be the long-term costs over the period of time the equipment is used. The high initial equipment cost would also be reduced if the presses were leased.

Similarly, the grooved-pipe system costs are higher because of high aluminum prices (versus FRP) and pipe preparation costs (grooving) in addition to the cost of the mechanical couplings employed. Since 20-foot pipe sections were used in the system design (versus 40-foot sections for other concepts) the number of joints are therefore doubled, pipe preparation costs and coupling costs could be halved if 40-foot sections are used.

The Racebilt system costs, however, represents the price of prepared pipe. The cost of the Racebilt aluminum pipe with the couplings welded on the ends is 35% less for pipe which is approximately 50% lighter in weight than the schedule 40 pipe.

For all four systems, the cost of consumable materials used in installation is relatively insignificant (under $4,000) compared to other costs. There is little variation in the transportation cost for all systems. Although the Racebilt

Table 10. Cost and Weight of 4-, 6-, and 8-Inch Pipe

| | Concept 220DB CIBA-GEIGY PRONTO-LOCK | Concept 22341 Aluminum Grooved-Pipe Coupling | Concept 225F1 Race and Race Racebilt | Concept 240E1 Zapata ZAP-LOK |
|---|---|---|---|---|
| Effective Length per Section (ft) | 40 | 20.04 | 40.58 | 39.12 |
| Section Weight (lb) | | | | |
| 4-Inch Nominal | 44.0 | 74.7 | 86.1 | 149.3 |
| 6-Inch Nominal | 74.8 | 131.3 | 142.4 | 262.6 |
| 8-Inch Nominal | 145.2 | 197.6 | 205.4 | 395.2 |
| Sections per 100-mile String (Plus 10%) | 14,520 | 28,982 | 14,312 | 14,847 |
| Pipe Cost per 100-mile String (1000 dollars)* | | | | |
| 4-Inch Nominal | 1,095 | 2,303 | 1,529 | 3,743 |
| 6-Inch Nominal | 2,036 | 4,148 | 2,600 | 5,346 |
| 8-Inch Nominal | 2,704 | 5,973 | 3,596 | 7,060 |

*Includes special equipment, couplings, and consumable materials where applicable.

system weighs considerably less than either the ZAP-LOK or grooved pipe system and the PRONTO-LOCK system weighs less than Racebilt, respective transportation charges are calculated on volume, which differs little.

The results of comparisons of the four selected concepts using the Pipeline Scoring Matrix, Figure 23, are shown in Figure 32. On the basis of the

| | |
|---|---|
| 220DB | CIBA-Geigy "Pronto-Lock" |
| 22341 | Aluminum Grooved-Pipe Couplings |
| 225F1 | Race and Race "Racebilt" |
| 240E1 | "Zap-Lok" |

| | | PROPOSED CONCEPTS | | | |
|---|---|---|---|---|---|
| | | 220DB | 22341 | 225F1 | 240E1 |
| PROPOSED CONCEPTS | 220DB | | 750 / 766 | 798 / 778 | 673 / 786 |
| | 22341 | 766 / 750 | | 781 / 766 | 691 / 791 |
| | 225F1 | 778 / 798 | 766 / 781 | | 665 / 805 |
| | 240E1 | 786 / 673 | 791 / 691 | 805 / 665 | |

**Figure 32. Comparison scores of proposed pipe concepts.**

performance and design criteria selected, the Racebilt system placed the highest of the four selected systems, folowed by PRONTO-LOCK, aluminum grooved-pipe couplings, and ZAP-LOK. The ranking is based on comparison scores of all four systems taken in pairs, i.e. Racebilt had higher scores when paired with each of the other systems; ZAP-LOK had lower scores for all three pairings. The scoring indicates that, considering a wide range of operating conditions and general requirements, Racebilt is the superior system. However, it is recognized that under certain circumstances and given specific requirements, another system could perform as well as or better than Racebilt. All systems, therefore, are capable of meeting the Military requirements.

On the basis of material and system design characteristics which served as the basis for scoring, pipe sections for all systems have the same nominal diameter. The pipe material then is a critical factor, With the exception of PRONTO-LOCK, all systems use aluminum pipe that can be bent as required in the field. The ZAP-LOK system is the most permanent of the four, heaviest in weight, and allows the longest unsupported length of pipeline. ZAP-LOK operates under the highest working pressure (1000 $lb/in^2$), hence requires fewer pumping stations thereby increasing the maximum system reliability. Conversely, the PRONTO-LOCK system can be disassembled and reused and employs the lightest sections of pipe, but its nonmetallic construction requires more support per pipeline length and is more vulnerable to abuse (from terrain) than any of the other systems. PRONTO-LOCK operates under the lowest working pressure (150 $lb/in^2$) and, on the basis of the number of pumping stations required, this limits the maximum mathematically possible system reliability that can be achieved.

The installation procedures individually selected for the four systems were considered by VECO to be the most efficient means of achieving the required installation rate. Racebilt required the least amount of skill to install. The joining operation involves little more than aligning two mating pipe ends and bringing them together with enough thrust to lift two spring-loaded latches over two lugs. The grooved-pipe and PRONTO-LOCK systems also are relatively easy to install. The ZAP-LOK system requires the most skill to install thus making desirable for the joining machine operators to have some prior training. Since the ZAP-LOK joint is relatively permanent, an improperly made joint is not readily corrected, meaning some delay in the construction operation. ZAP-LOK pipe, for that matter, can be joined only with mechanized equipment, whereas Racebilt, PRONTO-LOCK, and grooved-pipe sections can be assembled by hand. The grooved-pipe installation requires the longest time per joint (8.55 minutes) and Racebilt the shortest (54 seconds). Installation times are clearly subject to climatic conditions at the site. Low temperatures would affect the time required to apply and cure the epoxy used in the ZAP-LOK system. All systems except Racebilt would require low-temperature lubricants. All factors considered, Racebilt is the easiest system to install and ZAP-LOK the most difficult.

Racebilt also requires the least number of crews (total manpower) to meet the required installation rate, and grooved-pipe requires the most manpower (Table 11). A Racebilt crew requires no tools for installation; a PRONTO-LOCK crew requires only the use of a spanner wrench; a grooved-pipe coupling crew would use a torque wrench and an alignment cage; and a ZAP-LOK crew would use a hydraulic press to join pipe. Major installation, supply, and joining equipment requirements for each system (shown in Table 12) are dependent on many variables. Although no one system is superior in terms of equipment utilized, each system requirement is large considering an operation of this scale. Once the pipe is joined, the ZAP-LOK system is the most difficult of the four to repair and maintain. Replacement of a damaged section would require a crew to cut out the damaged section and to bell and groove mating ends in the field unless another joining operation is considered. Repair of damaged sections in the other systems would require simple replacement of the damaged sections.

Development of any of the four concepts into a Military system would not require an extended time period, nor would it involve a high risk. All the system concepts are based upon commercially proven components. There are, however, certain areas which require investigation if the systems are to perform satisfactorily in the Military environment.

The durability of the CIBA-GEIGY PRONTO-LOCK fiberglass pipe material would need to be established with respect to ultraviolet (sunlight) exposure, extreme cold temperatures, and physical abuse. The integrity of field-bonded pin-end (male) fittings when accomplished under extreme climatic conditions would need to be established. The characteristics and limitations of the grooved-pipe system using malleable iron couplings and steel pipe are well established. Similar limits with regard to strength and durability would need to be set for the aluminum system. The primary areas of concern with the Race and Race Racebilt system would be the strength of the fairly light gage (schedule 10) pipe, the durability and vulnerability of the cast-end fittings, and the effectiveness of the rubber seal at low temperatures and low line pressures. Development of the ZAP-LOK system would involve developing a "militurized" version of the joining equipment, tailoring the equipment for 4-, 6-, and 8-inch pipe only, and other similar changes. A reliable means for applying the epoxy sealant in extreme cold and wet conditions would also need to be developed.

Based on the results of their contract effort, the Value Engineering Company concluded:

Table 11. Manpower Requirements for Pipeline Construction

| Operation | Number of Men Required Per 10-Hour Shift | | | |
|---|---|---|---|---|
| Operation | Concept 240E1 Zapata ZAP-LOK | Concept 225F1 Race & Race Racebilt | Concept 22341 Grooved-Pipe Coupling | Concept 220DB CIBA-GEIGY PRONTO-LOCK |
| Stringing | 10 | N/A | N/A | N/A |
| Supply (Truck Operators) | 35 | N/A | N/A | N/A |
| Joining | 28 | 45 | 315 | 48 |
| Supply (Truck Operators) | | 40 | 35 | 38 |
| Valve Installation | N/A | N/A | 20 | N/A |
| Depot | | | | |
| Pipe Preparation | 5 | N/A | N/A | N/A |
| Loading | 12 | 36 | 40 | 40 |
| Subtotal | 90 | 121 | 410 | 126 |
| Clearing, Surveying (Day shift only) | 12 | 12 | 12 | 12 |
| Inspect, Test, Repair | 5 | 5 | 5 | 5 |
| Total Men per Shift | 107 | 138 | 427 | 143 |

Table 12. Equipment Requirements for Pipeline Construction

| Operation/Equipment | Items of Equipment Required to Install 30 Kilometers of Pipeline per Day | | | |
|---|---|---|---|---|
| | Concept 240E1 Zapata ZAP-LOK | Concept 225F1 Race & Race Racebilt | Concept 22341 Grooved-Pipe Coupling | Concept 220DB CIBA-GEIGY PRONTO-LOCK |
| Stringing | | | | |
| Flatbed Semitrailers | 35 | N/A | N/A | N/A |
| Unloaders/Tractors | 2 | N/A | N/A | N/A |
| Joining | | | | |
| Flatbed Semitrailers | N/A | 40 | N/A | 38 |
| Bolster Truck w/Trailer | N/A | N/A | 35 | N/A |
| Unloaders/Tractors | 8 | N/A | N/A | N/A |
| Joining Equipment | 4 | N/A | N/A | N/A |
| Valve Installation | | | | |
| 2-½-ton Supply Trucks | 1 | 2 | 10 | 2 |
| Depot | | | | |
| Pipe-Forming Equipment | 1 | N/A | * | N/A |
| Cranes | N/A | N/A | N/A | N/A |
| Other Utility Trucks | 11 | 11 | 10 | 13 |

* Optional: pipe can be purchased grooved.

On the basis of contact with professional and trade organizations and private industry, only a few areas of pipeline technology have shown marked progress or development in the last several years. For example, automatic welding techniques have improved the quality of joints; but because there has been no great reduction in time required, rapid-welded pipeline installation is not possible. Hose is relatively versatile and can be easily transported and installed; but it's application is limited by its low working pressures.

As a result of the information obtained and on the basis of preliminary findings, the development of an effective concept for rapid installation of a system for distribution of bulk fuel appears feasible, using relatively proven technology.

All four concepts under consideration appear to be superior, on the limited basis of the preliminary evaluation, to the Military systems currently available.

Of the four system concepts, the Racebilt system ranks highest by the scoring matrix criteria, while the PRONTO-LOCK system has the lowest projected costs. The ZAP-LOK system required the fewest men. Depending on the Army's area of emphasis, any of the four concepts explored would be suitable for further development as a military system.

9. **Ancillary Equipment.** In addition to the pipe and pumping equipment, there is a wide variety of components required for safe, efficient pipeline operation. Design requirements for each of these pipeline components are dependent on many factors, particularly the pipeline size, operating pressure, and flow rate. Selection of the proper ancillary equipment is an essential part of designing a well-integrated pipeline system.

A detailed examination of each type of component included in a pipeline system is beyond the scope of this report. Thus, the following discussion is intended only to identify some of the major issues that must be considered in pipeline design. To the extent possible, the potential impact on overall system cost and operational effectiveness is presented.

a. **Pump Station Manifolds.** A typical layout of a pump station, including four pumps interconnected for series operation, is illustrated in Figure 33. This manifold layout allows maximum flexibility in the series mode of pump station operation. Any desired combination from one to four pumps may be operated simultaneously. Valves in the manifold allow each pump to be isolated from the manifold pressure.

The pump station manifold can be considered to consist of several discrete sections. The incoming pipeline or trunk line terminates at the inlet to the incoming pipe cleaner station. This section of the manifold is required to catch or trap, without interrrupting flow, any internal pipe scrapers, pigs, or other pipe cleaning devices being pumped through the pipelines. Similarly, the outgoing pipe cleaner station provides the capability to introduce scrapers, pigs, or other cleaning devices into the flow stream as it leaves the pump station.

The intake sandtraps collect dirt, scale, sludge, and other debris pumped through the pipeline following intital startup, after the line has been broken for maintenance, or,that has been loosened by an internal scraper or pig. Sandtraps are intended to remove large particles and debris which might damage a pump, lodge in valves rendering them inoperative, or otherwise cause operational problems. Sandtraps are not intended to serve as quality control devices.

The unit pump manifold identified in Figure 33 includes that part of the pump station manifold required to connect one pump to the pipeline. Thus, the pump station manifold, as shown, includes four unit pump manifolds. It is this portion of the manifold that changes if the pump station is designed for parallel pump operation. Figure 34 shows schematically a typical layout for series operation of a four-pump station. Incoming and outgoing pipe cleaner stations and a sandtrap station identical to those illustrated in Figure 33 would be required with the manifold connecting the pumps in parallel.

The number of valves, fittings, and pipe sections are approximately equal for either pump stations operating in parallel or series assuming each station includes an equal number of booster pumps. Examination of Figures 33 and 34 shows that a substantial amount of construction effort will be required if a complete manifold is delivered to the installation site as individual components. The size and weight would preclude shipping a pump station manifold preassembled as a complete unit. However, it would be possible to preassemble the incoming and outgoing pipe cleaner stations, intake sandtraps, and at least the major portion of the pump manifolds as separate units.

(1) **Valves.** Control of flow in a pipeline system is accomplished by the use of valves. Essentially, valves perform the following basic functions:

(a) Start or stop flow.

(b) Determine and change direction or path of flow.

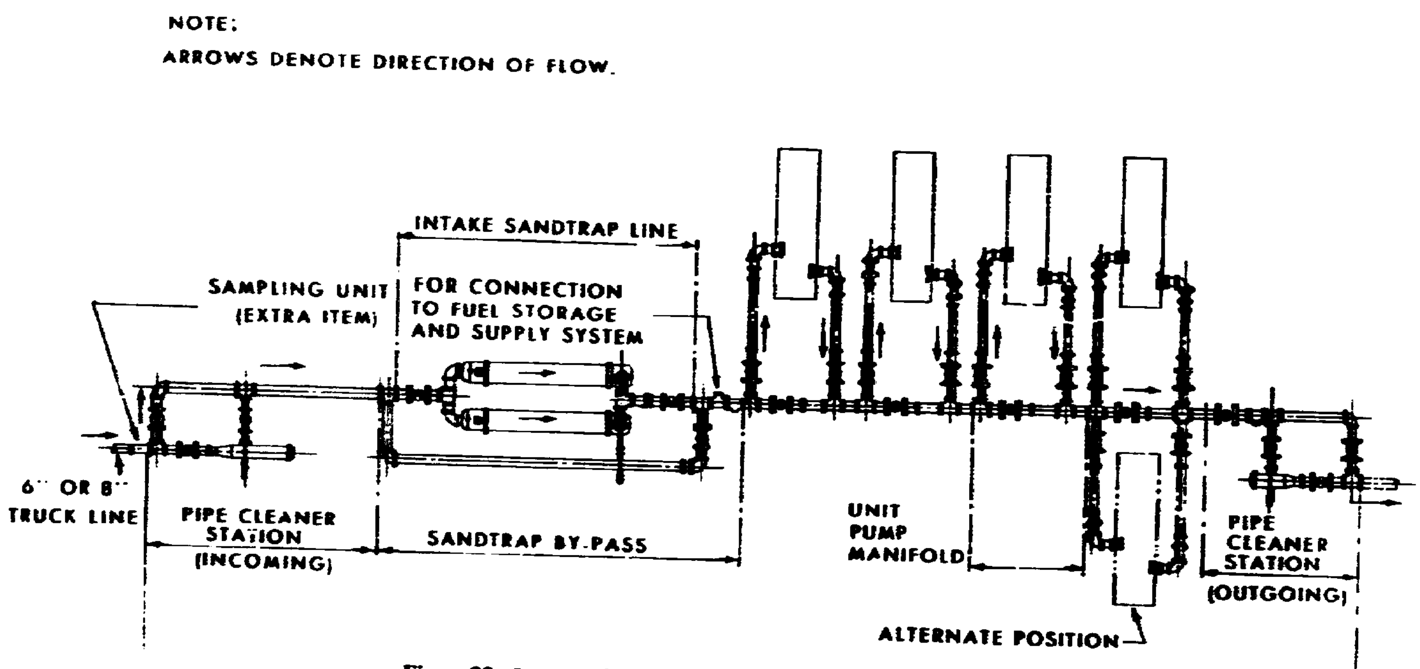



Figure 33. Layout of pump station manifold for series operation.

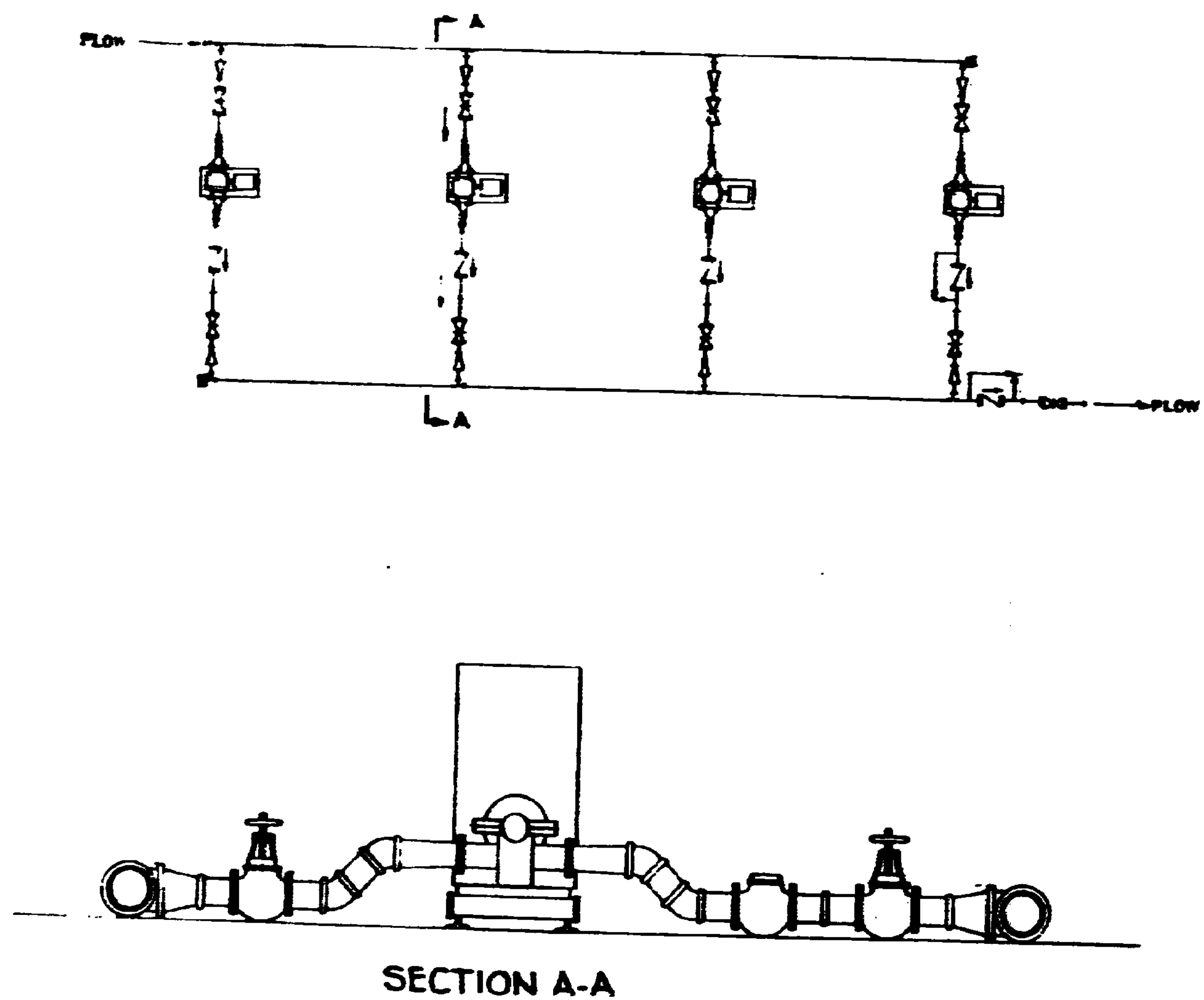



Figure 34. Schematic of pump station manifold for parallel operation.

(c) Prevent backflow.

(d) Relieve or regulate pressure.

To meet the varied flow control requirements, the list of types of valves is virtually endless. Valves must be properly selected and maintained to provide the desired service. Because of the many types of valves available and the differing operational requirements, this study does not attempt to make an in-depth evaluation of valves. Instead, this discussion is limited to the contribution of valves to the cost and weight of pump station manifolds.

Table 13 lists the approximate cost and weight of cast steel rising stem gate valves and swing check valves. The manifold for a single pump booster station of the configuration shown in Figure 33 includes at least 10 gate valves and 2 check valves. For each additional pump added to the manifold, 2 gate valves and 1 check valve are required. Because of the number of valves required and their high cost and weight, valves represent more than half the total cost and weight of a pump station manifold.

Although difficult, it would be possible to install 4-inch, 150- and 300-pound-class valves and 6-inch, 150-pound-class valves without the aid of materials handling equipment. Beyond these sizes and weights, it becomes essential to have some type of support equipment available for valve installation. Even then, assembly of pump station manifolds will be a slow, laborious task requiring several men. Maximum preassembly of pump station manifolds will substantially reduce the time and manpower required for pump station construction.

Improvements in valve technology in recent years have been primarily through the introduction to new materials. Coatings applied to internal valve parts have led to substantial improved performance of valves in highly corrosive applications. Reinforced-plastic valves are finding acceptance for some low-pressure applications. Typically, butterfly valves are smaller, weigh much less, and are less expensive than othe types of valves. Improved designs have led to greater utilization of butterfly valves for applications up to 150-lb/in$^2$ pressure differential. Butterfly valves with pressure ratings up to 720 lb/in$^2$ for sizes up to 12 inches have recently become available from a few manufacturers. Virtually no data is available regarding the reliability and maintenance characteristics of the valves.

The changes in valve technology do not indicate a need for any significant change from the types of valves currently in use throughout the petroleum pipeline industry. However, future Military pipeline design and development programs should include a thorough survey of the valve industry to insure that no opportunity for improvement has been overlooked.

Table 13. Cost and Weight of Pipeline Valves

| Description | ASA Class | Pressure Rating ($lb/in^2$) | Cost ($) | | | Weight ($) | | |
|---|---|---|---|---|---|---|---|---|
| | | | 4-Inch | 6-Inch | 8-Inch | 4-Inch | 6-Inch | 8-Inch |
| Gate valve, rising stem, cast steel | 150 | 275 | $ 472 | $ 771 | $1206 | $ 130 | $ 213 | $ 382 |
| | 300 | 720 | 673 | 1173 | 1766 | 226 | 450 | 700 |
| | 600 | 1440 | 1392 | 2570 | 4386 | 334 | 706 | 1154 |
| Swing check valve, cast steel | 150 | 275 | 432 | 642 | 1105 | 119 | 177 | 251 |
| | 300 | 720 | 526 | 1059 | 1814 | 204 | 333 | 537 |
| | 600 | 1440 | 1050 | 1811 | 3299 | 378 | 550 | 895 |

(2) **Fittings**. Included in the broad category of fittings are elbows, tees, wyes, crosses, reducers, unions, plugs, return bends, and many other specialty items. Fittings used to connect the various parts of a system may be made of a wide range of materials to meet various service requirements. Fittings are manufactured in a wide range of standard types and sizes for use with all types of mechanical couplings as well as for welded joints.

Individually, fittings represent a small part of the cost and weight of a complete manifold. However, due to the large number required in a complex manifold, fittings may represent a significant part of the total manifold cost and weight. Use of standard fittings and elimination of the need for specialty items is an essential part of good manifold design.

Virtually every conceivable technique suitable for joining pipe could be used for connecting parts within a pump station manifold. However, the technique best suited for joining pipe may not necessarily be well suited for all joints within a pump station manifold. For example, welding would not be a suitable means for connections to pumps, valves, and other components which may require removal for repair or replacement.

Selection of the type of fittings and method of joining to be used within pump station and tank-farm manifolds will require careful study after the pipeline joining method is selected. It is important to remember that leaks are most likely to occur at mechanical joints. Thus, it is imperative that the fittings selected have a pressure rating compatible with the pipeline operating conditions and that the number of joints be held to a minimum. The versatility of the currently standard Military grooved-end mechanical couplings makes this joining technique extremely well suited for Military applications. Other than the possible use of aluminum fittings in lieu of steel to reduce maintenance requirements, preliminary evaluations indicate little potential for improving Military manifold designs.

The approximate cost of pump station manifolds using grooved couplings is shown in Figure 35. The top three curves represent the estimated costs for 4-, 6-, and 8-inch-nominal-diameter manifolds for a one pump station of the general configuration illustrated in Figure 33. The cost of a unit pump manifold is represented by the lower three curves in Figure 35. Thus, the cost of a complete four-pump manifold as shown in Figure 33 would be equal to the cost read from the manifold cost curve for the appropriate line size plus three times the unit manifold cost read from the appropriate unit manifold curve in Figure 35.

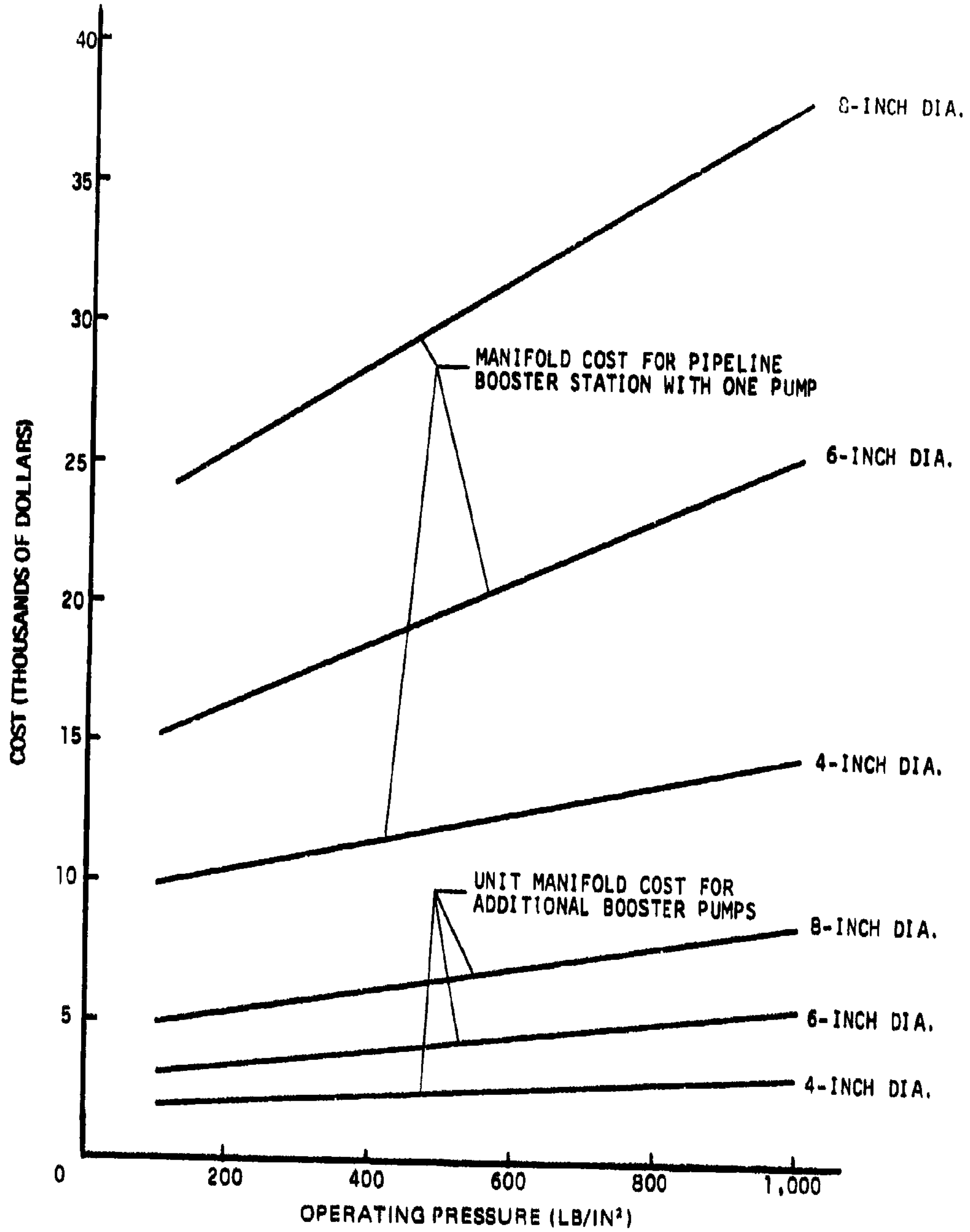



Figure 35. Cost of pump station manifolds.

Using the same approach, the approximate weight of a pump station manifold can be determined using Figure 36. The valves, sand trap, and cleaner stations comprise the majority of a pump station manifold cost and weight. Use of a joining technique other than grooved couplings would have little effect on the total manifold cost or weight. Therefore, Figures 35 and 36 are used throughout this report as being representative of pump station manifold costs and weights without regard for the pipeline joining technique used.

b. **Pressure Regulation.** The pressure at any point in a pipeline is a function of both static and dynamic head. Because of the effects of gravity, liquids always tend to move toward the center of the earth. This characteristic creates a pressure, normally referred to as static head, proportional to the vertical distance between the liquid surface and the point where the pressure is measured.

If flow in a pipe is uphill, the static head resists flow and adds to the energy that must be supplied by a pump to obtain the desired rate of flow. When flow is downhill, the static pressure tends to push the liquid through the pipeline, helping to overcome the friction loss from the fuel flow. When pumping is interrupted and the pipeline is shut down, there is no friction loss to offset the static head. Thus, on the downhill run, the pressure at the lowest point in the pipeline may be higher under no-flow conditions than when flowing.

The need for pressure regulation in Military pipeline was first identified during World War II. Construction of pipelines over the Himalaya Mountains in China-Burma Theater and in mountainous terrain, such as found in Northern Italy, found locations where static pressure could become excessive.

In 1953, the U.S. Army Engineer Research and Development Laboratories (USAERDL) at Fort Belvoir, Virginia, initiated a study of the problem of Military pipeline pressure regulation requirements. Approximately 22 manufacturers were contacted to determine commercial availability of suitable pressure regulating equipment. At that time, one valve manufacturer's control equipment appeared suitable for application to a Military, portable, pressure-regulating station. Evaluation tests on these valves began in October 1954 and revealed that the rubber expandable tubes which were the prime components of the valves would not operate effectively at subzero temperatures and a low-temperature, fuel-resistant rubber was not available.

In 1956, a contract was awarded to Arland Engineering Company to investigate, evaluate, and select suitable pressure-regulating equipment. The final report, titled "Pressure-Regulation Valves for Military Pipelines," was submitted to the Chief of Engineers, U.S. Army, in April 1957. This report recommended a valve,

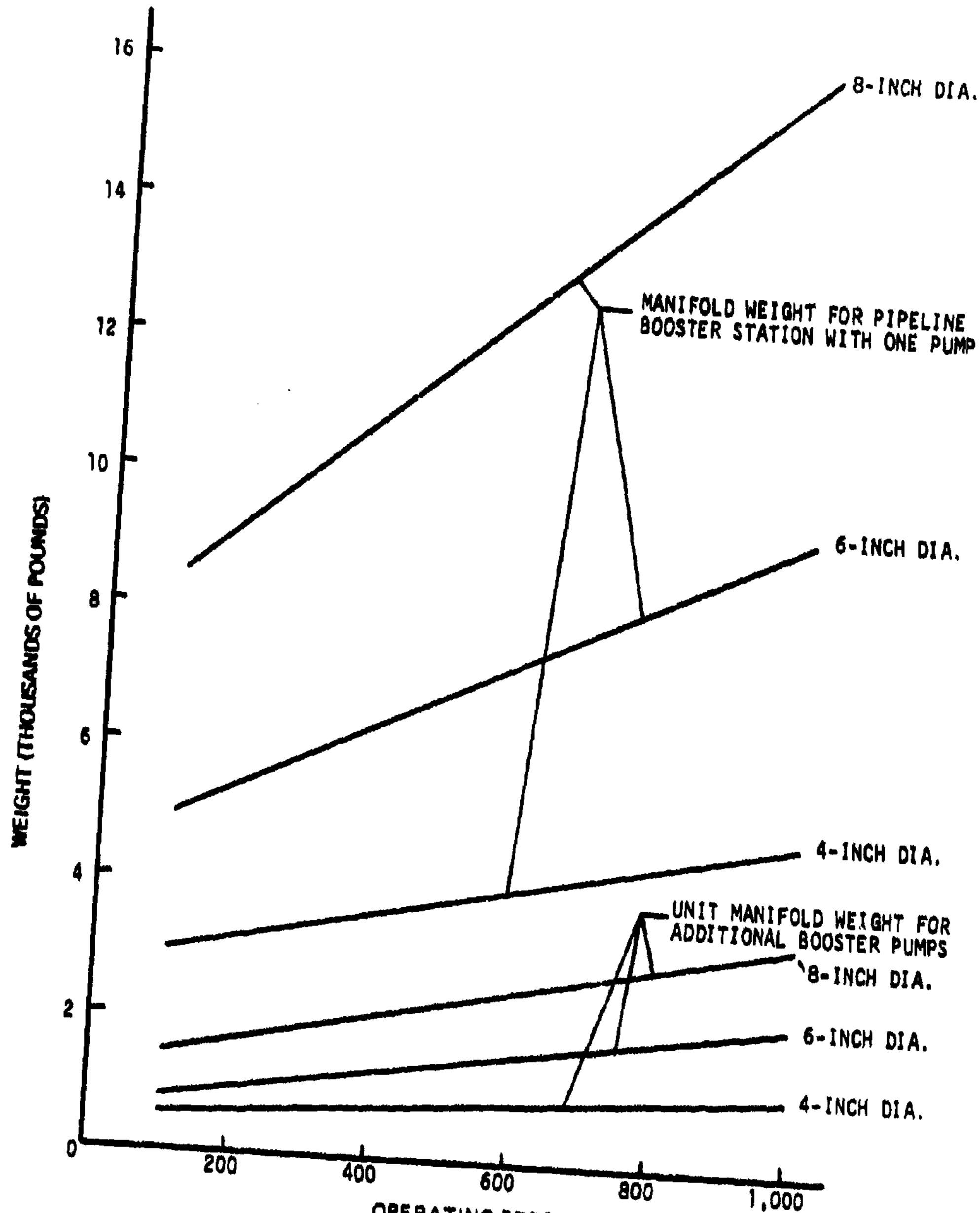



Figure 36. Weight of pump station manifolds.

but, again. it was ineffective for operation at subzero temperatues. During this study, assistance was solicited from 41 valve manufacturers, 35 pipeline operators, 5 pipeline design and engineering organizations, and the American Petroleum Institute Committee on Pipeline Transportation. The general comments received from these organizations indicated the following:

(1) The type of pressure regulation used on commercial overland pipelines is determined by the requirements of a specific location and application.

(2) Each design is peculiarly suited for that location.

(3) No two designs are necessarily alike.

The design restraints or requirements for commercial applications are, therefore, somewhat different from the initial Military objective of using one standard pressure-regulating station assembly for all requirements. It is noteworthy that, subsequent to this period (1956-57), the Military objective changed to one of applying a variety of pressure-regulating stations to meet all Military requirements, rather than one regulating station for all Military requirements. During this 1956-57 study, correspondence from a leading pipeline design and construction firm indicated that it was being faced with a complex pressure-regulating problem concerning a proposed pipeline over rugged terrain from Sicasica, Bolivia, to Arica, Chile. The remoteness of this pipeline indicated a need for pressure regulating stations that were operated solely by hydraulic pressure and were self-regulated, automatic, and unattended.

In June 1960, the Petroleum Equipment Branch (USAERDL) completed a study to investigate the requirements, methods, and equipment for pressure regulation in long, downhill, Military-pipeline sections or where the pipeline profile forms a deep gorge. Included in this study was additional testing and evaluation of an improved version of the regulating valve that was evaluated in the two previous studies of 1953 and 1956-57. As in previous programs, the testing found the valve would not operate effectively at subzero temperatures. Conclusions drawn in USAERDL Technical Report 1639-TR, "Pressure Regulation in Long Downhill Sections of a Military Pipeline," dated June 1960, by M. A. Pachuta, indicated additional study, design, and development were required to obtain suitable pressure-regulating equipment.

Commercial methods used to overcome the high pressure resulting from large changes in elevation include:

(1) Use of welded pipeline construction exclusively, which inherently will withstand higher pressures than coupled lightweight tubing.

(2) Change to a heavier-wall pipe in critical areas of a pipeline where excessive pipe pressures could be encountered.

(3) Use of a smaller-diameter pipe while maintaining or increasing the wall thickness to accommodate higher pressures associated with critical sections of pipeline.

(4) Installation of a relief valve and piping to a storage tank in areas of a pipeline where critical pressures could be encountered. Whenever excessive pressures occur, product is relieved to storage and later pumped back into the pipeline.

(5) Control of pipeline pressure limits with pressure-regulating valves or shutoff valves that are powered with an external source of power, such as electricity or compressed air.

(6) Control of pipeline pressure limits with pressure-regulating valves that are powered with the hydraulic pressure of the product being conveyed in the pipeline.

In November 1965, the Department of the Army approved a Small Development Requirement for a "Family of Pressure-Regulating Equipment, 6-, 8-, and 12-Inch, Military Petroleum-Products Pipelines."

In October 1969, Williams Brothers Engineering Company, Resource Sciences Center, was awarded Contract DAAK02-70-C-0119 for the design of portable pressure-regulating stations for critical downhill sections of 6-, 8-, and 12-inch Military petroleum fuel pipelines. The final report, "Portable Pressure-Regulating Systems for Critical Downhill Section of 6-, 8-, and 12-Inch Military Petroleum Pipelines, Report No. 2," dated September 1970, contains drawings and specifications for pressure-regulating stations.

Funding limitations at that time prevented the fabrication and testing of the Military-designed pressure-regulating station. However, Williams Brothers Engineering Company has built and installed a commercial station which is a modified version of the Military station. Complete details of the 1969-70 development effort are contained in Technical Report 2017, "Portable Pressure-Regulation Station for Critical Downhill Sections of 6-, 8-, and 12-Inch Military Fuel Pipelines," prepared by H. N. Johnston, Fuels Handling Equipment Division, MERDC, dated November 1971.

The need for a pressure-reducing station suitable for worldwide use remains. Specific operational requirements must be established based on the line sizes, operating pressures and flow rates determined to be suitable for future Military pipelines.

c. **Controls.** Today every commercial pipeline uses some automatic and/or remote controls. Some of the more sophisticated facilities allow one dispatcher to operate an entire complex pipeline system. This high degree of automation is made possible through the use of computers which can monitor almost every performance characteristic throughout the entire system. In some instances, the complex instrumentation, sensing a given condition, sends a signal to the computer which makes a decision and transmits a signal to the appropriate automatic control device making the necessary change in operating conditions. In other cases, the sensor signal may be displayed visually for the dispatcher to interpret and initiate the required action.

The pipeline control functions can be divided into two categories: dispatching and pump station control. The dispatching can include control of all storage terminals associated with the pipeline or can be limited to just the pumping equipment, control valves, and other components related solely to operation of the pipeline itself.

In a totally automated and centrally controlled system, each storage tank would be equipped with a sensor device to provide the dispatcher with an indication of the quantity of fuel in the tank. All flow-control valves would be operated by electric, hydraulic, or pneumatic actuators and controlled by a switch on the dispatcher control panel. Instrumentation would be required to provide the dispatcher with an indication of each valve position. In addition, the dispatcher would have the capability to start, stop, control, and monitor the performance of all pumping equipment associated with the system. Because of the remoteness of the dispatch r to many of the facilities, a complex data communications system is requireu.

There is not universal agreement among pipeline operating companies regarding the degree of automation that can be justified economically. Automated dispatcher control systems may prove feasible for commercial operation where, over a period of several years, the reduction in operator personnel cost may offset the initial investment costs. However, for the relatively short duration a Military pipeline would normally be in service, the high investment cost of an automated central dispatcher control capability cannot be justified.

Automation of some pump station control functions can be considered optional. However, there are some pressure control functions that require automatic control for safe, efficient pipeline operation.

Automation of the normal pump station startup and shutdown procedures are optional. Automatic controls are commercially available which, upon receiving the proper signal, will follow a prescribed procedure and sequence to start up the pump station and bring it to some predetermined operating condition. The same automatic controls can be programmed to shut down the entire station upon command from the dispatcher or from a local operator. As with fully automating a pipeline system for single dispatcher control, the cost of a fully automated pump station startup and shutdown capability is not economically justified for military pipeline systems.

The essential requirements for automatic pump station controls involve maintaining acceptable suction and discharge pressures for pump stations in tight line operations. The requirement for automatic control of pump station suction and discharge pressures can be seen by examining the hydraulic gradients shown in Figure 37. If, in Figure 37, the normal pump station operating conditions are 300 $lb/in^2$ disc'arge pressures and 20 $lb/in^2$ suction pressure, the normal hydraulic gradient will be as shown. If 300 $lb/in^2$ represents the maximum safe working pressure, controls must be provided to prevent a higher pressure. In addition, if 20 $lb/in^2$ is the minimum required suction pressure, the controls must prevent loss of suction pressure or damage to the pumps will result.

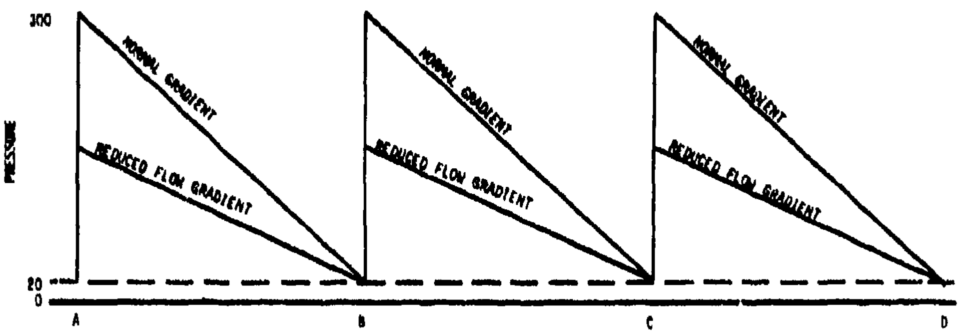



**Figure 37. Hydraulic gradient for three-pump-station pipeline.**

If the dispatcher wants to reduce the throughput rate, he will reduce the discharge pressures at pump station A. This pressure reduction will result in a reduced flow gradient between pump stations A and B. If pump stations B and C attempt to continue to operate on their normal gradients, the gradient between A and B will attempt to assume the same flow rate or gradient. Since the dispatcher has reduced the pressure at A, the only way the gradient between A and B can be made to assume a steeper angle than the reduced gradient shown is by lowering the suction pressure at B. Since 20 $lb/in^2$ is the minimum required suction pressure,

station B must reduce its flow rate to match the new flow rate at station A to maintain adequate suction pressure. Similarly, when station B reduces the throughput rate, station C must reduce its throughput rate to avoid a low suction pressure. In a totally automated pipeline, the control system will automatically adjust the operating conditions at stations B and C so that each station is pumping at the same rate it is receiving.

Without an automatic control system, the pump station operators must make these adjustments. Extremely close coordination between pump station operators is required. Every adjustment of operating conditions at one station affects the conditions at every other station along the pipeline. Unless the pump station operators follow the appropriate procedures, every adjustment of operating conditions becomes a continuous process of adjustments as each pump station "hunts" for the desired operating conditions.

A more serious condition occurs in the event of an unexpected change of operating conditions. Assume a pipeline system is operating at the conditions indicated by the normal gradient in Figure 37 and pump station B shuts down due to a mechanical failure of the pump. Operating conditions along the entire system will be affected immediately.

When station B shuts down, the suction pressure at B will rise because the pump is no longer taking the flow away from the receiving line. If pump station A attempts to continue operating at the same throughput rate, the increase in suction pressure will cause the normal gradient from A to B to move up trying to maintain the same slope. This results in an increase in discharge pressure at pump station A. If the 300 $lb/in^2$ discharge pressure at each pump station is equal to the maximum safe operating pressure, the increased operating pressure resulting from a pump failure at station B, will cause the discharge pressure at station A to exceed the maximum safe operating pressure. To avoid a possible line rupture, each pump station must have discharge pressure controllers to reduce the throughput rate or shut the system down in the event any pump station except the first is shut down unexpectedly.

Continuing the same example, when pump station B shuts down, the discharge pressure will drop. This drop in discharge pressure at station B will result in a drop in the suction pressure at station C. As discussed previously, the throughput rate at C must be reduced accordingly to avoid operating at conditions below the minimum required suction pressure.

A wide range of events can cause changes in pipeline operating conditions which necessitate adjustments in throughput rates. Other events may require total shutdown of all pump stations. For example, consider what would happen if, in Figure 37, station D is a receiving terminal and, in the process of

switching tanks, all valves are inadvertently closed before opening a valve to another receiving tank. All pump stations must be shut down immediately to avoid overpressuring the pipeline.

A sudden drop in operating pressure must also result in shutdown of all pump stations. A drop in suction pressure at station B, Figure 37, could result from numerous causes including an intentional reduction of throughput rate at station A or failure of one pump in a multipump operation. In both of these cases, it would be acceptable for stations B and C to immediately adjust their operating conditions to match station A and allow the system to continue to operate at a reduced rate. However, the drop in suction pressure at station B could also be a result of a partial rupture in the pipeline somewhere between stations A and B. If the rupture is very close to station B, the discharge pressure at station A will not be affected significantly.

Without the aid of extensive condition monitoring equipment to immediately identify the cause of any abrupt change in operating conditions, the pump pressure controllers must shut the pumps down when preset limits are exceeded. If after determining the cause of the shift in operating conditions it is found safe to resume operations, normal startup procedures can be followed to establish the desired operating conditions. This operating procedure will result in shutting the system down sometimes when it may not be necessary, but it also precludes unnecessary damage to equipment and excessive fuel spills.

In addition to monitoring and controlling suction and discharge pressure, safety devices are required to protect pumping equipment against excessive temperature of cooling water or lubricating oil, insufficient lubricating oil pressure, and overspeed of the pump engine. Sensors, transducers, actuators, and other automatic equipment suitable for monitoring and control of pipeline pumping conditions are available commercially. Very little specialized equipment is required to provide all necessary controls.

d. **Flow Measurement.** In the past, the Army has placed little emphasis on continuous-flow measurement devices as components of pipeline systems. However, recently there has been growing interest in volumetric measurement of fuels at all levels in Military fuels distribution systems. An examination of pipeline operations finds that accurate flow measurement data can be used profitably in the management and control of pipelines.

There are a variety of volumetric measurement techniques in commercial use today. A number of new flow-measuring instruments have been developed recently to satisfy exacting industrial requirements and to overcome many of the problems associated with traditional devices in special applications. Still, the

most common type of volumetric measuring devices in use today are positive displacement meters. These meters use some mechanical method to divide the flow through the meter into a sequence of fixed volumes. By counting the number of fixed quantities passing through the meter, a highly accurate measure of rate of flow is determined.

Desirable features of positive displacement meters include their high accuracy, long life, and direct drive of mechanical readout devices eliminating the need for an external power source. Disadvantages include high initial cost, difficulty in calibration, and heavy weight. A large number of manufacturers produce standard models of positive displacement meters offering every conceivable capacity range and pressure rating.

Vortex velocity meters have gained limited Military acceptance. In these meters, a paddle-wheel- or squirrel-cage-type rotor is mounted in an offset chamber so that one side of the rotor extends into the flow stream. The motion of the liquid through the meter turns the rotor at a speed proportional to the rate of flow. Like the positive displacement meters, vortex velocity meters can drive a mechanical readout device without using external power. If necessary, the meter can be used to drive a signal generator with the output fed to a remote electrical readout device.

The principal advantages of vortex velocity meters are low cost, light weight, and ease of maintenance and calibration. They have good accuracy over the rated flow range but suffer a relatively high pressure loss. At low flow rates, the meters are highly inaccurate; thus, it is imperative that a vortex velocity meter be of the proper size for the specific application.

Produced by Ball Manufacturing, Inc., North Salt Lake, Utah, the vortex velocity meter is available in five standard sizes, having rated flow ranges of 6 to 50, 25 to 200, 60 to 500, 120 to 1,000 and 260 to 2,600 gallons per minute. The 25 to 200 and 60 to 500-gal/min sizes conform to the requirements of Military Specifications MIL-M-82180 (MC).

A variety of meters have been evaluated by MERADCOM for volumetric measurement of fuel at large bulk-storage tank installations. Details of this test and evaluation program are contained in USAMERDC Report 2024, "Bidirectional Meter Used for Volumetric Measurement of Military-Standard Hydrocarbon Liquid Fuels at Bulk Storage Installations," dated February 1972, by Joe Medrano. As a result of this program, a vortex velocity meter is included as a part of the ancillary support equipment for the 25,000-barrel hasty storage reservoir type-classified in 1976.

Turbine meters use a multivane propeller or turbine rotor positioned so that the axis of rotation coincides with the centerline of the pipeline. Flow through the meter turns the rotor at a speed proportional to the rate of flow. Magnetic elements on the turbine blades passing sensing devices on the meter body generate an electronic pulse. This signal may be fed directly to a readout device or transmitted to a remote readout device. When properly installed, turbine meters are very accurate. However, they are sensitive to changes in pipeline configuration immediately ahead and downstream of the meter. The turbine meter is extremely lightweight, but remote electronic readout devices requiring explosion-proof cases can become bulky and heavy. Available from several manufacturers, turbine meters tend to be relatively expensive.

Pressure differential or orifice-type meters operate on the principle that a change in velocity of the liquid flowing through the meter produces a change in pressure. The amount of pressure change is dependent on and can be correlated to the rate of flow. Mechanical devices can be used to convert the differential pressure to an indication of flow rate. More commonly, it is desired to record the throughput. In this event, an electronic integrator must be used in conjunction with some type of timing device. Differential pressure meters are lightweight and low in cost. Problems include the need to calibrate the meter in the installed position and the accuracy that is affected by the specific gravity of the liquid being measured.

Several flow-measuring devices based on the vortex-shedding phenomenon have emerged recently. When a liquid must pass around a fixed obstruction in the flow stream, vortices form on the downstream side of the obstruction. The formation of these vortices is accompanied by a pressure pulse. The frequency of the pressure pulses involved in vortex formation can be related to fluid velocity which is a function of flow rate. A number of different types of obstructions are used to produce vortex formation depending on the technique used to convert the pressure pulses to an electronic signal that can be integrated with a time signal and fed to the desired flow measurement readout.

The Coanda effect and jet deflection principle have been borrowed from fluidic technology for two new methods of flow measurement. In each case the fluidic phenomenon is used to generate a pressure differential which is proportional to the rate of flow. Analog circuits convert the pressure differential data to an electronic signal which can be displayed on the desired readout device.

Other types of electronic flow meters use electromagnetic flow-sensing elements, ultrasonic Doppler-effect, and differential capacitance of pressure-sensing diaphragms. These volumetric measuring techniques along with the vortex-shedding, Coanda effect, and jet deflection principles involve no mechanical devices. Using solid

state electronics, these flow meters should have good reliability. Like pressure differential meters, these flow measuring devices should be calibrated in the installed position and they require a constant supply of electrical power.

Features to be considered in selecting the meter most suitable for Army field use include:

(1) Linear rangeability.
(2) Repeatability.
(3) Meter reproducibility.
(4) Sensitivity to viscosity.
(5) Meter factor adjustment.
(6) Meter factor consistency.
(7) Compatibility to fuels.
(8) Service life.
(9) Readout restrictions.
(10) Ease of maintenance.
(11) Calibration requirements.
(12) Physical characteristics.
(13) Cost.

Evaluation of meter characteristics against these factors finds the vortex velocity meters to most nearly satisfy all requirements.

e. **Product Loss Reduction Service.** Past history of military pipeline operations show large losses of fuels have occurred due to ruptured or broken pipelines. Some of these losses have resulted from operational failures; however, most of the losses have been the result of hostile actions, sabotage, or pilferage. As a result of such losses, the Vietnam Laboratory Assistance Program (VLAPA) requested a means of automatic shutoff in a pipeline so that fuel would not drain from the entire line in the event of damage or pilferage.

Under modification P0003 to MERADCOM Contract No. DAAK02-70-C-0119, Williams Brothers Engineering Company designed a system consisting of three major functional components:

(1) A full-opening ball valve equipped with an actuator controlled by back pressure, a lockout, and an exhaust pilot valve.

(2) An excess-flow pilot valve designed to function with an upstream orifice.

(3) A volumetric flowmeter.

Unique characteristics of the system are as follows:

(1) The system will shut down a pipeline in both the flowing and static conditions in case of a line break without using any external power sources.

(2) Maintenance may be performed with a minimum of special equipment.

(3) The system may be used in worldwide environments.

(4) The system provides a means to isolate and locate a line break.

(5) The system provides for the use of pipeline scrapers to clean the line.

The design study[22] concludes that:

(1) To provide a means of automatic shutoff for reduction of product loss due to failure or deliberate destruction of military pipelines, the line must be divided into sections with automatic shutoff valves, thus reducing the amount of product that will drain from the line at one point.

(2) By using different pressure settings on the exhaust pilot valves, operating personnel will be able to determine the general location of a line failure under static conditions.

(3) The flow rate indicator can be used to locate leaks while the line is operating in flowing conditions.

(4) The designed system should be satisfactory for military use.

This system can play an important role in both the reduction of fuel loss due to pipeline failure or deliberate destruction and in locating leaks. The design needs to be tested to determine if the operational characteristics are satisfactory.

f. **Interface Detection.** Most pipeline operations will involve handling more than one type of fuel. Three fuels – motor gasoline, diesel fuel, and turbine fuel – comprise the majority of Military fuels used today. Aviation gasoline is still used but

[22] H. N. Johnston, *Potential Methods for Reduction of Product Loss in Military Pipelines*, Report 2034, U.S. Army Mobility Equipment Research and Development Center, Fort Belvoir, Virginia, August 1972.

in such small quantities that methods of shipment other than pipelines are more practical.

The process of pumping more than one fuel through a single pipeline is known as "batching." In some cases a rubber ball or other type of batch separator is inserted into the pipeline to segregate batches of fuel in the pipeline. More frequently, no separator is used and the commingling of the two products at the interface between batches is negligible. The most important factor in preventing excessive commingling between batches is "cutting" the pipeline throughput into the appropriate receiving tank.

The pipeline dispatcher is responsible for control of all injections of product into the pipeline. By knowing the time of injecting a new product, the flow rate, and pipe size, the dispatcher can compute the approximate time the interface will pass any point on the pipeline. However, variations in flow rate and line size, although slight, make it impossible to predict the time the interface will arrive with sufficient accuracy for the receiving terminal to cut the incoming product to a different tank. Meters can be used to provide an indication of the arrival of an interface; however, inaccuracies of less than one percent allow the potential for excessive commingling.

To solve this problem, a batch interface detector is used to detect the arrival of a product interface. These instruments monitor the specific gravity of the fuel in the pipeline with sufficient accuracy to detect the change when an interface passes the sensor. Upon sensing an abrupt change in specific gravity, a signal will actuate both audible and visual signals. As a safety precaution it is a good idea to draw a fuel sample from the pipeline periodically and confirm the fuel type by color and appearance. This visual check should be made just prior to the expected arrival of an interface and immediately upon receiving an interface signal from the batch interface detector to insure the interface is cut at the proper point to segregate the two products in the proper tanks.

Batch interface detectors conforming to Military Specification MIL-D-52840 (ME) are currently being procured. These units are satisfactory for all Military pipeline operations.

## IV. DISCUSSION

**10. Reliability Assessment.** Data available relating to the reliability of a complete pipeline system are virtually nonexistent. It is impossible to develop any standard model for a detailed reliability assessment since every pipeline system is unique. However, there are some interrelationships between certain components that exist in every pipeline system. This reliability assessment uses those interrelationships to estab-

lish the effect pump station reliability has on pipeline system reliability. Knowing the interdependence between pump station and pipeline system reliability, it is possible to draw some conclusions regarding the reliability and maintainability characteristics required for individual pump units.

a. **System Model.** The pipeline system reliability model is illustrated schematically in Figure 38. A marine terminal tank delivers fuel through a flood pump station to the first of three pipeline booster pump stations. The pipeline from the third booster-pump station discharges the fuel directly into tankage at a storage terminal. Because of the tremendous number of factors that impact on the performance of a pipeline system, the number of variables can easily become unmanageable. To preclude this condition, the following assumptions were made.

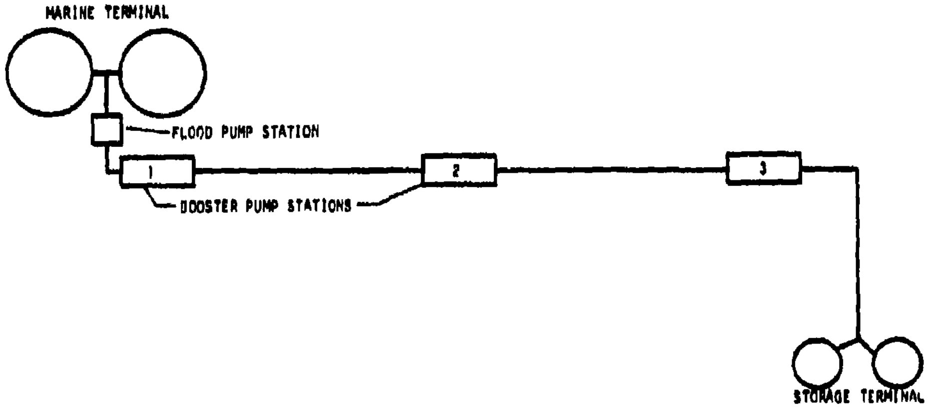



**Figure 38. Schematic of pipeline system reliability model.**

(1) The storage capacity and receipt capability of the marine terminal is sufficient to maintain a continuous supply of fuel to the pipeline so that empty tanks at the marine terminal will never prevent the pipeline from operating at the desired throughput rate.

(2) The flood pump station has the capability of continuous delivery of fuel to the number 1 booster station at the required suction pressure so that the pipeline will never be prevented from operating because of insufficient suction pressure.

(3) For any given simulation, all booster pumps will be identical, having the same performance characteristics. The relationship between pump station discharge pressure and flow rate shall be as shown in Figure 39. This curve represents typical pump performance for a double-suction centrifugal pump having a specific speed ($N_s$) = 1800.

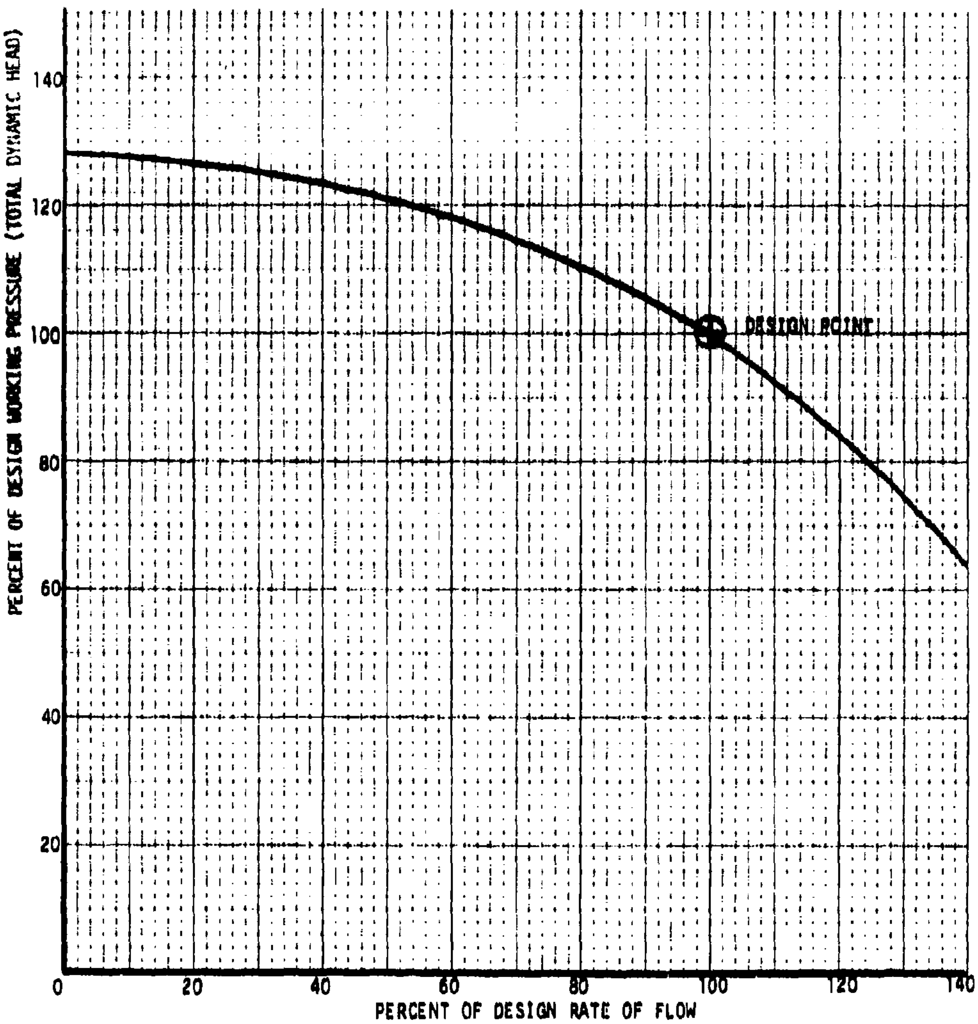



**Figure 39. Pump station performance characteristics for pipeline system reliability model.**

(4) Each pipeline booster pump station may include one or more booster pumps operating in series. However, all pump stations, when operating at design conditions, shall have the same number of pumps on-line (operating).

(5) Standby pumps may be used at the booster pump stations. The performance characteristics of the standby pumps shall be the same as all other booster pumps. The number of standby pumps at booster pump station 1 need not be the same as at stations 2 and 3.

(6) The flow loss characteristics of each pipeline segment (i.e., between station 1 and 2, between station 2 and 3, and between station 3 and the storage terminal) shall be the same.

(7) The reliability of the pipe is assumed to be 1 for the purposes of this analysis.

(8) All simulations shall begin with the storage terminal empty and run for a period of 90 days. Each "tank empty" event at the storage terminal, excluding at initiation of the simulation, shall constitute a mission failure.

The friction head, or loss of head, from a fluid flowing in a pipe can be calculated using the Darcy-Weisbach equation:

$$H_f = \frac{0.031\ fLQ^2}{d^5}$$

where

$H_f$ = head loss in feet.
$f$ = dimensionless friction factor.
$L$ = length of pipe in feet.
$Q$ = rate of flow in gallons per minute.
$d$ = inside diameter of pipe in inches.

The friction factor is a function of the roughness of the inside surface of the pipe and of Reynold's Number. The Reynold's Number is a dimensionless number which can be calculated using the equation:

$$R = \frac{3{,}160\ Q}{d\gamma}$$

where

$R$ = Reynold's Number.
$Q$ = rate of flow in gallons per minute.
$d$ = inside diameter of pipe in inches.
$\gamma$ = kinematic viscosity of the liquid in centistokes.

After computing the Reynold's Number, the friction factor is read from the general resistance diagram for uniform flow in conduits shown in Figure 40.

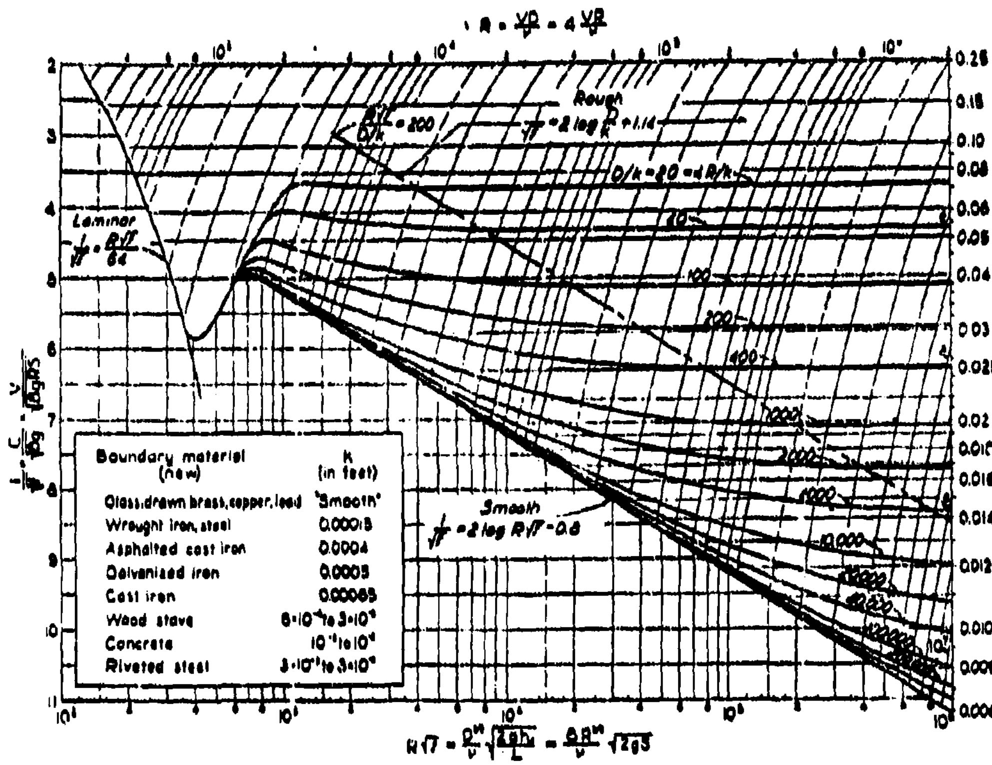



**Figure 40. General resistance diagram for uniform flow in conduits.**

For a given pipeline design, the values for L and d in the Darcy-Weisbach equation become constants. The change in friction factor, f, over the normal range of flow rates in pipelines is small. Thus, we can assume it to be a constant without introducing a large error in calculated values. The Darcy-Weisbach equation then becomes:

$$H_f = CQ^2$$

where:

$$C = \frac{0.031\ fL}{d^5}$$

If the loss of head, $H_f$, and rate of flow, Q, are expressed as a percent of their magnitudes at the pipeline design point, both variables will have a value of 100 percent at the design point. Substituting these values into the simplified equation and solving:

$$H_f = CQ^2$$

$$100 = C\,(100)^2$$

$$C = 0.01$$

Therefore:

$$H_f = 0.01\ Q^2$$

when: $H_f$ is expressed as a percent of the head loss at the design rate of flow.

Q is expressed as a percent of the design rate of flow.

This equation is plotted in Figure 41 with the pump performance characteristics from Figure 39. Intersection of the two curves at 100 percent of design rate of flow and design working pressure represents the design point. In order to evaluate the effects of changes in flow loss and pump performance characteristics, these two curves are used as the booster pump station design characteristics in analyzing all variation of the reliability model.

It was stipulated in the definition of the model, paragraph 10a(6), that the flow characteristics of each pipeline segment shall be the same. Therefore, the equation $H_f = 0.01\ Q^2$ represents the flow loss between any two adjacent pump stations. By definition then, the flow loss through any two adjacent pipeline segments, from station A to C or station B to D, would be twice the flow loss through a single segment.

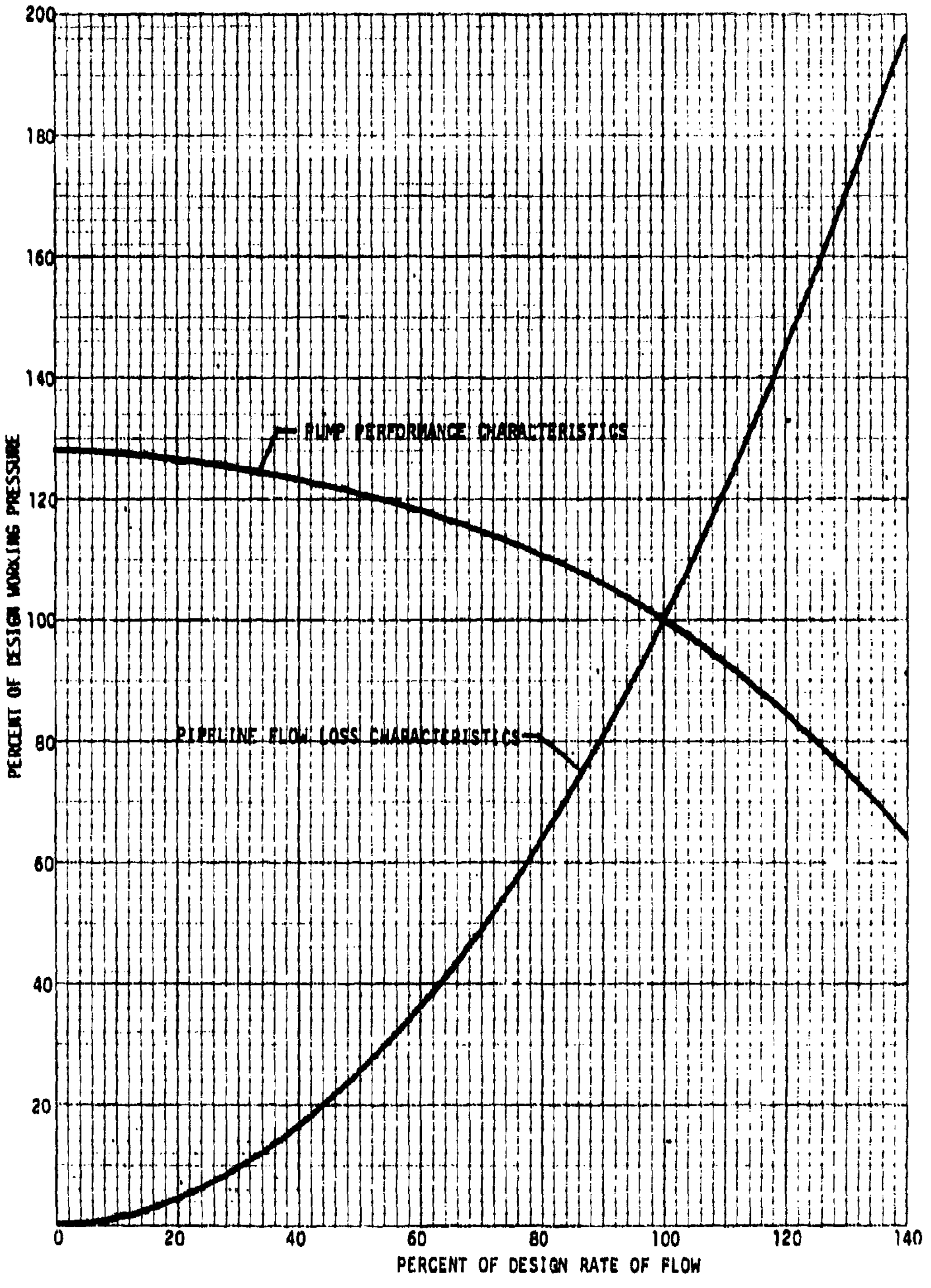



**Figure 41. Flow loss and pump performance for pipeline system reliability model.**

If $H_{2f}$ represents the flow loss through two adjacent pipeline segments, then:

$$H_{2f} = (2)(H_f) = (2)(0.01Q^2)$$
$$= 0.02Q^2$$

Following the same rationale, the flow loss through all three segments of the pipeline in Figure 38, will be equal to three times the loss through one pipeline segment. If $H_{3f}$ is the loss through three pipeline segments, then:

$$H_{3f} = (3)(H_f) = (3)(0.01Q^2)$$
$$= 0.03Q^2$$

The equations for $H_f$, $H_{2f}$ and $H_{3f}$ are plotted in Figures 42 and 43 as curves $L_1$, $L_2$, and $L_3$, respectively. In Figure 42, curve D represents the pump station performance characteristics from Figure 39. Since the pump stations in the model are in series, the combined discharge head from two pump stations would be twice that for a single station. Curve H in Figure 42 represents working pressure values that are twice the working pressure values of curve D indicating the combined discharge head from two pump stations. Using curves D and H in conjunction with the pipeline flow loss curves $L_1$, $L_2$, and $L_3$, it is possible to determine the flow characteristics of the reliability model pipeline system for all possible operating conditions when each booster pump station includes only one pump unit.

Under normal operating conditions the pump unit at each pump station would provide the pressure to overcome the pressure loss in each respective pipeline segment. Thus, the design point at the intersection of curves D and $L_1$ indicates operation at 100 percent of design rate of flow and design pressure. Note that curve H, the combined head for two pump units, intersects curve $L_2$, the combined loss through two pipeline segments, at 100 percent of the design rate of flow. This condition is valid only when all pumps and pipeline segments have the same flow characteristic as specified in the model definition.

The normal hydraulic gradients for the pipeline system reliability model are shown in Figure 44. Those gradients correspond to the intersections of curves D and $L_1$ in Figure 42. With only one pump at each booster station, a pump failure completely eliminates the pumping capability at that station. If the failure occurs at station 1, the entire pipeline is inoperative since there is no pressure to push the fuel through the pipeline segment from station 1 to station 2. This is an important consideration in determining the need for standby or backup units.

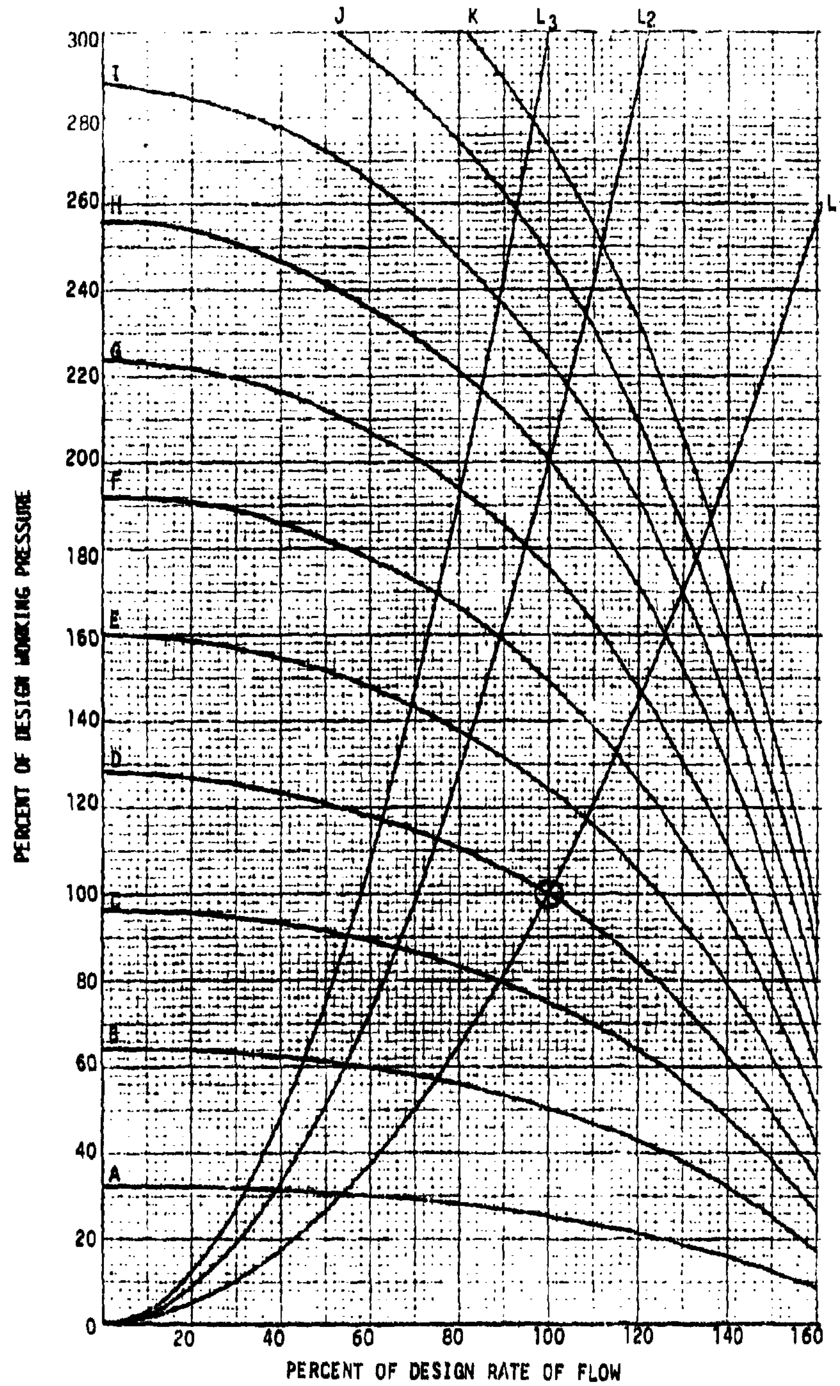



**Figure 42.** Pipeline flow characteristics for reliability model using one, two, or four pumps per booster station.

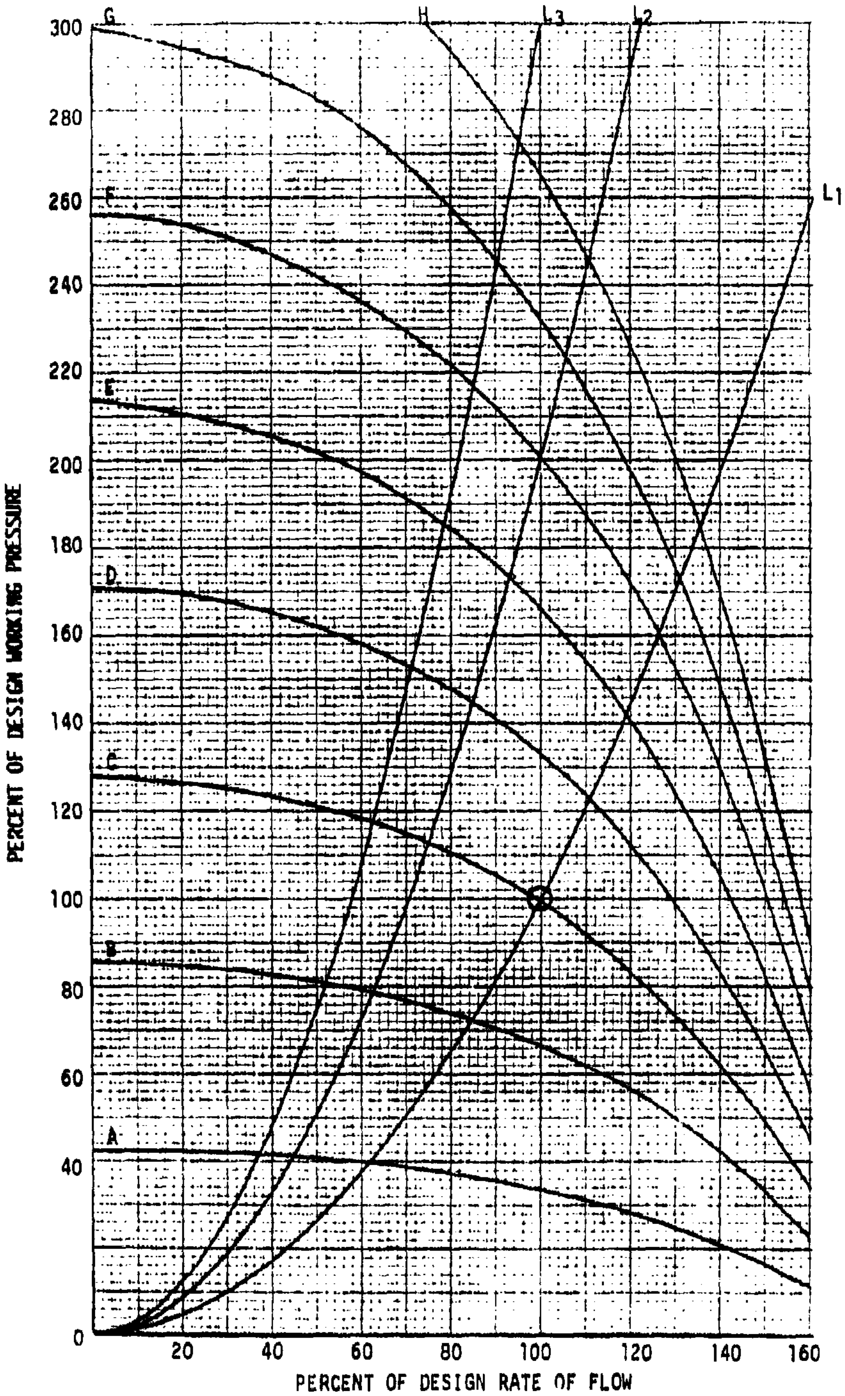



**Figure 43. Pipeline flow characteristics for reliability model using one or three pumps per booster station.**

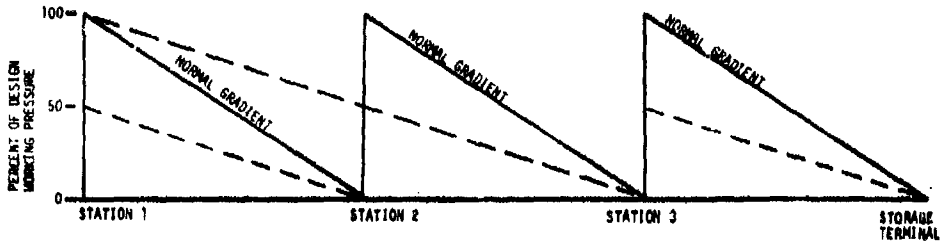



**Figure 44. Hydraulic gradients for reliability model pipeline system.**

Failure of the pump at booster station 2 eliminates the pressure increase at that point. Booster station 1 must then push the fuel from station 1 to station 3, causing the hydraulic gradient in Figure 44 to shift to the dotted gradient line between station 1 and station 3. To maintain adequate suction pressure, booster station 3 must reduce its pumping rate to obtain a hydraulic gradient having the same slope, shown as a dotted line between station 3 and the storage terminal. If the pump at station 1 continues to operate at the same speed, the new flow rate will be 75 percent of the design rate of flow, indicated by the intersection of curves D and $L_2$ in Figure 42. As the flow conditions move from curve $L_1$ to $L_2$ on curve D, the pressure increases from 100 to 113 percent of the design working pressure. If the pipeline will not withstand this increase in operating pressure it will be necessary to reduce the pump speed at station 1 until the pump performance curve, D, intersects the curve $L_2$ at 100 percent of the design working pressure, corresponding to 71 percent of the design rate of flow This study assumes the pipeline design working pressure is based on a safety factor sufficient to allow all pumps to operate at the increased pressure resulting from operating at the intersection of the normal performance curve with flow loss curves $L_2$ and $L_3$.

The preceeding procedure used to determine the throughput rate in the event of a failure at pump station 2 can be used to determine the flow conditions for any possible combination of pumps operating. Table 14 shows the intersections of curves in Figure 42 corresponding to the various combinations of pumps that can be operating in the reliability model pipeline system when each pump station consists of a single pump.

Table 14. Interaction of Flow Loss and Pump Performance Curves for Single-Pump Booster Stations

| Number of Booster Pumps Operational | | | Intersection Figure 42 |
|---|---|---|---|
| Station 1* | Station 2 | Station 3 | |
| 1 | 1 | 1 | D-$L_1$ |
| 1 | – | 1 | D-$L_2$ |
| 1 | 1 | – | H-$L_3$ |
| 1 | – | – | D-$L_3$ |

* The system is inoperable unless the pump at booster station 1 is operational.

Curve B, Figure 42, represents the pump performance characteristics when two pumps in series are used at each station to obtain the design rate of flow and working pressure. Curve D is a multiple of two of curve B; that is, for any rate of flow the working pressure on curve D is twice that on curve B. Curve F is a multiple of three of curve B reflecting the characteristics of three pumps in series. Curves H and J are multiples of four and five, respectively, of curve B. Using two pumps per station in the reliability model pipeline system, design conditions would be indicated by the intersection of curves D and $L_1$, in Figure 42 and the normal gradients in Figure 44. Failure of one pump at booster station 1 would result in the operating conditions moving down curve $L_1$ in Figure 42 to the intersection with curve B. This corresponds to the dotted hydraulic gradient line between stations 1 and 2 in Figure 44. To maintain adequate suction pressure at stations 2 and 3 after the failure occurs at station 1, the pumping rate at stations 2 and 3 must be reduced to yield the same hydraulic gradients for their respective pipeline segments. This can be accomplished at stations 2 and 3 by reducing the speed of all pumps or shutting down one pump at each station.

The flow rate resulting from any possible combination of pumps operating with two pumps at each station can be determined from the intersections of curves B, D, F, H, and J with curves $L_1$, $L_2$ and $L_3$ in Figure 42. The maximum possible flow rate for any situation will be the percent of design rate of flow corresponding to the intersection of the two curves farthest to the right that will yield hydraulic gradients having the same slopes for all sections of the pipeline. The curve intersections indicating the rate of flow for all possible situations with two pumps at each booster station are listed in Table 15.

Figure 43 contains pump performance and flow loss curves for the reliability model pipeline system when three pumps in series are used at each station to develop the design working pressure. Table 16 lists the appropriate curve intersections for the flow rates under all potential operational conditions with three pumps per station. Figure 42 includes the pump performance curves for determining operating conditions when four pumps are used at each booster station to develop the design working pressure.

b. **Elements of Analysis.** The reliability model is constructed to analyze the interrelationships between the following parameters and project their affect on the mission reliability of the pipeline system.

(1) **Pump Station Configuration.** The pump station configuration defines the number of pumps operating at each pump station when the pipeline is operating at design conditions. In addition, the number of reserve or standby units at each pump station must be specified.

Table 15. Interaction of Flow Loss and Pump Performance Curves for Two-Pump Booster Stations

| Number of Booster Pumps Operational | | | Intersection Figure 42 |
|---|---|---|---|
| Station 1* | Station 2 | Station 3 | |
| 2 | 2 | 2 | D-$L_1$ |
| 2 | 2 | 1 | J-$L_3$ |
| 2 | 2 | – | H-$L_3$ |
| 2 | 1 | 2 | F-$L_3$ |
| 2 | 1 | 1 | H-$L_3$ |
| 2 | 1 | – | F-$L_3$ |
| 2 | – | 2 | D-$L_2$ |
| 2 | – | 1 | D-$L_2$ |
| 2 | – | – | D-$L_3$ |
| 1 | 2 | 2 | B-$L_1$ |
| 1 | 2 | 1 | B-$L_1$ |
| 1 | 2 | – | D-$L_2$ |
| 1 | 1 | 2 | B-$L_1$ |
| 1 | 1 | 1 | B-$L_1$ |
| 1 | 1 | – | D-$L_3$ |
| 1 | – | 2 | B-$L_2$ |
| 1 | – | 1 | B-$L_2$ |
| 1 | – | – | B-$L_3$ |

* At least one pump must be operational at booster station 1 for the system to operate.

(2) **Mean Time Between Failure (MTBF).** For each simulation, the MTBF for the booster pumps is specified. The simulation model uses the specified MTBF to determine, on a random basis, the time pump failures occur based on an exponential distribution.

(3) **Mean Time to Repair (MTTR).** The MTTR for the pumps is specified for each configuration as a triangular distribution with the minimum, most likely, and maximum repair time stated in hours. The MTTR includes the total time a pump unit is inoperative due to a failure and, therefore, includes active repair time and administrative down time. Administrative down time encompasses the time required for maintenance personnel to travel to the pump station site after being notified of a pump failure and time spent waiting to receive repair parts, as well as any other down time not spent actually repairing the failure.

Table 16. Interaction of Flow Loss and Pump Performance Curves for Three-Pump Booster Stations

| Number of Booster Pumps Operational | | | Intersection Figure 43 |
|---|---|---|---|
| Station 1* | Station 2 | Station 3 | |
| 3 | 3 | 3 | C-$L_1$ |
| 3 | 3 | 2 | H-$L_3$ |
| 3 | 3 | 1 | G-$L_3$ |
| 3 | 3 | 0 | F-$L_3$ |
| 3 | 2 | 3 | E-$L_2$ |
| 3 | 2 | 2 | G-$L_3$ |
| 3 | 2 | 1 | F-$L_3$ |
| 3 | 2 | 0 | E-$L_3$ |
| 3 | 1 | 3 | D-$L_2$ |
| 3 | 1 | 2 | D-$L_2$ |
| 3 | 1 | 1 | E-$L_3$ |
| 3 | 1 | – | D-$L_3$ |
| 3 | – | 3 | C-$L_2$ |
| 3 | – | 2 | C-$L_2$ |
| 3 | – | 1 | D-$L_3$ |
| 3 | – | – | C-$L_3$ |
| 2 | 3 | 3 | B-$L_1$ |
| 2 | 3 | 2 | G-$L_3$ |
| 2 | 3 | 1 | F-$L_3$ |
| 2 | 3 | – | E-$L_3$ |
| 2 | 2 | 3 | D-$L_2$ |
| 2 | 2 | 2 | B-$L_1$ |
| 2 | 2 | 1 | E-$L_3$ |
| 2 | 2 | – | D-$L_3$ |
| 2 | 1 | 3 | C-$L_2$ |
| 2 | 1 | 2 | C-$L_2$ |
| 2 | 1 | 1 | D-$L_3$ |
| 2 | 1 | – | C-$L_3$ |
| 2 | – | 3 | B-$L_2$ |
| 2 | – | 2 | B-$L_2$ |
| 2 | – | 1 | B-$L_2$ |
| 2 | – | – | B-$L_3$ |
| 1 | 3 | 3 | A-$L_1$ |
| 1 | 3 | 2 | A-$L_1$ |
| 1 | 3 | 1 | A-$L_1$ |
| 1 | 3 | – | A-$L_1$ |
| 1 | 2 | 3 | A-$L_1$ |

Table 16. Interaction of Flow Loss and Pump Performance Curves for Three-Pump Booster Stations (Cont'd)

| Number of Booster Pumps Operational | | | Intersection Figure 43 |
|---|---|---|---|
| Station 1* | Station 2 | Station 3 | |
| 1 | 2 | 2 | A-$L_1$ |
| 1 | 2 | 1 | A-$L_1$ |
| 1 | 2 | – | A-$L_1$ |
| 1 | 1 | 3 | A-$L_1$ |
| 1 | 1 | 2 | A-$L_1$ |
| 1 | 1 | 1 | A-$L_1$ |
| 1 | 1 | – | B-$L_3$ |
| 1 | – | 3 | A-$L_2$ |
| 1 | – | 2 | A-$L_2$ |
| 1 | – | 1 | A-$L_2$ |
| 1 | – | – | A-$L_3$ |

* At least one pump must be operational at booster station 1 for the system to be operable.

(4) **Consumption Versus Throughput Ratio.** The average daily consumption of the forces being supplied by the pipeline is specified as a ratio in relation to the daily throughput rate of the pipeline when operating at design conditions. For example, if a pipeline is designed to deliver 30,000 barrels of fuel per day and the average daily consumption rate is 27,000 barrels of fuel, the consumption versus throughput ratio is (27,000/30,000) = 0.9. Since this ratio is a dimensionless number, the analysis remains independent of specific flow rates.

(5) **Reserve Storage.** The total capacity of the fuel storage tanks at the storage terminal, Figure 38, constitutes the reserve storage. For each simulation the amount of reserve storage available is specified as a multiple of the average daily consumption rate.

(6) **Restart Point.** The pipeline design throughput rate is always in excess of the average daily consumption. Therefore, barring excessive system failures, the storage terminal will ultimately be filled to capacity. The model shuts the pipeline down at this time. The pipeline remains in a shutdown condition until consumption reduces the fuel on hand to a predetermined level. At this "restart point," the pipeline resumes delivery of fuel from the marine terminal to the storage terminal. Thus, the restart point is stated as a percentage of the reserve storage capacity.

c. **Mission Reliability.** The mission of the pipeline is to maintain a supply of fuel at the storage terminal adequate to satisfy consumption demands. Thus, as stated in paragraph 10a(8), a mission failure occurs when the reserve storage tanks are drawn down to an empty condition. This event may result from either of two conditions. The most critical situation would result from a complete failure of the pipeline so that no fuel is available at the storage terminal. A less severe situation occurs when pump failures have reduced the pipeline throughput rate below the consumption rate. Under this condition, the pipeline would be continuing to deliver some fuel but at a rate insufficient to satisfy total demand. In either case, a tank-empty event is considered a mission failure since both instances would require some curtailment of activities.

d. **Simulation Results.** Consider first simulation results based on the following conditions.

(1) **Pump Station Configuration.** One pump at each pump station with a standby unit at station 1.

(2) **MTBF.** Simulations to be run for MTBF values of 150, 300, 450, and 600 hours.

(3) **MTTR.** Triangular distribution having a minimum value of 9 hours, a most likely value of 12 hours, and a maximum value of 18 hours.

(4) **Consumption Versus Throughput Ratio.** The average daily consumption is 0.9 times the pipeline design throughput rate.

(5) **Reserve Storage.** The total available storage capacity at the storage terminal is equal to three days consumption.

(6) **Restart Point.** Pipeline operation is resumed when the total fuel on hand is drawn down to 90 percent of the reserve storage capacity.

The conditions listed in paragraphs 10a(1) through (6) actually represent four sets of operational conditions because of the four separate MTBF values given in paragraph 10a(2). Since the MTBF and MTTR values for each pump failure are selected on a random basis, the actual system reliability value determined during one simulation run will be a function of those particular values selected at random by the computer model for that simulation run. Because these variations in MTBF and MTTR values will cause a significant variation in the system reliability, it is necessary to run a series of simulations to get convergence of the results toward the actual system reliability that can be expected. Each of the four data points plotted to develop the

upper curve in Figure 45 represent an average of the actual values obtained from 50 simulation runs. It is readily apparent from this curve, that increasing pump MTBF increases mission reliability rapidly up to about 200 hours. The increase in mission reliability continues until the pump MTBF reaches approximately 500 hours at a mission reliability 0.96. Beyond that point very little improvement in system mission reliability can be achieved by increasing the MTBF for pump units. The center curve in Figure 45 represents the results of simulation runs using the conditions stated above, except the minimum, most likely, and maximum MTTR values are increased to 36, 48, and 72 hours, respectively. Similarly, the lower curve in Figure 45 represents results from the simulation model developed using the same operating conditions but minimum, most likely, and maximum MTTR values of 72, 96, and 144 hours, respectively. From Figure 45, it is readily apparent that MTTR is a significant factor in determining mission reliability for a pipeline system.

Continuing the reliability analysis process, the system designed characteristics stated in paragraphs 10a(1) through (6) are used with the exception that the number of pumps at each pump station is changed to include two pumps at each booster station with a standby unit at pump station 1. Mission reliability results obtained from this pipeline configuration are plotted in Figure 46. The top curve represents the mission reliability obtained using an MTTR distribution having minimum, most likely, and maximum values of 9, 12, and 18 hours, respectively. The two lower curves represent the system reliability that will result from this pipeline configuration with increases in the MTTR values. A comparison of Figures 45 and 46 will show that for low MTBF and MTTR values, mission reliability for a system using one pump per station with a standby at station 1 exceeds mission reliability for a similar system using two pumps per station with a standby unit at station 1. Further examination of the two figures shows that the mission reliability for the two-pump-per-station configuration exceeds the single-pump-per-station configuration for all conditions when the MTBF is greater than 350 hours.

The effects of changes in pump station configurations are more readily discernible in Figure 47. The three curves in Figure 47 represent mission reliability values obtained using the system configuration identified in paragraphs 10a(1) through (6) with changes in number of pumps at each station. The top two curves are the top curves from Figures 45 and 46 based on the MTTR values of 9, 12, and 18 hours. The lower curve reflects what happens when the standby unit is eliminated from the first pump station of a system having only one pump at each booster station. In a pump station of this configuration, the total pipeline system is rendered inoperative with the failure of the pump unit at the first station.

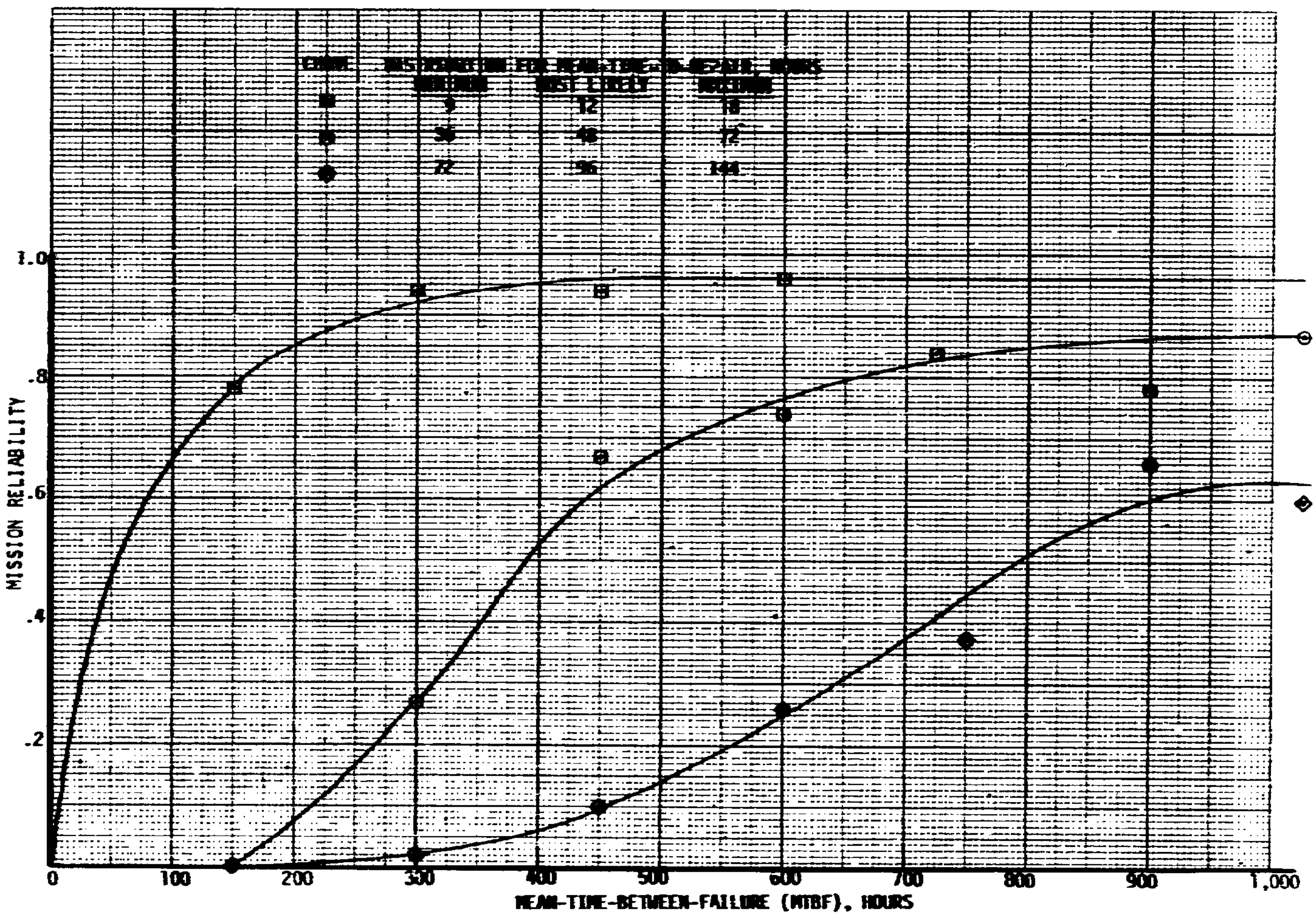



Figure 45. Reliability model simulation results using one pump at each booster station with standby unit at Station 1.

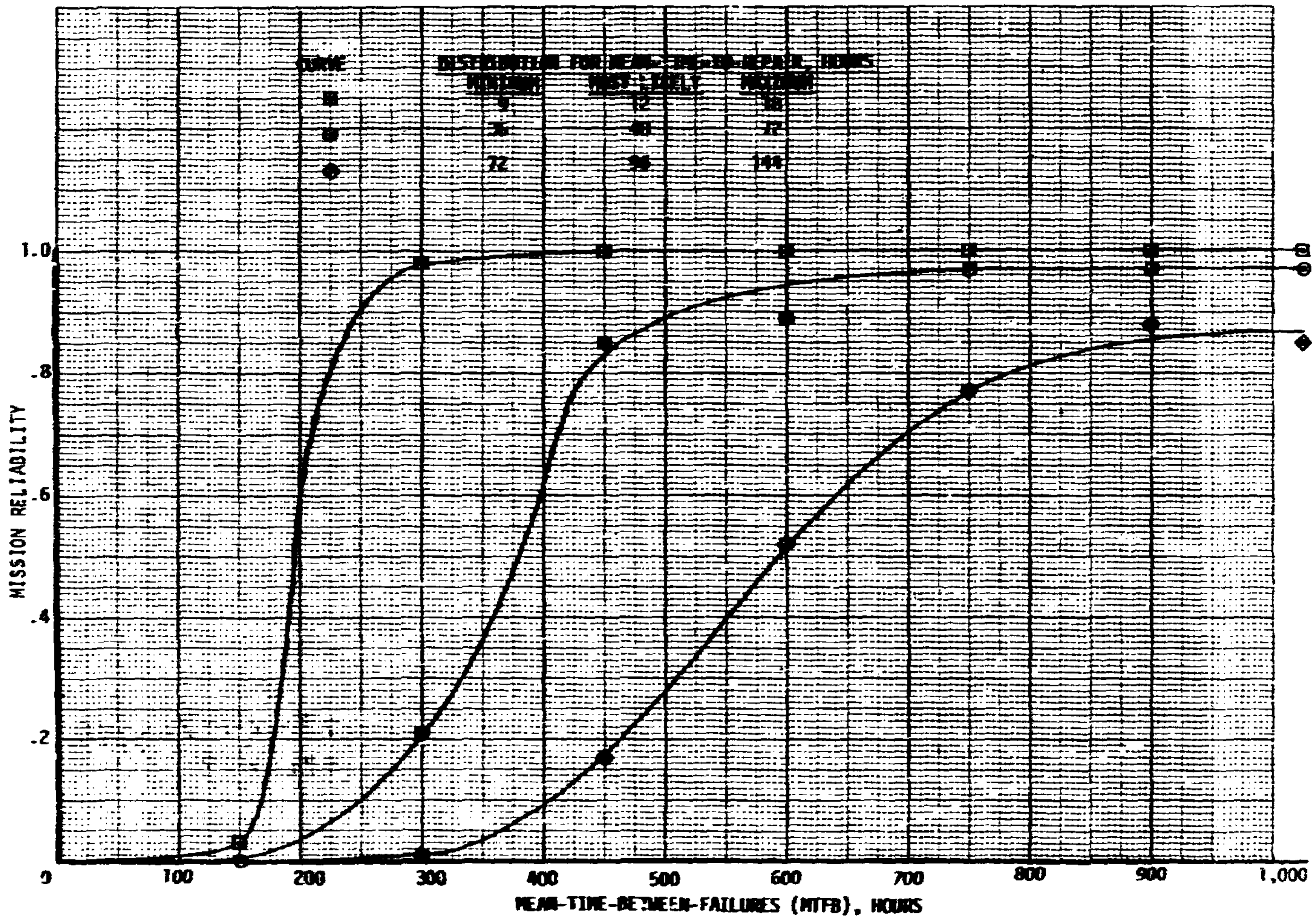



Figure 46. Reliability model simulation results using two pumps at each booster station with standby unit at Station 1.

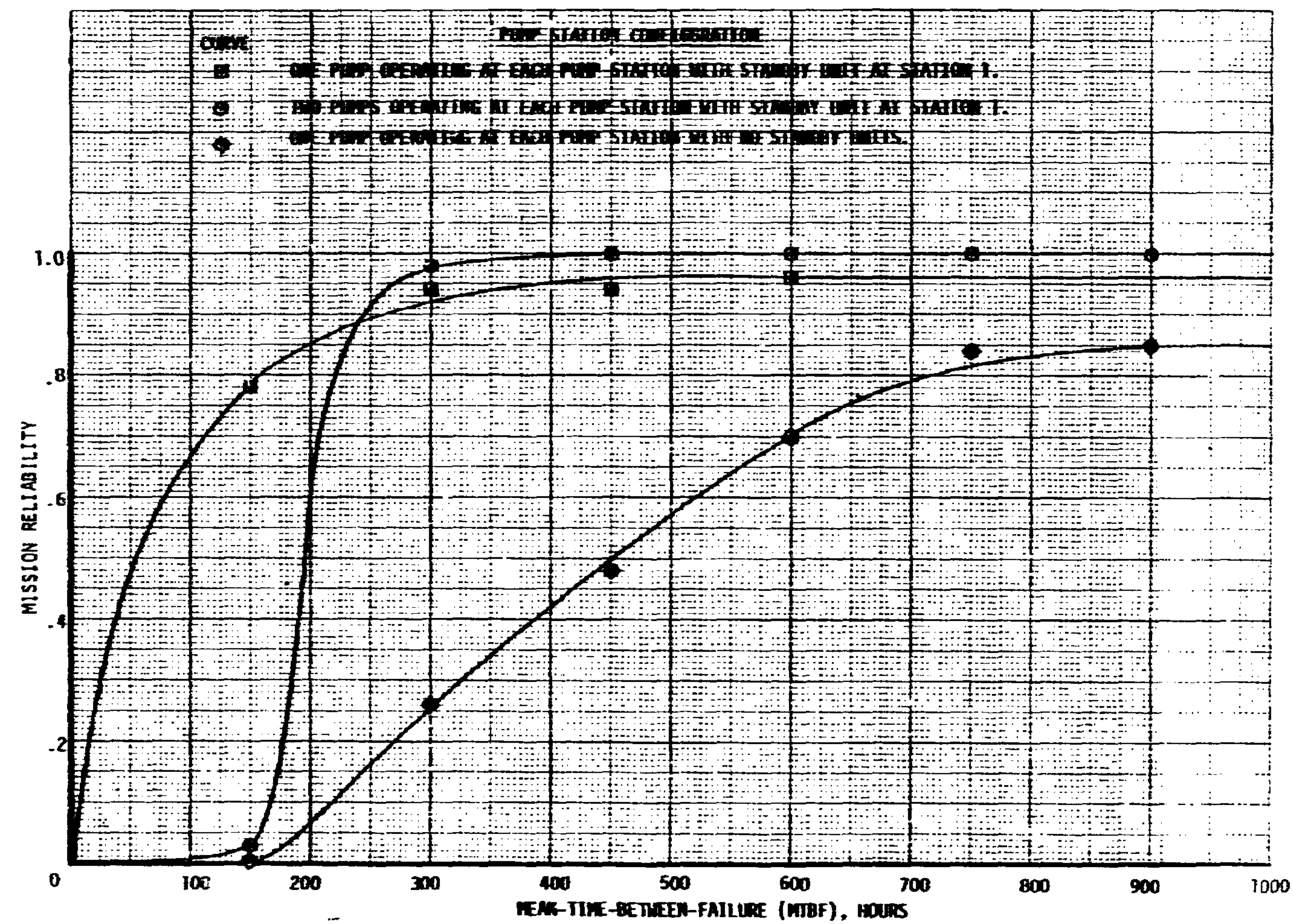



Figure 47. Comparison of reliability model mission reliability results using variations in pump station configuration.

Variations in the pipeline system reliability model discussed in the previous paragraph represent only a few of the many possible variations in pipeline system design and operating conditions. Thus, a reliability analysis which will pinpoint the most desirable pipeline system design and operating conditions becomes a complex study in itself. The results presented in the preceding paragraphs are intended to provide an indication that the preferred pipeline system design will include at least two pumps in each booster station with a standby unit at pump station 1. Because of the complexity of the pipeline system reliability analysis, the complete results of such a study are not included herein. The reliability analysis work is continuing at MERADCOM and the results will be published later.

As noted earlier, data relating to the reliability of pipeline systems are virtually nonexistent. The primary factor contributing to the reliability of the pipeline system is considered to be the reliability of the engines used to drive booster pumps. Because pipeline pump reliability data were not available, it was necessary to obtain reliability data on engines from other sources. Figure 48 shows the relationship between engine horsepower and Mean Time Between Failure, in hours, for diesel and gas turbine engines.

The MTBF curve for diesel engines is based on data collected during more than 100,000 hours of reliability testing on 28 diesel engine generators at MERADCOM. The engines on these generator sets were rated from 36 to 340 brake horsepower. The test records were analyzed to determine which failures were related specifically to the engine. The curve for diesel engines in Figure 48 shows the relationship between horsepower and MTBF reflected by these data. Contrary to a commonly held belief, the MTBF for diesel engines decreases with increase in size.

A comparative analysis of gas-turbine and diesel engines conducted by the Naval Ship Systems Command[23] identifies a correlation between the reliability characteristics for the two types of engines. Using that relationship and the data on diesel engine generators, it is possible to develop a relationship between horsepower and MTBF for turbine engines. That relationship is represented by the turbine engine curve in Figure 48.

Comparison of the curves in Figure 48 with Figures 45 and 46 shows the reliability of the existing gas-turbine engine to be adequate to meet pipeline pump requirements throughout the horsepower range shown providing the MTTR can be held to low average values. From previous analysis it was determined that the brake horsepower rating of diesel engines when derated for altitude and temperature, must be limited to slightly more than 300 brake horsepower. On this basis we can expect the

[23] *Comparative Analysis of Selected Gas Turbine and Diesel Engines*, Report No. 1080, Naval Ship Systems Command, Washington, D.C., April 1969.

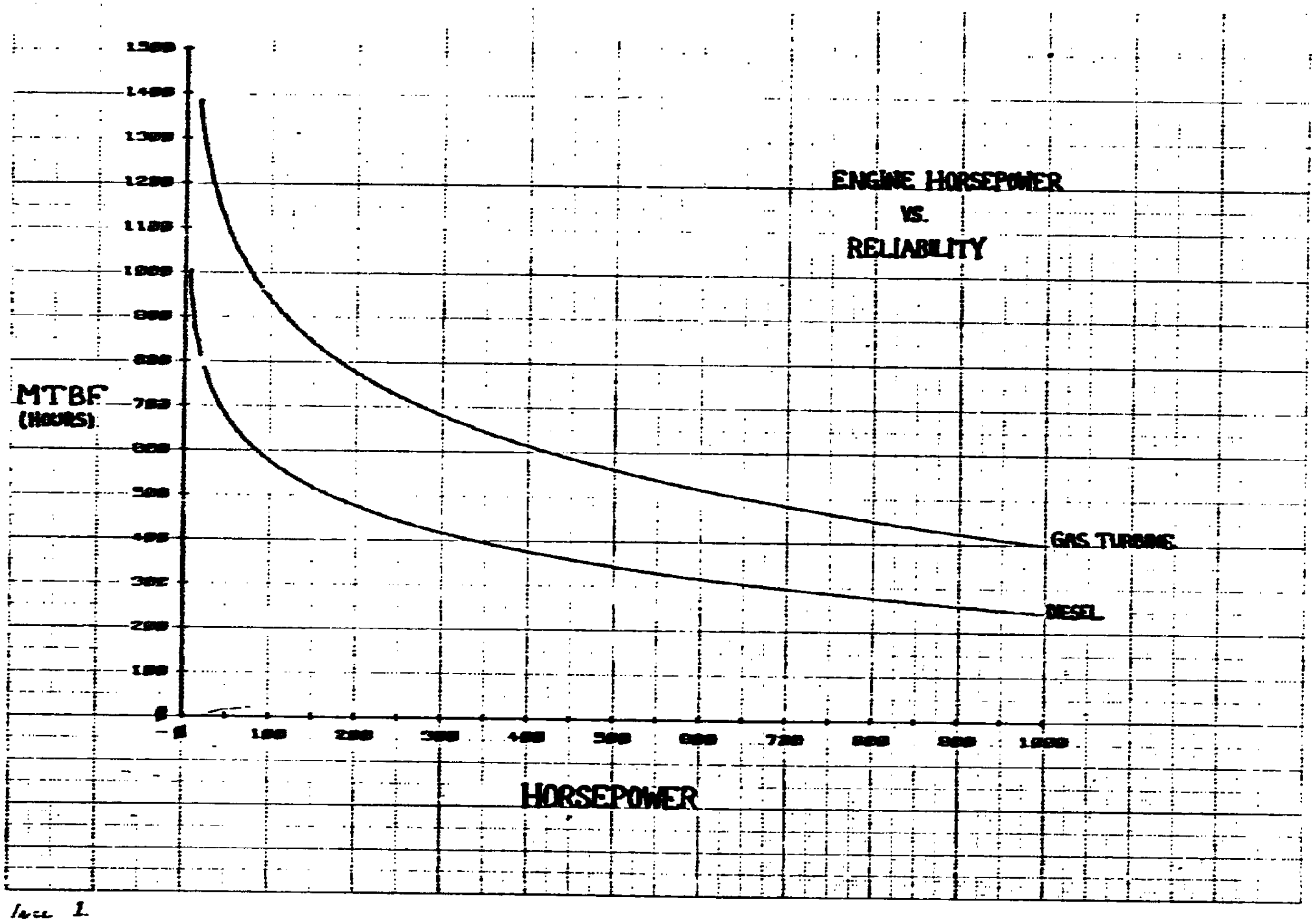



Figure 48. Engine horsepower vs. reliability for gas-turbine and diesel engines.

MTBF for diesel-engine-driven pumps to be in the range of 400 hours. Again, examining Figures 45 and 46, it is found that diesel-engine-driven pumps of this size will meet pipeline system reliability requirements provided that MTTR is again held to low values. The data presented in Figures 45 and 46 indicate clearly that achieving acceptable pipeline system reliability, is dependent on the ability to repair pump failures in a minimum amount of time.

Table 17 presents maintenance data extracted from the Navy study on diesel and turbine engines. Analysis of these data shows an MTTR of 12.2 hours for gas-turbine engines and 9.0 hours for diesel engines. Precautions must be exercised in comparing these repair times to those for equipment operating in remote environments, such as at pipeline booster pumping stations. Two factors contribute to the fact that the Navy repair time will probably be substantially less than those encountered by the Army in pipeline systems operations. The Navy data are based on equipment mounted on ships where personnel, facilities, and repair parts are available immediately. Second, because the Navy personnel live with this equipment over extended periods of time performing daily maintenance, they are inherently more familiar with the maintenance requirements.

Table 17. Comparison of Gas-Turbine and Diesel Engine Reliability and Maintainability Characteristics

| Characteristics | Turbine Engine | Diesel Engine |
|---|---|---|
| Total Maintenance Actions | 1071 | 1224 |
| Total Maintenance Manhours | 8340 | 4980 |
| Preventative Maintenance Actions | 395 | 666 |
| Preventative Maintenance Manhours | 2008 | 834 |
| Corrective Maintenance Actions | 382 | 372 |
| Corrective Maintenance Manhours | 4670 | 3389 |

This factor focuses on a critical issue associated with a selection of pipeline pump engines. If pipeline pump failures cannot be repaired and the pumps returned to service in a few hours, the system reliability will not be acceptable. At best, the data in Table 17 are indicative of the minimum active maintenance hours which can be expected to be required to support gas-turbine and diesel engines. The fact that the MTTR for diesel engines is less than for gas turbines tends to favor diesel engines for pipeline pump applications. However, it is anticipated that in the environment where pipeline pumps will be operating, administrative downtime will probably exceed active maintenance time. To minimize the logistical support requirements for pipeline booster pumps, it is desirable to select pipeline pump engines that are used in other high-density items of Military equipment. Since pipeline pumps are typically

low-density items, the logistical support is often difficult. Selection of high-density engines from other applications will provide a logistical support system that is well established. Approach to design of pipeline pumps can go far in minimizing the integrated logistical support requirement for pipeline systems.

11. **Technological Risk.** The pipeline industry has established a broad technological base through many years of proven experience. The high initial investment costs for pipeline construction and the continuing high cost of pipeline operation and maintenance provide a constant incentive for advances in technology which will improve the cost effectiveness of pipeline operations. Because the initial investment costs are high there are high economic risks associated with the introduction of any new technology that may affect the serviceability of the pipeline system being installed. For this reason, advances in pipeline technology tend to be highly developed and proven by a number of trial cases before gaining wide acceptance by the pipeline industry.

The reader should not construe the foregoing comments to indicate a lack of emphasis on technological progress within the pipeline industry. To the contrary, extensive research and development programs covering many different facets of pipelining are conducted and/or sponsored by individual companies and trade associations. Because of these programs, a continual evolution of pipeline technology can be expected. Because of the nature of the pipeline industry, however, the prospects for radical advances in pipeline technology which would render existing technology obsolete in the near term are highly improbable. On this basis, a Military pipeline system developed using current and emerging technology available from the private sector is not subject to becoming obsolete within the foreseeable future.

Viewed strictly from the standpoint of available technology, the technological risks associated with Military pipeline construction appear extremely low. However, factors which have little, if any, influence in the private sector become important considerations in Military pipeline construction. The need for construction of a new commercial pipeline can be anticipated well in advance of the time the facility must be operational. Extensive planning and economic analysis precede the actual design and construction. Every pipeline is unique, designed to satisfy a well-defined set of operational requirements. No attempt is made to standardize on one design to satisfy a variety of requirements. Economic factors are the driving forces influencing the selection of pipe materials, pumping equipment, operating pressures, methods of construction, etc. There is little incentive for rapid rates of construction unless a cost advantage can be realized. The necessary personnel and skills are available within the civilian labor force.

The need for rapid construction rates, to minimize manpower and skill requirements, to use a common design to satisfy all potential operational requirements, and to react quickly to needs at any location in the world restricts the direct application of commercial pipeline technology to Military pipeline construction. Inability of existing pipe-joining techniques to provide the desired rates of construction and problems relating to extreme environmental conditions appear as the greatest barriers to be overcome in developing a pipeline system which is totally responsive to Military operational requirements. Technology emerging from industrial development programs cannot be expected to provide solutions to these problems. Thus, if these problems are to be solved, it will be incumbent upon the Army to carry out the necessary research and development.

The survey of the pipeline industry conducted by Value Engineering Company for MERADCOM did not identify a pipeline installation technique which would allow one construction crew to lay 30 kilometers of pipeline per day. However, at least two of the pipe-joining techniques evaluated are amenable to automation such that a properly designed pipe-laying machine could possibly achieve the desired pipe-laying rate. Foreign intelligence reports indicate the Soviet military forces currently have such a machine. Based solely on a technical assessment, developing such a machine presents limited risks. The question remaining to be resolved is whether the need for rapid rates of pipeline construction and to minimize manpower requirements is sufficient to justify the cost.

Construction of the trans-Alaska pipeline has demonstrated that the problems inherent to pipeline construction in extreme environments can be overcome. However, the Alyeska brute-force solution of employing a massive amount of personnel and equipment at a tremendous cost is not an acceptable approach for Military pipeline construction. The exorbitant costs cannot be justified to support a conflict of probable short duration. More important, personnel and equipment in the numbers needed could not be amassed in the available response time.

The trans-Alaska pipeline project is an extremely ambitious undertaking far outstripping any Military pipeline construction requirement that can be envisioned. A realistic look at the problems associated with pipeline construction under extreme cold conditions indicates the inherent problems are amplified as the size of the construction effort grows. Thus, a smaller military pipeline construction effort should require less complex solutions than those employed by the Alyeska Pipe Line Co. At the same time, some problems, such as protection of personnel and equipment from the extremes of the environment, will not change and many of the same solutions can be applied. In terms of the ability to meet future Military requirements, the primary significance of the current trans-Alaskan pipeline construction effort is not how the task is being accomplished, but that, given sufficient resources, the hostile arctic environment can be circumvented.

Solutions to existing Military pipeline construction requirements are achievable through the application and adaptation of currently available technology. Requirements peculiar to the Military mode of operation do not present any impenetrable technical barriers. Development of a pipeline system providing greater cost and operational effectiveness than the existing Military standard, lightweight steel, coupled pipelines appears to be possible with virtually no risk. The capability to install pipelines at rates up to 30 kilometers per day is also attainable and presents few technological problems. The number of lay crews required to achieve this rate of construction will be affected by the construction technique selected. Satisfying the extreme environmental requirements will be more difficult and involves a significant level of risk.

The crucial element in the development of an improved Military pipeline system appears to be identification of the appropriate pipe material and the proper pipe-joining technique. Once these vital issues have been resolved, development of the proposed pipeline system can proceed to a successful conclusion with only minor technical problems to be resolved. It is anticipated that development of the special support equipment necessary to achieve the rapid installation rate will require a more comprehensive development program than the pipeline system itself and, consequently, a longer development period. This should not be cause for delaying the development of the pipeline system using an interim, less rapid, means of installation if there is an immediate need for an improved pipeline system.

Satisfying the requirements for pipeline construction under extreme environments will be the most costly, requiring the greatest amount of effort and time and presenting the greatest risk. As with the rapid-laying capability, this effort can also lag the pipeline development effort if appropriate consideration is given to environmental requirements during the pipeline system development effort. In establishing development requirements, it must be clearly understood that extreme climatic conditions are going to result in reduced performance characteristics and impose greater personnel and support equipment requirements. Failure to recognize these critical issues may preclude achieving the established goals. Attempting to satisfy operational requirements in excess of the absolute minimum acceptable performance levels will unnecessarily increase development time and costs.

12. **Synthesis of Candidate Systems.** The general mission requirements defined in paragraph 5 for Scenarios I and II are used as the basis for comparing alternative pipeline system concepts. Outlined in the following paragraphs are basic system design characteristics for each alternative system concept and scenario. In addition, similar data are presented for an 8-inch-diameter pipeline constructed using Military standard, lightweight steel, coupled pipe.

Alternatives I through IV are 8-inch-diameter pipelines using the four pipe concepts selected by Value Engineering Company for detailed analysis during their investigation of pipe materials and joining techniques. Alternative V employs the same basic concept as Alternative III except a thicker wall pipe is used to allow a higher working pressure. Alternative VI is a variation of Alternative IV considering thinner wall pipe to reduce the level of effort required for system installation.

Alternatives VII and VIII are 6-inch-diameter pipelines using the same basic system concepts as Alternatives III and IV, respectively. The fuel delivery requirements for both scenarios will require parallel 6-inch pipelines. However, because of the low initial consumption rate in Scenario I, a single 6-inch pipeline can satisfy delivery requirements through the first 19 days. Thus, the level of construction effort required to establish an initial operational capability is significantly reduced.

Because of the emphasis placed on rapid pipeline construction by the combat developer, the feasibility of automatic or semiautomatic mechanized pipe-laying techniques is considered. Two of the joining techniques being considered are amenable to mechanical assembly by automatic equipment. These include the RACE-BILT couplings included in Alternatives III, V, and VII and the ZAP-LOK joining technique employed by Alternatives IV, VI, and VIII. References herein to Alternatives III, IV, V, and VI when a mechanized assembly process is used will be as Alternatives III-A, IV-A, V-A, and VI-A, respectively.

Military pipeline design criteria states that the throughput of different types of fuels to be pumped must be considered, and the heaviest fuel making up 24 percent or more of the total throughput is to be taken as the design fuel.[24] Diesel fuel is the heaviest of all fuels likely to be pumped through a Military pipeline. The evaluation criteria established in paragraph 5 herein states diesel fuel represents 30 percent of the total throughput. Therefore, all pipeline design calculations are based on diesel fuel at 60°F having a 0.8448 specific gravity[25] and a kinematic viscosity of 3.85 centistokes.[26]

The maximum daily throughput requirements for Scenario I is 27,620 barrels per day. Using a design rate of flow of 950 gal/min will allow delivery of the maximum required daily throughput in approximately 20 hours of operation. This flow rate is within the flow range normally considered efficient for 8-inch-diameter pipelines.

The throughput rate for Scenario II is specified as 35,000 barrels in 23 hours. This equates to a design rate of flow of 1,065 gal/min.

---

[24] Department of the Army Technical Manual, *Military Petroleum Pipeline Systems*, TM 5-343, February 1969, P. 6-1.

[25] *Ibid.* p. 6-2.

[26] *Ibid.* p. C-4.

A summary of the design calculations used to determine the system characteristics for each alternative is contained in Appendix E. The principle design features for each Alternative are tabulated below:

a. **Alternative I.**

Pipe – Fiberglass-reinforced epoxy resin manufactured by Ciba-Geigy Corporation with PRONTO-LOCK, threaded, mechanical couplings bonded on pipe ends. (Refer to Figure 28).

| | |
|---|---|
| Nominal Outside Diameter (in.) | 8-5/8 |
| Nominal Inside Diameter (in.) | 8.3 |
| Nominal Wall Thickness (in.) | 0.15 |
| Maximum Safe Working Pressure (lb/in²) | 150 |
| (feet of diesel fuel) (From manufacturer's literature) | 410 |

| | Scenario I | Scenario II |
|---|---|---|
| Design Working Pressure (lb/in²) | 147 | 147 |
| (feet of diesel fuel) | 402 | 401 |
| Number of Booster Pump Stations | 25 | 24 |
| Power Required at Each Booster Station (bhp) | 105 | 117 |
| Number of Pressure Regulation Stations | 6 | N/A |

b. **Alternative II.**

Pipe – Aluminum, 6061-T6 Alloy, Schedule 40 with grooved-end mechanical couplings and gaskets. (Refer to Figure 29.)

| | |
|---|---|
| Outside Diameter (in.) | 8.625 |
| Inside Diameter (in.) | 7.981 |
| Wall Thickness (in.) | 0.322 |
| Maximum Safe Working Pressure (lb/in²) | 800 |
| (feet of diesel fuel) | 2,187 |

| | Scenario I | Scenario II |
|---|---|---|
| Design Working Pressure (lb/in²) | 800 | 761 |
| (feet of diesel fuel) | 2,187 | 2,081 |
| Number of Booster Pump | 4 | 5 |
| Power Required at Each Booster (bhp) | 631 | 675 |
| Number of Pressure Regulation Stations | 1 | N/A |

c. **Alternative III.**

Pipe - Aluminum, 6063-T6 Alloy, Schedule 10 with RACEBILT Industrial couplings manufactured by Race and Race, Inc. (Refer to Figure 30.)

| | |
|---|---|
| Outside Diameter (in.) | 8.625 |
| Inside Diameter (in.) | 8.329 |
| Wall Thickness (in.) | 0.148 |
| Maximum Safe Working Pressure ($lb/in^2$) | 350 |
| (feet of diesel fuel) (as recommended by coupling manufacturer) | 957 |

| | Scenario I | Scenario II |
|---|---|---|
| Design Working Pressure ($lb/in^2$) | 342 | 341 |
| (feet of diesel fuel) | 935.5 | 934 |
| Number of Booster Pump Stations | 9 | 10 |
| Power Required at Each Booster Station (bhp) | 264 | 294 |
| Number of Pressure Regulation Stations | 3 | N/A |

d. **Alternative IV.**

Pipe – Aluminum, 6061-T6 Alloy, Schedule 40 with swaged bell-and-spigot joints formed by the ZAP-LOK process. (Refer to Figure 31.)

| | |
|---|---|
| Outside Diameter (in.) | 8.625 |
| Inside Diameter (in.) | 7.981 |
| Wall Thickness (in.) | 0.322 |
| Maximum Safe Working Pressure ($lb/in^2$) | 1,000 |
| (feet of diesel fuel) | 2,734 |

| | Scenario I | Scenario II |
|---|---|---|
| Design Working Pressure, ($lb/in^2$) | 807 | 946.5 |
| (feet of diesel fuel) | 2,207.5 | 2,588 |
| Number of Booster Pump Stations | 4 | 4 |
| Power Required at Each Booster Station (bhp) | 611 | 840 |
| Number of Pressure Regulation Stations | None | N/A |

e. Alternative V.

Pipe Aluminum, 6063-T6 Alloy with RACEBILT Industrial couplings manufactured by Race and Race, Inc.

| | |
|---|---|
| Outside Diameter (in.) | 8.625 |
| Inside Diameter (in.) | 8.225 |
| Wall Thickness (in.) | 0.200 |
| Maximum Safe Working Pressure (lb/in²) | 482 |
| (feet of diesel fuel) | 1,318 |

| | Scenario I | Scenario II |
|---|---|---|
| Design Working Pressure (lb/in²) | 464 | 449 |
| (feet of diesel fuel) | 1,269 | 1,227 |
| Number of Booster Pump Stations | 7 | 8 |
| Power Required at Each Booster Station (bhp) | 362 | 393 |
| Number of Pressure Regulation Stations | 1 | N/A |

f. Alternative VI.

Pipe – Aluminum, 6061-T6 Alloy, Schedule 10 with swaged bell-and-spigot joints formed by the ZAP-LOK process.

| | |
|---|---|
| Outside Diameter (in.) | 8.625 |
| Inside Diameter (in.) | 8.329 |
| Wall Thickness (in.) | 0.148 |
| Maximum Safe Working Pressure (lb/in²) | 661 |
| (feet of diesel fuel) | 1,807 |

| | Scenario I | Scenario II |
|---|---|---|
| Design Working Pressure (lb/in²) | 661 | 659 |
| (feet of diesel fuel) | 1,807 | 1,803 |
| Number of Booster Pump Stations | 5 | 5 |
| Power Required at Each Booster Station (bhp) | 522 | 583 |
| Number of Pressure Regulation Stations | 1 | N/A |

g. **Alternative VII.**

Pipe – Aluminum, 6063-T6 Alloy, 6-Inch, Schedule 10 with RACEBILT Industrial couplings manufactured by Race and Race, Inc.

| | |
|---|---|
| Outside Diameter (in.) | 6.625 |
| Inside Diameter (in.) | 6.361 |
| Wall Thickness (in.) | 0.134 |
| Maximum Safe Working Pressure ($lb/in^2$) | 410 |
| (feet of diesel fuel) | 1,121 |

| | Scenario I | Scenario II |
|---|---|---|
| Design Working Pressure ($lb/in^2$) | 396 | 376 |
| (feet of diesel fuel) | 1,083 | 1,028 |
| Number of Booster Pump Stations | 8 | 10 |
| Power Required at Each Booster Station (bhp) | 121 | 133 |
| Number of Pressure Regulation Stations | 2 | N/A |

h. **Alternative VIII.**

Pipe – Aluminum, 6061-T6 Alloy, 6-Inch, Schedule 10 with swaged bell-and-spigot joints formed by the ZAP-LOK process.

| | |
|---|---|
| Outside Diameter (in.) | 6.625 |
| Inside Diameter (in.) | 6.361 |
| Wall Thickness (in.) | 0.134 |
| Maximum Safe Working Pressure ($lb/in^2$) | 780 |
| (feet of diesel fuel) | 2,123 |

| | Scenario I | Scenario II |
|---|---|---|
| Design Working Pressure (lb/in ) | 678 | 732 |
| (feet of diesel fuel) | 1,855 | 2,001 |
| Number of Booster Pump Stations | 4 | 5 |
| Power Required at Each Booster Station | 278 | 337 |
| Number of Pressure Regulation Stations | 1 | N/A |

### i. Military Standard Pipeline System.

Pipe  Lightweight Steel Tubing, 8-Inch with grooved pipe nipples welded on ends in accordance with MIL-T-425 for grooved-end mechanical couplings and gaskets.

| | | |
|---|---|---|
| Outside Diameter (in.) | | 8.625 |
| Inside Diameter (in.) | | 8.415 |
| Wall Thickness (in.) | | 0.1046 |
| Maximum Safe Operating Pressure (lb/in²) | | 500 |
| (feet of diesel fuel) | | 1,367 |

| | Scenario I | Scenario II |
|---|---|---|
| Design Working Pressure (lb/in²) | 436 | 483 |
| (feet of diesel fuel) | 1,192 | 1,321 |
| Number of Booster Pump Stations | 7 | 7 |
| Power Required at Each Booster Station (bhp) | 340 | 423 |
| Number of Pressure Regulation Stations | 1 | N/A |

13. **Cost Effectiveness Analysis.** This analysis reviews the major cost elements associated with the construction, operation, and maintenance of each of the candidate pipeline systems. Specific costs contributing directly to the total cost of satisfying the operational requirement of the applicable scenarios are estimated, based on the pipeline system design characteristics listed in paragraph 12. Providing an indication of the relative cost of the alternative pipeline systems, these cost data are used to assist in identifying the candidate system best suited to Military pipeline requirements. A formal TRADOC/DARCOM Cost and Operational Effectiveness Analysis (COEA) will be required to establish the total life-cycle cost for any candidate system selected for continued development.

a. **Basis for Comparison.** For each alternative pipeline system, the following costs are identified in constant FY76 dollars.

(1) **Procurement Costs.** The procurement costs listed represent the projected costs of purchasing the pipe, valves, manifold components, pressure regulation stations, and pumping equipment necessary for construction of 100 miles of pipeline. The costs are estimates based on the general cost relationships presented in Section III rather than a detailed cost estimate for specific items of materiel. For example, the costs of all pumping units were determined as a function of computed derated brake horsepower using the cost versus horsepower relationships shown in Figure 12. These cost data are indicative only of major end-item manufacturing costs.

No attempt has been made to include research and development, investment non-recurring, or initial provisioning and training costs.

(2) **Transportation Costs.** The transportation cost for each candidate system includes the estimated cost of delivery from the manufacturer to the user in an overseas theater-of-operations for each major component included in the pipeline system. These costs are based on the factors for second destination to overseas users contained in the "Cost Estimating Guidance for Transportation Cost," included herein as Appendix D. These data do not include the cost of delivering the required construction equipment to the theater-of-operations. It is assumed the construction equipment will be available to support a variety of construction projects in addition to installing pipelines.

(3) **Construction Costs.** This category includes the cost of the personnel and equipment involved directly in the installation of the pipe, pump stations, and pressure regulation stations. Excluded are the costs of clearing and grading, grade and river crossings, and other special construction requirements which are primarily dependent on the terrain traversed rather than the method of pipeline construction. In each case, where a trade-off of additional equipment for fewer personnel could be made, the option of fewest personnel was selected unless such a choice would add inordinate costs or equipment requirements. The total resources applied represent the minimum capable of achieving a construction rate of 30 kilometers per day.

For the purposes of computing personnel costs, an average military pay grade of E-5 is assumed. From CMDRAMC message, AMCCP-FA, dated 10 December 1975, listing composite standard rates for use in computing the cost of Military personnel services (Army), the rate for pay grade E-5 is $4.63 per hour. The estimates of manhours expended include all personnel, including equipment operators, directly involved in the installation of the pipeline, pumps, and pressure regulation stations. The associated cost of administrative and support personnel are not included.

The construction equipment costs are computed using the daily ownership and hourly operating and overhaul costs from the COST REFERENCE GUIDE FOR CONSTRUCTION EQUIPMENT, compiled by National Research and Approval Company and published by Equipment Guide-Book Company, 1975 copyright. Where data for the specific items of equipment required were not available, cost data for items of equipment considered to be comparable in cost of ownership and operation were adapted. These cost data account for depreciation, fuel, lubricants, tires, parts, and overhaul and repair labor.

(4) **Operating Costs.** Included in the operating costs are personnel, fuel, and lube oil. Operating manhours are based on a crew consisting of four opera-

tors and a crew chief operating each pump station and performing organizational maintenance. This allows continuous operation having at least two operators on duty at all times without anyone being required to work more than 10 hours per day. As with construction labor costs, the operating costs include only those manhours directly involved with the operation of the pipeline.

(5) **Maintenance Costs.** Included in the total maintenance costs are labor, repair parts, materials, and supplies required for performance of both scheduled and unscheduled maintenance. For Scenario II, the total number of operating hours represents several overhaul periods for the pumping units and, in some cases, exceeds the expected service life of the pump engines. In all cases, pump unit overhaul costs have been included together with the associated cost of transportation round trip to a CONUS overhaul facility. Where appropriate, replacement pump costs, containing overseas transportation costs, have been included in the cost of maintenance.

b. **Cost Analysis.** A summary of the Scenario I and II costs computed for Military standard coupled lightweight steel pipelines is contained in Table 18. These estimated costs for procurement, transportation, construction, operation, and maintenance are based on the system design characteristics described in paragraph 11 with one diesel-engine-driven pump at each booster station.

Table 18. Summary of Costs for Military Standard Pipeline Systems

| Parameters | Cost (thousands of dollars) | |
|---|---|---|
| | Scenario I | Scenario II |
| Procurement | $3,653 | $ 3,639 |
| Transportation | 634 | 633 |
| Construction | 366 | 365 |
| Operation | 562 | 10,420 |
| Maintenance | 268 | 3,755 |
| Total | $5,483 | $18,812 |

The estimated costs for the alternative pipeline systems described in paragraph 11 are tabulated in Tables 18 through 30. The difference in cost resulting from use of diesel-engine-driven pumps and gas-turbine-engine-driven pumps is shown for each proposed alternative 8-inch-diameter pipeline system concept. Turbine engines were not evaluated for the 6-inch-diameter pipeline concepts because commercial turbine engines with the required power ratings are not readily available.

Table 19. Summary of Costs: Alternative I

| Parameters | Cost (Thousands of Dollars) | | | |
|---|---|---|---|---|
| | One Pump per Station | | Two Pumps per Station | |
| | Diesel | Turbine | Diesel | Turbine |
| **Scenario I** | | | | |
| Procurement | $ 4,315 | $ 5,645 | $ 4,799 | $ 7,215 |
| Transportation | 1,339 | 1,310 | 1,376 | 1,329 |
| Construction | 168 | 168 | 178 | 178 |
| Operation | 958 | 1,660 | 966 | 1,782 |
| Maintenance | 324 | 308 | 354 | 321 |
| Total | $ 7,104 | $ 9,091 | $ 7,673 | $10,825 |
| **Scenario II** | | | | |
| Procurement | $ 4,211 | $ 5,532 | $ 4,665 | $ 7,096 |
| Transportation | 1,334 | 1,306 | 1,368 | 1,322 |
| Construction | 166 | 166 | 174 | 174 |
| Operation | 15,263 | 26,907 | 15,343 | 28,977 |
| Maintenance | 5,317 | 6,064 | 6,572 | 8,979 |
| Total | $26,291 | $39,975 | $28,122 | $46,548 |

Table 20. Summary of Costs: Alternative II

| Parameters | Cost (Thousands of Dollars) | | | |
|---|---|---|---|---|
| | One Pump per Station | | Two Pumps per Station | |
| | Diesel | Turbine | Diesel | Turbine |
| **Scenario I** | | | | |
| Procurement | Transportability limits on size and weight of pump units prohibit using one diesel-engine-driven pump to develop the total hydraulic head required at each pump station. | $ 7,773 | $ 7,396 | $ 8,468 |
| Transportation | | 573 | 593 | 577 |
| Construction | | 364 | 366 | 366 |
| Operation | | 1,094 | 657 | 1,271 |
| Maintenance | | 274 | 282 | 276 |
| Total | | $10,078 | $ 9,294 | $10,958 |
| **Scenario II** | | | | |
| Procurement | | $ 8,008 | $ 7,551 | $ 8,287 |
| Transportation | | 577 | 603 | 582 |
| Construction | | 364 | 366 | 366 |
| Operation | | 18,030 | 11,041 | 20,912 |
| Maintenance | | 4,174 | 4,116 | 4,863 |
| Total | | $31,153 | $23,677 | $35,010 |

Table 21. Summary of Costs: Alternative III

| Parameters | Cost (Thousands of Dollars) | | | |
|---|---|---|---|---|
| | One Pump per Station | | Two Pumps per Station | |
| | Diesel | Turbine | Diesel | Turbine |
| Scenario I | | | | |
| Procurement | $ 4,775 | $ 5,397 | $ 4,951 | $ 5,959 |
| Transportation | 1,308 | 1,291 | 1,321 | 1,299 |
| Construction | 148 | 148 | 152 | 152 |
| Operation | 602 | 1,084 | 615 | 1,220 |
| Maintenance | 307 | 302 | 317 | 307 |
| Total | $ 7,140 | $ 8,222 | $ 7,356 | $ 8,937 |
| Scenario II | | | | |
| Procurement | $ 4,772 | $ 5,478 | $ 4,967 | $ 6,103 |
| Transportation | 1,312 | 1,292 | 1,325 | 1,299 |
| Construction | 148 | 148 | 152 | 152 |
| Operation | 11,265 | 20,264 | 11,507 | 22,887 |
| Maintenance | 4,416 | 5,061 | 4,670 | 6,191 |
| Total | $21,913 | $32,243 | $22,621 | $36,632 |

Table 22. Summary of Costs: Alternative III-A

| Parameters | Cost (Thousands of Dollars) | | | |
|---|---|---|---|---|
| | One Pump per Station | | Two Pumps per Station | |
| | Diesel | Turbine | Diesel | Turbine |
| Scenario I | | | | |
| Procurement | $ 4,775 | $ 5,397 | $ 4,951 | $ 5,959 |
| Transportation | 1,308 | 1,291 | 1,321 | 1,299 |
| Construction | 235 | 235 | 239 | 239 |
| Operation | 602 | 1,084 | 615 | 1,220 |
| Maintenance | 307 | 302 | 317 | 307 |
| Total | $ 7,227 | $ 8,309 | $ 7,443 | $ 9,024 |
| Scenario II | | | | |
| Procurement | $ 4,772 | $ 5,478 | $ 4,967 | $ 6,103 |
| Transportation | 1,312 | 1,292 | 1,325 | 1,299 |
| Construction | 235 | 235 | 239 | 239 |
| Operation | 11,265 | 20,264 | 11,507 | 22,887 |
| Maintenance | 4,416 | 5,061 | 4,670 | 6,191 |
| Total | $22,000 | $32,330 | $22,708 | $36,719 |

Table 23. Summary of Costs: Alternative IV

| Parameters | Cost (Thousands of Dollars) | | | |
|---|---|---|---|---|
| | One Pump per Station | | Two Pumps per Station | |
| | Diesel | Turbine | Diesel | Turbine |
| Scenario I | | | | |
| Procurement | Transportability | $ 7,255 | $ 6,711 | $ 7,118 |
| Transportation | limits on size and | 572 | 592 | 576 |
| Construction | weight of pump | 341 | 342 | 342 |
| Operation | units prohibit | 851 | 519 | 988 |
| Maintenance | using one diesel- | 249 | 255 | 251 |
| Total | engine-driven | $ 9,268 | $ 8,419 | $ 9,275 |
| | pump to develop | | | |
| Scenario II | the total hydraulic | | See Note | |
| Procurement | head required at | $ 7,030 | $ 6,656 | $ 7,230 |
| Transportation | each pump station. | 574 | 604 | 578 |
| Construction | | 341 | 344 | 344 |
| Operation | | 17,399 | 10,709 | 19,396 |
| Maintenance | | 2,892 | 3,078 | 3,296 |
| Total | | $28,236 | $21,391 | $30,844 |

NOTE: Transportability limits on size and weight of pump unit require three diesel-engine-driven pump units to develop the total hydraulic horsepower required at each pump station.

Table 24. Summary of Costs: Alternative IV-A

| Parameters | Cost (Thousands of Dollars) | | | |
|---|---|---|---|---|
| | One Pump per Station | | Two Pumps per Station | |
| | Diesel | Turbine | Diesel | Turbine |
| Scenario I | | | | |
| Procurement | Transportability | $ 7,255 | $ 6,711 | $ 7,118 |
| Transportation | limits on size | 572 | 592 | 576 |
| Construction | and weight of | 298 | 300 | 300 |
| Operation | pump units | 851 | 519 | 988 |
| Maintenance | prohibit using | 249 | 255 | 251 |
| Total | one diesel-engine- | $ 9,225 | $ 8,377 | $ 9,233 |
| | driven pump to | | | |
| | develop the total | | | |
| Scenario II | hydraulic head | | See Note | |
| Procurement | required at each | $ 7,030 | $ 6,656 | $ 7,230 |
| Transportation | pump station. | 574 | 604 | 578 |
| Construction | | 298 | 301 | 300 |
| Operation | | 17,399 | 10,709 | 19,396 |
| Maintenance | | 2,892 | 3,078 | 3,296 |
| Total | | $28,193 | $21,348 | $30,800 |

NOTE: Transportability limits on size and weight of pump units require three diesel-engine-driven pump units to develop the total hydraulic horsepower required at each pump station.

Table 25. Summary of Costs: Alternative V

| Parameters | Cost (Thousands of Dollars) | | | |
|---|---|---|---|---|
| | One Pump per Station | | Two Pumps per Station | |
| | Diesel | Turbine | Diesel | Turbine |
| Scenario I | | | | |
| Procurement | Transportability | $ 6,327 | $ 5,905 | $ 6,751 |
| Transportation | limits on size and | 1,286 | 1,312 | 1,292 |
| Construction | weight of pump | 154 | 157 | 157 |
| Operation | units prohibit | 996 | 600 | 1,188 |
| Maintenance | using one diesel- | 309 | 322 | 314 |
| Total | engine-driven | $ 9,072 | $ 8,296 | $ 9,702 |
| | pump to develop | | | |
| Scenario II | the total hydraulic | | | |
| Procurement | head required at | $ 6,512 | $ 6,012 | $ 7,000 |
| Transportation | each pump station. | 1,289 | 1,320 | 1,296 |
| Construction | | 155 | 158 | 158 |
| Operation | | 19,639 | 11,476 | 22,682 |
| Maintenance | | 4,894 | 4,621 | 5,353 |
| Total | | $32,489 | $23,587 | $36,489 |

Table 26. Summary of Costs: Alternative V-A

| Parameters | Cost (Thousands of Dollars) | | | |
|---|---|---|---|---|
| | One Pump per Station | | Two Pumps per Station | |
| | Diesel | Turbine | Diesel | Turbine |
| Scenario I | | | | |
| Procurement | Transportability | $ 6,327 | $ 5,905 | $ 6,751 |
| Transportation | limits on size and | 1,286 | 1,312 | 1,292 |
| Construction | weight of pump | 233 | 235 | 235 |
| Operation | units prohibit | 996 | 600 | 1,188 |
| Maintenance | using one diesel- | 309 | 322 | 314 |
| Total | engine-driven | $ 9,151 | $ 8,374 | $ 9,780 |
| | pump to develop | | | |
| Scenario II | the total hydraulic | | | |
| Procurement | head required at | $ 6,512 | $ 6,012 | $ 7,000 |
| Transportation | each pump station. | 1,289 | 1,320 | 1,296 |
| Construction | | 233 | 236 | 236 |
| Operation | | 19,639 | 11,476 | 22,682 |
| Maintenance | | 4,894 | 4,621 | 5,353 |
| Total | | $32,567 | $23,665 | $36,567 |

Table 27. Summary of Costs: Alternative VI

| Parameters | Cost (Thousands of Dollars) | | | |
|---|---|---|---|---|
| | One Pump per Station | | Two Pumps per Station | |
| | Diesel | Turbine | Diesel | Turbine |
| Scenario I | | | | |
| Procurement | Transportability limits on size and weight of pump units prohibit using one diesel-engine-driven pump to develop the total hydraulic head required at each pump station. | $ 3,847 | $ 3,399 | $ 4,019 |
| Transportation | | 561 | 579 | 563 |
| Construction | | 368 | 370 | 370 |
| Operation | | 952 | 525 | 1,005 |
| Maintenance | | 267 | 273 | 267 |
| Total | | $ 5,995 | $ 5,146 | $ 6,224 |
| Scenario II | | | | |
| Procurement | | $ 3,895 | $ 3,467 | $ 4,171 |
| Transportation | | 560 | 583 | 564 |
| Construction | | 368 | 370 | 370 |
| Operation | | 16,118 | 9,750 | 18,746 |
| Maintenance | | 3,891 | 3,800 | 4,840 |
| Total | | $24,832 | $17,970 | $28,691 |

Table 28. Summary of Costs: Alternative VI-A

| Parameters | Cost (Thousands of Dollars) | | | |
|---|---|---|---|---|
| | One Pump per Station | | Two Pumps per Station | |
| | Diesel | Turbine | Diesel | Turbine |
| Scenario I | | | | |
| Procurement | Transportability limits on size and weight of pump units prohibit using one diesel-engine-driven pump to develop the total hydraulic head required at each pump station. | $ 3,847 | $ 3,399 | $ 4,019 |
| Transportation | | 561 | 579 | 563 |
| Construction | | 300 | 301 | 301 |
| Operation | | 952 | 525 | 1,005 |
| Maintenance | | 267 | 273 | 267 |
| Total | | $ 5,927 | $ 5,077 | $ 6,155 |
| Scenario II | | | | |
| Procurement | | $ 3,895 | $ 3,467 | $ 4,171 |
| Transportation | | 560 | 583 | 564 |
| Construction | | 299 | 301 | 301 |
| Operation | | 16,118 | 9,750 | 18,746 |
| Maintenance | | 3,891 | 3,800 | 4,840 |
| Total | | $24,763 | $17,901 | $28,622 |

Table 29. Summary of Costs: Alternative VII

| Parameters | Cost (Thousands of Dollars) | |
|---|---|---|
| | Scenario I | Scenario II |
| Procurement | $6,596 | $ 6,778 |
| Transportation | 1,748 | 1,760 |
| Construction | 136 | 136 |
| Operation | 762 | 15,696 |
| Maintenance | 452 | 5,386 |
| Total | $9,694 | $29,756 |

Table 30. Summary of Costs: Alternative VIII

| Parameters | Cost (Thousands of Dollars) | |
|---|---|---|
| | Scenario I | Scenario II |
| Procurement | $4,502 | $ 4,700 |
| Transportation | 918 | 932 |
| Construction | 396 | 400 |
| Operation | 562 | 12,082 |
| Maintenance | 398 | 7,306 |
| Total | $6,776 | $25,420 |

Several cost considerations bearing on the selection of the best pipeline system design are evident in Table 18. Examining first the Scenario I costs for the Military standard pipeline system, one finds that procurement costs represent approximately 67 percent of the total costs identified. Including procurement, transportation, and construction, the cost of establishing an operational capability represents 85 percent of the total Scenario I cost for the Military standard system. Thus, for the short duration conflict, the operation and maintenance costs are not of primary concern.

Continued examination of the data presented in Table 18 reveals a contraposition of Scenario II. For this longer period of operation (3 years), the cost of operation becomes the major cost factor constituting more than 55 percent of the total scenario costs identified. Furthermore, the cost for maintenance exceeds procurement cost, becoming the second largest contributor to total scenario costs.

The total Scenario I and Scenario II costs from Tables 18 through 30 are summarized in Table 31. The contribution of procurement, transportation, construction, operation, and maintenance costs to total scenario costs are shown in Table 32 expressed as a percentage of total scenario cost. In general, the relationship between the various components of total scenario costs outlined previously for the Military standard lightweight, coupled, steel system hold true.

For a specific type and size, pipe cost varies in direct proportion to wall thickness. Since the maximum safe working pressure is a direct function of pipe wall thickness, the cost of pipe varies directly with operating pressure. Pipe is the highest cost item in a pipeline system, normally representing more than half the total investment in materiel. As a result, changes in pipeline design characteristics which affect the operating pressure have a corresponding effect on the cost of pipe and significantly affect the total cost of procurement.

In contrast, maintenance costs tend to vary in inverse proportion to operating pressure. Two factors bear on this relationship. In general, a high-pressure, thick-wall pipe will be less susceptible to damage and deterioration than thin-walled, low-pressure pipe. Thus, the high-pressure pipeline will normally require less maintenance than a low-pressure pipeline. In addition, the number of pump stations required is inversely proportional to the operating pressure. The number of pump units in a pipeline system has a greater effect on maintenance costs than the size of the pump units. Therefore, an increase in pipeline operating pressure allows a reduction in the number of pump units and the associated maintenance costs.

The cost of operating a pipeline also tends to vary indirectly with operating pressure. Fewer large pump stations are slightly more efficient than numerous small stations thus creating some cost savings for high-pressure pipelines. More important, the number and cost of operating personnel are a direct function of the number of pump stations. Thus, it is desirable to use as few pump stations as possible.

For both Scenarios I and II, Alternative VI-A has the lowest total cost. Using 8-inch-diameter, 6061-T6 alloy aluminum schedule 10 pipe joined by an automated ZAP-LOK process, this candidate system has a proposed maximum operating pressure of 660 lb/in$^2$. The next lowest cost candidate system is Alternative VI which is the same system concept except commercial construction practices are to be used in lieu of the automatic joining equipment. Using two diesel-engine-driven pumps per station, these are the only alternatives that have total scenario costs that are less than the total cost of a Military standard system.

Table 31. Cost Summary: Scenarios I and II

| Alternative | Cost (Thousands of Dollars) | | | |
|---|---|---|---|---|
| | One Pump per Station | | Two Pumps per Station | |
| | Diesel | Turbine | Diesel | Turbine |
| Scenario I | | | | |
| MIL-STD | $ 5,483 | | | |
| I | 7,104 | $ 9,091 | $ 7,673 | $10,825 |
| II | See Note 1 | 10,078 | 9,294 | 10,958 |
| III | 7,140 | 8,222 | 7,365 | 8,937 |
| III-A | 7,227 | 8,309 | 7,443 | 9,024 |
| IV | See Note 1 | 9,268 | 8,419 | 9,275 |
| IV-A | See Note 1 | 9,225 | 8,377 | 9,233 |
| V | See Note 1 | 9,072 | 8,296 | 9,702 |
| V-A | See Note 1 | 9,151 | 8,374 | 9,780 |
| VI | See Note 1 | 5,995 | 5,146 | 6,224 |
| VI-A | See Note 1 | 5,927 | 5,077 | 6,155 |
| VII | – | – | 9,694 | – |
| VIII | 6,776 | – | – | – |
| Scenario II | | | | |
| MIL-STD | 18,812 | | | |
| I | 26,291 | 39,975 | 28,122 | 46,548 |
| II | See Note 1 | 31,153 | 23,677 | 35,010 |
| III | 21,913 | 32,243 | 22,621 | 36,632 |
| III-A | 22,000 | 32,330 | 22,708 | 36,719 |
| IV | See Note 2 | 28,236 | 21,391[2] | 30,844 |
| IV-A | See Note 1 | 28,193 | 21,348[2] | 30,800 |
| V | See Note 1 | 32,489 | 23,587 | 36,489 |
| V-A | See Note 1 | 32,567 | 23,665 | 36,567 |
| VI | See Note 1 | 24,832 | 17,970 | 28,691 |
| VI-A | See Note 1 | 24,763 | 17,901 | 28,622 |
| VII | – | – | 29,756 | – |
| VIII | 25,420 | – | – | – |

NOTES:

1. Transportability limits on size and weight of pump units prohibit using one diesel-engine-driven pump to develop the total hydraulic head required at each pump station.
2. Transportability limits on size and weight of pump units require three diesel-engine-driven pump units to develop the total hydraulic horsepower required at each pump station.

Table 32. Breakdown of Scenario Costs in Percentage of Total Costs

| Alternative | Procurement (%) | Transportation (%) | Construction (%) | Operation (%) | Maintenance (%) | Total (%) |
|---|---|---|---|---|---|---|
| Scenario I | | | | | | |
| MIL-STD | 66.6 | 11.6 | 6.7 | 10.2 | 4.9 | 100 |
| I | 60.7 | 18.8 | 2.4 | 13.5 | 4.6 | 100 |
| II | 79.6 | 6.4 | 3.9 | 7.1 | 3.0 | 100 |
| III | 66.9 | 18.3 | 2.1 | 8.4 | 4.3 | 100 |
| III-A | 66.1 | 18.1 | 3.3 | 8.3 | 4.2 | 100 |
| IV | 79.7 | 7.0 | 4.1 | 6.2 | 3.0 | 100 |
| IV-A | 80.1 | 7.1 | 3.6 | 6.2 | 3.0 | 100 |
| V | 71.2 | 15.8 | 1.9 | 7.2 | 3.9 | 100 |
| V-A | 70.5 | 15.7 | 2.8 | 7.2 | 3.8 | 100 |
| VI | 66.1 | 11.3 | 7.2 | 10.2 | 5.3 | 100 |
| VI-A | 67.0 | 11.4 | 5.9 | 10.3 | 5.4 | 100 |
| VII | 68.0 | 18.0 | 1.4 | 7.9 | 4.7 | 100 |
| VIII | 66.6 | 11.6 | 6.7 | 10.2 | 4.9 | 100 |
| Scenario II | | | | | | |
| MIL-STD | 19.3 | 3.4 | 1.9 | 55.4 | 20.0 | 100 |
| I | 16.0 | 5.1 | 0.6 | 58.1 | 20.2 | 100 |
| II | 31.9 | 2.5 | 1.6 | 46.6 | 17.4 | 100 |
| III | 21.8 | 6.0 | 0.7 | 51.4 | 20.1 | 100 |
| III-A | 21.7 | 6.0 | 1.1 | 51.2 | 20.0 | 100 |
| IV | 31.1 | 2.8 | 1.6 | 50.2 | 14.4 | 100 |
| IV-A | 31.2 | 2.8 | 1.4 | 48.6 | 14.4 | 100 |
| V | 25.5 | 5.6 | 0.7 | 48.5 | 19.6 | 100 |
| V-A | 25.4 | 5.6 | 1.0 | 54.3 | 19.5 | 100 |
| VI | 19.3 | 3.2 | 2.1 | 54.5 | 21.1 | 100 |
| VI-A | 19.4 | 3.2 | 1.7 | 52.7 | 21.2 | 100 |
| VII | 22.8 | 5.9 | 0.5 | 47.5 | 18.1 | 100 |
| VIII | 18.5 | 3.7 | 1.6 | 55.4 | 28.7 | 100 |

The three candidate systems (Alternatives II, IV, and IV-A) having the highest maximum safe operating pressures (800, 1,000 and 1,000 lb/in², respectively) are also the three highest cost systems for Scenario I. This emphasizes the fact that high procurement costs associated with high operating pressures cannot be offset by savings in operation and maintenance costs if the pipeline is to be in operation only a short period of time.

As operating time increases, the contribution of operation and maintenance costs becomes significantly more important. This is readily apparent from the data in Table 32. For Scenario I, operation and maintenance costs are estimated to represent approximately 10 percent of the total cost for the 90-day period. For the 3 years of operation in Scenario II, operation and maintenance costs represent approximately 70 percent of the total cost.

Table 33 lists each alternative in order of total scenario cost. This ranking shows that 8 of the 13 alternatives fall in the same order for both scenarios. For the other 5 alternatives, there is no correlation between their positions in the rankings. Solely on the basis of cost, Alternatives VI and VI-A offer the only opportunity for improvement over the existing Military standard system. The next lowest cost systems for both short- and long-term operations would be Alternatives III or III-A, followed by Alternatives V or V-A.

Table 33. Ranking of Alternatives in Order of Total Scenario Costs

| Scenario I | | Scenario II | |
|---|---|---|---|
| Alternative | Total Cost | Alternative | Total Cost |
| VI-A | $5,077 | VI-A | $17,901 |
| VI | 5,146 | VI | 17,970 |
| MIL-STD | 5,483 | MIL-STD | 18,812 |
| VIII | 6,776 | IV-A | 21,348 |
| I | 7,104 | IV | 21,391 |
| III | 7,140 | III | 21,913 |
| III-A | 7,227 | III-A | 22,000 |
| V | 8,296 | V | 23,587 |
| V-A | 8,374 | V-A | 23,665 |
| IV-A | 8,377 | II | 23,677 |
| IV | 8,419 | VIII | 25,420 |
| II | 9,294 | I | 26,291 |
| VII | 9,694 | VII | 29,756 |

NOTES:
1. Cost shown in thousands of dollars.
2. Cost shown is for the least-cost pump station configuration for each alternative.

Without exception, diesel-engine-driven pumps offer significant savings in comparison to turbine-engine-driven pumps for every alternative system. For Scenario I, this savings ranges from 10 to 41 percent. For this 90-day mission, the higher initial procurement cost of turbine engines is the major factor contributing to the higher overall cost. For Scenario II, the savings with diesel-engine-driven pumps ranges from 43 to 66 percent. In this longer term application the higher fuel cost for turbine engines overrides all other cost considerations. As the petroleum shortage continues to force fuel prices upward, the high fuel consumption of turbine engines becomes an ever-increasing liability. Turbine engines may offer some savings in maintenance costs, however, those savings cannot offset their higher initial investment and fuel costs.

The principal advantage of turbine-engine-driven pumps for Military pipeline applications is their low weight in comparison to diesel-engine-driven pumps of equal capacity. This feature is of particular importance if the intent is to use only one pump unit at each pump station in a high-pressure pipeline. Reliability considerations which indicate the need for at least two pump units at each pump station diminishes this weight advantage, since each unit will necessarily be smaller and lighter in weight. Since the size and weight of diesel-engine-driven pump units of the capacities being considered in this study do not exceed Military transportability limits, the higher costs associated with turbine-engine-driven pump units cannot be justified.

14. **Operational Effectiveness Analysis.** The purpose of this analysis is to provide a measure of effectiveness for each alternative pipeline system concept identified in paragraph 11. Employing the NSIA trade-off technique (reference Appendix B), each alternative is compared to the Military standard, lightweight, steel coupled pipeline. The result is a computed value indicative of the relative effectiveness for each candidate system design concept.

a. **Definition of Operational Effectiveness Evaluation Parameters.** An indeterminable number of factors having many intricate interrelationships bear on the operational effectiveness of a pipeline system. Thus, to contain this analysis within achievable bounds, only parameters having primary significance and a measurable effectiveness are evaluated.

Recognizing the pitfalls associated with less than an all-encompassing treatment of the subject, the considerations listed in Table 34 are selected as consistent with the objectives of this study. The definition of each consideration is purposely kept as broad as possible and still retain congruency of meaning in the evaluation of the alternative pipeline concepts.

Table 34. Operational Effectiveness Evaluation Criteria

| Parameters | Considerations | Relative Weighting | Basic Rating | | Adjusted Values | |
|---|---|---|---|---|---|---|
| | | | Undesirable | Desirable | Undesirable | Desirable |
| Construction | Rate | 4 | | | | |
| | Joint Reliability | 4 | | | | |
| | Number of Personnel | 4 | | | | |
| | Special Skills and Training | 2 | | | | |
| | Equipment Requirements | 3 | | | | |
| | Transportability | 2 | | | | |
| | Environmental Factors | 1 | | | | |
| Operation | Mission Reliability | 4 | | | | |
| | Number of Personnel | 4 | | | | |
| | Special Skills and Training | 2 | | | | |
| | Fuel Consumption | 3 | | | | |
| Maintenance | Ease of Repair | 3 | | | | |
| | Number of Personnel | 4 | | | | |
| | Special Skills and Training | 2 | | | | |
| | Equipment Requirements | 3 | | | | |
| Other | Vulnerability | 3 | | | | |
| | Durability | 2 | | | | |
| | Safety | 1 | | | | |
| | Storage | 3 | | | | |
| | Recoverability | 4 | | | | |
| Totals | | 58 | | | | |

Net Adjusted Value = Average Net Value =

(1) **Construction Parameters.** The following considerations relate to the construction effort required to establish a pipeline operational capability.

(a) **Rate.** In accordance with the study objectives set out in paragraph 4, the desired construction capability is to advance a pipehead as rapidly as possible with a goal of 30 kilometers per day. Assuming 20 hours construction operating time per day, the desired construction rate is 1.5 kilometers (or approximately 0.93 miles per hour). The basic rating for each alternative is computed as a ratio of the construction rate of a single crew using the proposed construction technique versus the rate of construction of a lightweight steel, coupled pipeline by a crew following the procedure outlined in reference.[27] The evaluation of construction rate is made independent of the number of personnel or amount of equipment employed by the construction crew.

(b) **Joint Reliability.** For the purposes of this evaluation, joint reliability refers to the probability that, when assembled, following normal operating procedures, the joints in the assembled pipeline will have adequate strength and will

[27] *Military Petroleum Pipeline Systems*, Headquarters, Department of the Army, Washington, D.C., TM5-343, February 1969.

not leak. Weak or leaking joints requiring rework before the pipeline can be placed in operation require additional construction effort. The net result is a reduction in the effective rate of construction. On this basis, joint reliability is an essential element of rapid pipeline construction.

(c) **Number of Personnel.** Each alternative pipeline concept is evaluated on the basis of the number of construction manhours required to emplace 100 miles of 8-inch surface-laid pipeline, including pump stations and pressure-regulation stations, under average conditions. Manpower requirements for clearing and grading, grade and river crossings, and other special construction requirements which are primarily a function of the terrain traversed by the pipeline rather than the method of joining or construction procedure are not considered in the evaluation. The estimated construction man-hours do reflect the use of multiple crews to achieve a construction rate of 30 kilometers per day.

(d) **Special Skills and Training.** The need for special skills and training for construction personnel employed in the proposed construction procedure are compared to existing Army pipeline construction training programs. Skills which can be developed and maintained only through formal training are of primary concern. Skills which can be developed to an acceptable degree of proficiency after minimal on-the-job training without formal or prior training have little bearing on the assigned rating.

(e) **Equipment Requirements.** The amount and type of equipment required to install the candidate pipeline system are compared to the requirements for installing lightweight, steel, coupled pipelines. The highly desired construction procedure would employ nothing more than a minimal number of standard Military vehicles to deliver the pipe to the job site. Requirements for excessive amounts of standard construction equipment are equally as undesirable as the need for highly specialized items of support equipment.

(f) **Transportability.** The movement of materials and equipment from the manufacturers to an overseas construction site involves a complex transportation effort. Consideration is given to all elements of the transportation system, both commercial and military, assessing the burden of moving the tremendous tonnage comprising a pipeline system. Of primary concern are any special handling requirements imposed by the equipment to be transported. The transportability limits this study places on equipment design precludes any unacceptable transportation demands.

(g) **Environmental Factors.** Encompassed in this consideration is any environmental factor that could impede achieving the desired construction capability.

(2) **Operation Parameters.** The following considerations relate to the operation of Military pipeline systems.

(a) **Mission Reliability.** As defined previously, mission reliability is the probability that a quantity of fuel equal to the minimum daily consumption can be transported from a port-of-entry to a bulk distribution breakdown point. Each alternative is evaluated by comparison to the expected performance of existing Military standard equipment. To the extent possible, all factors bearing on the operational reliability of a pipeline system are considered.

(b) **Number of Personnel.** This consideration is reflective of the number of personnel involved directly in the operation of a pipeline. Requirements for personnel for operation of marine terminals, tank farms, and other facilities which support the pipeline operation but are not directly affected by the pipeline design are not included in this evaluation.

(c) **Special Skills and Training.** Certain basic skills are necessary to operate any pipeline; however, the skills and training that are unique to a specific alternative pipeline concept are of special concern. In addition, consideration is given to the number of personnel that must possess skills unique to the pipeline operation.

(d) **Fuel Consumption.** This is a direct comparison of the estimated fuel consumptions for the alternative pipeline system versus Military standard diesel-engine-driven pumps delivering fuel through an 8-inch, lightweight, steel coupled pipeline.

(3) **Maintenance Parameters.** The following considerations related to the maintenance aspects of a Military pipeline system operation.

(a) **Ease of Repair.** The degree of difficulty encountered in repair of a pipeline has a significant bearing on the time required for repair. This consideration weighs all factors associated with the candidate pipeline system which may affect the capability to properly maintain the pipeline system. This considers such issues as unusual logistical support requirements which may involve excessive administrative down time as well as the level of physical effort associated with accomplishing maintenance and repair tasks.

(b) **Number of Personnel.** As with construction and operation, this consideration deals with the number of personnel involved directly in the maintenance of a pipeline. All estimates of required manpower include performance of scheduled and unscheduled maintenance including repair of a nominal amount of damage from hostile action.

(c) **Special Skills and Training.** The objective is to identify any unusual skills or training that may be required in the performance of pipeline maintenance. Equally important is any requirement to substantially increase the number of personnel in existing training programs or occupational specialities.

(d) **Equipment Requirements.** Of importance are requirements for special equipment or the need to dedicate standard items of equipment specifically to pipeline maintenance support.

(4) **Other Evaluation Parameters.** The following considerations do not fall conveniently into the category of construction, operation, or maintenance but are of sufficient importance to be included in the comparison of the operational effectiveness of the candidate systems.

(a) **Vulnerability.** The vulnerability of the pipeline encompasses the subjectivity to all modes of potential damage from accidental events, through pilferage by the indigenous population to all types of hostile action.

(b) **Durability.** The ability of the pipeline to exist in an operable condition for a long period of time is evaluated. Of interest are such factors as corrosion, deterioration of elastomers, pump and engine wearout, etc., which have an affect on useful service life.

(c) **Safety.** The probability of events involving personal injury, loss of life, or property damage is evaluated. It is assumed that sound engineering judgement is applied to all alternative pipeline designs.

(d) **Storage.** This consideration evaluates each alternative in terms of long-term storage of contingency reserves.

(e) **Recoverability.** The level of effort required to recover a pipeline system and the suitability of the equipment for redeployment is evaluated.

b. **Operational Effectiveness Evaluations.** Based on the findings of the reliability assessment and cost analysis, the operational effectiveness evaluation for each alternative considers only pump stations utilizing two diesel-engine-driven pumps at each booster station to develop the required total dynamic head. The exceptions to this are Alternatives IV and IV-A in which the weight and size of diesel-engine-driven pump units make it necessary to use three pumps at each booster station.

Table 35 shows the alternative pipeline systems in descending order of merit based on the magnitude of the NSIA trade-off scores computed in Tables 36 through 47. The procedure for applying the NSIA trade-off technique, stipulates that a basic rating value of +100 or -100 overrides all other considerations. In cases where one or more system characteristics are assigned a +100 rating while other characteristics receive a -100 rating, the negative or unacceptable rating takes precedence. Examination of the basic rating values in Tables 36 through 47 finds these limiting criteria apply to only two of the alternative pipeline concepts.

Table 35. Summary of Operational Effectiveness Evaluation Scores

| Alternative | NSIA Score |
|---|---|
| V | +16.9 |
| IV-A | +14.4 |
| V-A | +14.0 |
| III | +13.4 |
| III-A | +11.4 |
| II | +10.7 |
| VI-A | +9.7 |
| VI | +9.3 |
| IV | +8.9 |
| VII | +8.0 |
| VIII | -2.2 |
| I | -9.3 |

In the evaluation of Alternative I (Table 36) and Alternative VII (Table 46), the number of personnel required for pipeline operation is assigned a -100 rating. In the evaluation process, the basic rating for the number of operator personnel was equated inversely to the percentage change in operator personnel required when compared to a Military standard, lightweight, steel, coupled pipeline system. In the case of Alternative I, the low maximum safe operating pressure requires a large number of pump stations. This in turn increases the number of operator personnel by more than 240 percent. Any increase of 100 percent or more receives a basic rating of -100 excluding the alternative from consideration. For Alternative VII, the requirement to operate two parallel pipeline systems each having a large number of pump stations results in an increase in operator personnel of more than 100 percent.

The exclusion of Alternative I as a result of its being assigned a basic rating value of -100 has little impact on this analysis. Alternative I is one of two alternatives which have negative average net values. This negative value indicates the operational effectiveness of Alternative I would be less than for the existing Military

Table 36. Operational Effectiveness Evaluation of Alternative I

| Parameters | Considerations | Relative Weighting | Basic Rating | | Adjusted Values | |
|---|---|---|---|---|---|---|
| | | | Undesirable | Desirable | Undesirable | Desirable |
| Construction | Rate | 4 | | +25 | | +100 |
| | Joint reliability | 4 | | +50 | | +200 |
| | Number of personnel | 4 | | +65 | | +260 |
| | Special skills and training | 2 | | +35 | | +70 |
| | Equipment requirements | 3 | | +15 | | +45 |
| | Transportability | 2 | -35 | | -70 | |
| | Environmental factors | 1 | | | | |
| Operation | Mission reliability | 4 | -70 | | -280 | |
| | Number of personnel | 4 | -100 | | -400 | |
| | Special skills and training | 2 | -35 | | -70 | |
| | Fuel consumption | 3 | 0 | 0 | 0 | 0 |
| Maintenance | Ease of repair | 3 | -35 | | -105 | |
| | Number of personnel | 4 | -20 | | -80 | |
| | Special skills and training | 2 | 0 | 0 | 0 | 0 |
| | Equipment requirements | 3 | 0 | 0 | 0 | 0 |
| Other | Vulnerability | 3 | -35 | | -105 | |
| | Durability | 2 | 0 | 0 | 0 | 0 |
| | Safety | 1 | 0 | 0 | 0 | 0 |
| | Storage | 3 | -35 | | -105 | |
| | Recoverability | 4 | 0 | 0 | 0 | 0 |
| | Totals | 58 | | | -1,215 | +675 |

Net Adjusted Value = +675 -1,215 = -540

Average Net Value = -540/58 = -9.3

Table 37. Operational Effectiveness Evaluation of Alternative II

| Parameters | Considerations | Relative Weighting | Basic Rating | | Adjusted Values | |
|---|---|---|---|---|---|---|
| | | | Undesirable | Desirable | Undesirable | Desirable |
| Construction | Rate | 4 | 0 | 0 | 0 | 0 |
| | Joint reliability | 4 | 0 | 0 | 0 | 0 |
| | Number of personnel | 4 | 0 | 0 | 0 | 0 |
| | Special skills and training | 2 | 0 | 0 | 0 | 0 |
| | Equipment requirements | 3 | 0 | 0 | 0 | 0 |
| | Transportability | 2 | | +15 | | +30 |
| | Environmental factors | 1 | 0 | 0 | 0 | 0 |
| Operation | Mission reliability | 4 | | +35 | | +140 |
| | Number of personnel | 4 | | +30 | | +120 |
| | Special skills and training | 2 | 0 | 0 | 0 | 0 |
| | Fuel consumption | 3 | -15 | | -45 | |
| Maintenance | Ease of repair | 3 | 0 | 0 | 0 | 0 |
| | Number of personnel | 4 | -5 | | -20 | |
| | Special skills and training | 2 | 0 | 0 | 0 | 0 |
| | Equipment requirements | 3 | 0 | 0 | 0 | 0 |
| Other | Vulnerability | 3 | | +15 | | +45 |
| | Durability | 2 | | +35 | | +70 |
| | Safety | 1 | | +70 | | +70 |
| | Storage | 3 | | +70 | | +210 |
| | Recoverability | 4 | 0 | 0 | 0 | 0 |
| | Totals | 58 | | | -65 | +685 |

Net Adjusted Value = +685 - 65 = +620

Average Net Value = +620/58 = +10.7

Table 38. Operational Effectiveness Evaluation of Alternative III

| Parameters | Considerations | Relative Weighting | Basic Rating Undesirable | Basic Rating Desirable | Adjusted Values Undesirable | Adjusted Values Desirable |
|---|---|---|---|---|---|---|
| Construction | Rate | 4 | | +40 | | +160 |
| | Joint reliability | 4 | | +35 | | +140 |
| | Number of personnel | 4 | | +70 | | +210 |
| | Special skills and training | 2 | | +20 | | +40 |
| | Equipment requirements | 3 | | +20 | | +60 |
| | Transportability | 2 | -15 | | -30 | |
| | Environmental factors | 1 | | +35 | | +35 |
| Operation | Mission reliability | 4 | -25 | | -100 | |
| | Number of personnel | 4 | -40 | | -160 | |
| | Special skills and training | 2 | 0 | 0 | 0 | 0 |
| | Fuel consumption | 3 | 0 | 0 | 0 | 0 |
| Maintenance | Ease of repair | 3 | -15 | | -45 | |
| | Number of personnel | 4 | -5 | | -20 | |
| | Special skills and training | 2 | 0 | | 0 | |
| | Equipment requirements | 3 | | +35 | | +105 |
| Other | Vulnerability | 3 | 0 | 0 | 0 | 0 |
| | Durability | 2 | | +70 | | +140 |
| | Safety | 1 | | +35 | | +35 |
| | Storage | 3 | | +70 | | +210 |
| | Recoverability | 4 | 0 | 0 | 0 | 0 |
| | Totals | 58 | | | -355 | +1,135 |

Net Adjusted Value = +1,135 -355 = +780
Average Net Value = +780/58 = +13.4

Table 39. Operational Effectiveness Evaluation of Alternative III-A

| Parameters | Considerations | Relative Weighting | Basic Rating Undesirable | Basic Rating Desirable | Adjusted Values Undesirable | Adjusted Values Desirable |
|---|---|---|---|---|---|---|
| Construction | Rate | 4 | | +95 | | +380 |
| | Joint reliability | 4 | | +35 | | +140 |
| | Number of personnel | 4 | | +80 | | +320 |
| | Special skills and training | 2 | -70 | | -140 | |
| | Equipment requirements | 3 | -70 | | -210 | |
| | Transportability | 2 | -15 | | 0 | |
| | Environmental factors | 1 | | +35 | | +35 |
| Operation | Mission reliability | 4 | -25 | | -100 | |
| | Number of personnel | 4 | -40 | | -160 | |
| | Special skills and training | 2 | 0 | 0 | 0 | 0 |
| | Fuel consumption | 3 | 0 | 0 | 0 | 0 |
| Maintenance | Ease of repair | 3 | -15 | | -45 | |
| | Number of personnel | 4 | -5 | | -20 | |
| | Special skills and training | 2 | 0 | | 0 | |
| | Equipment requirements | 3 | | +35 | | +105 |
| Other | Vulnerability | 3 | 0 | 0 | 0 | 0 |
| | Durability | 2 | | +70 | | +140 |
| | Safety | 1 | | +35 | | +35 |
| | Storage | 3 | | +70 | | +210 |
| | Recoverability | 4 | 0 | 0 | 0 | 0 |
| | Totals | 58 | | | -705 | +1,365 |

Net Adjusted Value = +1,365 - 705 = +660

Average Net Value = +660/58 = +11.4

Table 40. Operational Effectiveness Evaluation of Alternative IV

| Parameters | Considerations | Relative Weighting | Basic Rating | | Adjusted Values | |
|---|---|---|---|---|---|---|
| | | | Undesirable | Desirable | Undesirable | Desirable |
| Construction | Rate | 4 | | +20 | | +80 |
| | Joint reliability | 4 | | +70 | | +280 |
| | Number of personnel | 4 | | +80 | | +320 |
| | Special skills and training | 2 | -35 | | -70 | |
| | Equipment requirements | 3 | -70 | | -210 | |
| | Transportability | 2 | | +20 | | +40 |
| | Environmental factors | 1 | -35 | | -35 | |
| Operation | Mission reliability | 4 | | +50 | | +200 |
| | Number of personnel | 4 | | +45 | | +180 |
| | Special skills and training | 2 | -15 | | -30 | |
| | Fuel consumption | 3 | -15 | | -45 | |
| Maintenance | Ease of repair | 3 | -75 | | -225 | |
| | Number of personnel | 4 | -15 | | -60 | |
| | Special skills and training | 2 | -60 | | -120 | |
| | Equipment requirements | 3 | -70 | | -210 | |
| Other | Vulnerability | 3 | | +50 | | +150 |
| | Durability | 2 | | +95 | | +190 |
| | Safety | 1 | | +70 | | +70 |
| | Storage | 3 | | +70 | | +210 |
| | Recoverability | 4 | -50 | | -200 | |
| | Total | 58 | | | -1,205 | +1,720 |

Net Adjusted Value = +1,720-1, 205 = +515

Average Net Value = +515/58 = +8.9

Table 41. Operational Effectiveness Evaluation of Alternative IV-A

| Parameters | Considerations | Relative Weighting | Basic Rating Undesirable | Basic Rating Desirable | Adjusted Values Undesirable | Adjusted Values Desirable |
|---|---|---|---|---|---|---|
| Construction | Rate | 4 | | +95 | | +380 |
| | Joint reliability | 4 | | +70 | | +280 |
| | Number of personnel | 4 | | +85 | | +340 |
| | Special skills and training | 2 | -70 | | -140 | |
| | Equipment requirements | 3 | -80 | | -240 | |
| | Transportability | 2 | | +20 | | +40 |
| | Environmental factors | 1 | -35 | | -35 | |
| Operation | Mission reliability | 4 | | +50 | | +200 |
| | Number of personnel | 4 | | +45 | | +180 |
| | Special skills and training | 2 | 0 | 0 | 0 | 0 |
| | Fuel consumption | 3 | -15 | | -45 | |
| Maintenance | Ease of repair | 3 | -75 | | -225 | |
| | Number of personnel | 4 | -15 | | -60 | |
| | Special skills and training | 2 | -25 | | -50 | |
| | Equipment requirements | 3 | -70 | | -210 | |
| Other | Vulnerability | 3 | | +50 | | +150 |
| | Durability | 2 | | +95 | | +190 |
| | Safety | 1 | | +70 | | +70 |
| | Storage | 3 | | +70 | | +210 |
| | Recoverability | 4 | -50 | | -200 | |
| | Totals | 58 | | | -1,205 | +2,040 |

Net Adjusted Value = +2,040 - 1,205 = +835

Average Net Value = +835/58 = +14.4

Table 42. Operational Effectiveness Evaluation of Alternative V

| Parameters | Considerations | Relative Weighting | Basic Rating | | Adjusted Values | |
|---|---|---|---|---|---|---|
| | | | Undesirable | Desirable | Undesirable | Desirable |
| Construction | Rate | 4 | | +35 | | +140 |
| | Joint reliability | 4 | | +35 | | +140 |
| | Number of personnel | 4 | | +70 | | +280 |
| | Special skills and training | 2 | | +20 | | +40 |
| | Equipment requirements | 3 | | +20 | | +60 |
| | Transportability | 2 | -15 | | -30 | |
| | Environmental factors | 1 | | +35 | | +35 |
| Operation | Mission reliability | 4 | -10 | | -40 | |
| | Number of personnel | 4 | -15 | | -60 | |
| | Special skills and training | 2 | 0 | 0 | 0 | 0 |
| | Fuel consumption | 3 | -10 | | -30 | |
| Maintenance | Ease of repair | 3 | -15 | | -45 | |
| | Number of personnel | 4 | 0 | 0 | 0 | 0 |
| | Special skills and training | 2 | 0 | | 0 | |
| | Equipment requirements | 3 | | +35 | | +105 |
| Other | Vulnerability | 3 | 0 | 0 | 0 | 0 |
| | Durability | 2 | | +70 | | +140 |
| | Safety | 1 | | +35 | | +35 |
| | Storage | 3 | | +70 | | +210 |
| | Recoverability | 4 | 0 | 0 | 0 | 0 |
| | Totals | 58 | | | -205 | +1,185 |

Net Adjusted Value = +1,185 - 205 = +980

Average Net Value = +980/58 = +16.9

Table 43. Operational Effectiveness Evaluation of Alternative V-A

| Parameters | Considerations | Relative Weighting | Basic Rating Undesirable | Basic Rating Desirable | Adjusted Values Undesirable | Adjusted Values Desirable |
|---|---|---|---|---|---|---|
| Construction | Rate | 4 | | +95 | | +380 |
| | Joint reliability | 4 | | +35 | | +140 |
| | Number of personnel | 4 | | +80 | | +320 |
| | Special skills and training | 2 | -70 | | -140 | |
| | Equipment requirements | 3 | -70 | | -210 | |
| | Transportability | 2 | -75 | | -30 | |
| | Environmental factors | 1 | | +35 | | +35 |
| Operation | Mission reliability | 4 | -10 | | -40 | |
| | Number of personnel | 4 | -15 | | -60 | |
| | Special skills and training | 2 | 0 | 0 | 0 | 0 |
| | Fuel consumption | 3 | -10 | | -30 | |
| Maintenance | Ease of repair | 3 | -15 | | -45 | |
| | Number of personnel | 4 | 0 | 0 | 0 | 0 |
| | Special skills and training | 2 | 0 | | 0 | |
| | Equipment requirements | 3 | | +35 | | +105 |
| Other | Vulnerability | 3 | 0 | 0 | 0 | 0 |
| | Durability | 2 | | +70 | | +140 |
| | Safety | 1 | | +35 | | +35 |
| | Storage | 3 | | +70 | | +210 |
| | Recoverability | 4 | 0 | 0 | 0 | 0 |
| | Totals | 58 | | | -555 | +1,365 |

Net Adjusted Value = +1,365 - 555 = +810
Average Net Value = +810/58 + 14.0

Table 44. Operational Effectiveness Evaluation of Alternative VI

| Parameters | Considerations | Relative Weighting | Basic Rating Undesirable | Basic Rating Desirable | Adjusted Values Undesirable | Adjusted Values Desirable |
|---|---|---|---|---|---|---|
| Construction | Rate | 4 | | +40 | | +160 |
| | Joint reliability | 4 | | +70 | | +280 |
| | Number of personnel | 4 | | +70 | | +280 |
| | Special skills and training | 2 | -35 | | -70 | |
| | Equipment requirements | 3 | -70 | | -210 | |
| | Transportability | 2 | | +20 | | +40 |
| | Environmental factors | 1 | -35 | | -35 | |
| Operation | Mission reliability | 4 | | +35 | | +140 |
| | Number of personnel | 4 | | +30 | | +120 |
| | Special skills and training | 2 | -15 | | -30 | |
| | Fuel consumption | 3 | 0 | 0 | 0 | 0 |
| Maintenance | Ease of repair | 3 | -75 | | -225 | |
| | Number of personnel | 4 | -15 | | -60 | |
| | Special skills and training | 2 | -60 | | -120 | |
| | Equipment requirements | 3 | -70 | | -210 | |
| Other | Vulnerability | 3 | | +70 | | +210 |
| | Durability | 2 | | +95 | | +190 |
| | Safety | 1 | | +70 | | +70 |
| | Storage | 3 | | +70 | | +210 |
| | Recoverability | 4 | -50 | | -200 | |
| | Totals | 58 | | | -1,160 | +1,700 |

Net Adjusted Value = +1,700 - 1,160 = +540

Average Net Value = +540/58 + 9.3

Table 45. Operational Effectiveness Evaluation of Alternative VI-A

| Parameters | Considerations | Relative Weighting | Basic Rating Undesirable | Basic Rating Desirable | Adjusted Values Undesirable | Adjusted Values Desirable |
|---|---|---|---|---|---|---|
| Construction | Rate | 4 | | +95 | | +380 |
| | Joint reliability | 4 | | +70 | | +280 |
| | Number of personnel | 4 | | +85 | | +240 |
| | Special skills and training | 2 | -70 | | -140 | |
| | Equipment requirements | 3 | -80 | | -240 | |
| | Transportability | 2 | | +20 | | +40 |
| | Environmental factors | 1 | -35 | | -35 | |
| Operation | Mission reliability | 4 | | +35 | | +140 |
| | Number of personnel | 4 | | +30 | | +120 |
| | Special skills and training | 2 | -15 | | -30 | |
| | Fuel consumption | 3 | 0 | 0 | 0 | 0 |
| Maintenance | Ease of repair | 3 | -75 | | -225 | |
| | Number of personnel | 4 | -15 | | -60 | |
| | Special skills and training | 2 | -60 | | -120 | |
| | Equipment requirements | 3 | -70 | | -210 | |
| Other | Vulnerability | 3 | | +50 | | +150 |
| | Durability | 2 | | +95 | | +190 |
| | Safety | 1 | | +70 | | +70 |
| | Storage | 3 | | +70 | | +210 |
| | Recoverability | 4 | -50 | | -200 | |
| | Totals | 58 | | | -1,260 | +1,820 |

Net Adjusted Value = +1,820 - 1,260 = +560
Average Net Value = +560/58 = +9.7

Table 46. Operational Effectiveness Evaluation of Alternative VII

| Parameters | Considerations | Relative Weighting | Basic Rating Undesirable | Basic Rating Desirable | Adjusted Values Undesirable | Adjusted Values Desirable |
|---|---|---|---|---|---|---|
| Construction | Rate | 4 | | +65 | | +260 |
| | Joint reliability | 4 | | +35 | | +140 |
| | Number of personnel | 4 | | +75 | | +300 |
| | Special skills and training | 2 | | +20 | | +40 |
| | Equipment requirements | 3 | | +20 | | +60 |
| | Transportability | 2 | -15 | | -30 | |
| | Environmental factors | 1 | | +35 | | +35 |
| Operation | Mission reliability | 4 | 0 | 0 | 0 | 0 |
| | Number of personnel | 4 | -100 | | -400 | |
| | Special skills and training | 2 | 0 | 0 | 0 | 0 |
| | Fuel consumption | 3 | -15 | | -45 | |
| Maintenance | Ease of repair | 3 | -15 | | -45 | |
| | Number of personnel | 4 | -80 | | -320 | |
| | Special skills and training | 2 | -10 | | -20 | |
| | Equipment requirements | 3 | | +35 | | +105 |
| Other | Vulnerability | 3 | 0 | 0 | 0 | 0 |
| | Durability | 2 | | +70 | | +140 |
| | Safety | 1 | | +35 | | +35 |
| | Storage | 3 | | +70 | | +210 |
| | Recoverability | 4 | 0 | 0 | 0 | 0 |
| | Totals | 58 | | | -860 | +1,325 |

Net Adjusted Value = +1,325 - 860 = + 465
Average Net Value = +465/58 = +8.0

Table 47. Operational Effectiveness Evaluation of Alternative VIII

| Parameters | Considerations | Relative Weighting | Basic Rating Undesirable | Basic Rating Desirable | Adjusted Values Undesirable | Adjusted Values Desirable |
|---|---|---|---|---|---|---|
| Construction | Rate | 4 | | +35 | | +140 |
| | Joint reliability | 4 | | +70 | | +280 |
| | Number of personnel | 4 | | +35 | | +140 |
| | Special skills and training | 2 | -35 | | -70 | |
| | Equipment requirements | 3 | -70 | | -210 | |
| | Transportability | 2 | -15 | | -30 | |
| | Environmental factors | 1 | -35 | | -35 | |
| Operation | Mission reliability | 4 | | +35 | | +140 |
| | Number of personnel | 4 | -15 | | -60 | |
| | Special skills and training | 2 | 0 | 0 | 0 | 0 |
| | Fuel consumption | 3 | -15 | | -45 | |
| Maintenance | Ease of repair | 3 | -75 | | -225 | |
| | Number of personnel | 4 | -80 | | -320 | |
| | Special skills and training | 2 | -25 | | -50 | |
| | Equipment requirements | 3 | -35 | | -105 | |
| Other | Vulnerability | 3 | | +35 | | +105 |
| | Durability | 2 | | +70 | | +140 |
| | Safety | 1 | | +70 | | +70 |
| | Storage | 3 | | +70 | | +210 |
| | Recoverability | 4 | -50 | | -200 | |
| | Totals | 58 | | | -1,350 | +1,225 |

Net Adjusted Value = +1,225 - 1,350 = -125
Average Net Value = -125/58 = -2.2

standard system. Similarly, Alternative VII is not a desirable concept because of its relatively low NSIA score.

In the statement of the study objectives, a 30-kilometer-per-day construction rate was specified as a goal. It was then assumed that any construction procedure that would achieve the desired objective of 30 kilometers per day would merit a +100 basic rating. Further, it is assumed that a RDT&E program to develop a fully automatic pipe-joining capability will be undertaken only if there is adequate evidence the 30-kilometer-per-day construction rate can be achieved. Based on these assumptions, all mechanized pipe-laying concepts could logically be assigned a +100 rating for construction rate. None of the alternatives being considered provides the capability for a single crew to construct 30 kilometers per day without using a fully automated pipe-joining machine. Applying the rule that a +100 rating overrides all other factors would then lead to a decision to develop a mechanized pipe-laying capability.

Although the design goal for any automatic pipe-laying machine would be 30 kilometers per day, there is some technological risk and the goal may not be fully achieved. Because of this risk, each mechanized alternative is assigned a maximum rating of +95 for construction rate. This precludes a predisposition to develop an automatic pipe-laying process with negligible effect on the average net values for the fully automated alternatives.

The pipeline concept receiving the highest NSIA trade-off score (16.9) for operational effectiveness is Alternative V. A key feature of this pipeline system concept appears to be the RACEBILT Industrial couplings which provide the capability to rapidly emplace and recover the pipeline while allowing a moderately high pipeline operating pressure. Review of the cost data summarized in Table 33 shows Alternative V to have comparatively high mission costs for both Scenario I and Scenario II. In contrast, Alternative IV-A ranks second in operational effectiveness with an NSIA trade-off score of 14.4 while having the lowest mission costs (refer to Table 33). Thus, the NSIA trade-off scores from Tables 36 through 47 and the cost data from Table 33 are used in Table 48 to compute values reflecting the combination of cost and operational effectiveness for each alternative pipeline system.

The cost ratios for Scenario I listed in column 3 of Table 48 are obtained by dividing the applicable mission cost for Scenario I by the estimated Scenario I mission cost for the present Military standard pipeline system. Similarly, the cost ratios for Scenario II are the Scenario II costs listed in Table 33 divided by the estimated Scenario II cost for the military standard system. The composite cost ratio for each alternative is obtained by dividing the sum of Scenario I and Scenario II costs by the sum of the two scenario costs for the Military standard pipeline system.

Table 48. Cost and Operational Effectiveness Results

| Alternative (1) | NSIA Trade-Off Score (2) | Cost Ratio | | | Cost and Operational Effectiveness Score | | |
|---|---|---|---|---|---|---|---|
| | | Scenario I (3) | Scenario II (4) | Composite (5) | Scenario I (6) | Scenario II (7) | Composite (8) |
| I | −9.3 | 1.40 | 1.50 | 1.47 | −13.0 | −14.0 | −13.7 |
| II | +10.7 | 1.70 | 1.41 | 1.47 | +6.3 | +7.6 | +7.3 |
| III | +13.4 | 1.34 | 1.20 | 1.23 | +10.0 | +11.2 | +10.9 |
| III-A | +11.4 | 1.36 | 1.21 | 1.24 | +8.4 | +9.4 | +9.2 |
| IV | +8.9 | 1.54 | 1.14 | 1.23 | +5.8 | +7.8 | +7.2 |
| IV-A | +14.4 | 1.53 | 1.13 | 1.22 | +9.4 | +12.7 | +12.7 |
| V | +16.9 | 1.51 | 1.25 | 1.31 | +11.2 | +13.5 | +12.9 |
| V-A | +14.0 | 1.53 | 1.26 | 1.32 | +9.2 | +11.1 | +10.6 |
| VI | +9.3 | 0.94 | 0.96 | 0.95 | +9.9 | +9.7 | +9.8 |
| VI-A | +9.7 | 0.93 | 0.95 | 0.95 | +10.4 | +10.2 | +10.2 |
| VII | +8.0 | 1.77 | 1.58 | 1.62 | +4.5 | +5.1 | +4.9 |
| VIII | −2.2 | 1.24 | 1.35 | 1.33 | −2.7 | −3.0 | −2.9 |

The cost and operational effectiveness scores listed in columns 6, 7, and 8 of Table 48 are equal to the NSIA trade-off scores, listed in column 2, divided[28] by the applicable cost ratio. This adjustment of the operational effectiveness scoring values in proportion to mission costs significantly changes the magnitude of most of the scores while having only a minor effect on a few alternatives. The rank order for the resulting cost adjusted values differ not only from the rank order of the basic NSIA trade-off scores but also between Scenario I and Scenario II. The rank order for the composite cost and operational effectiveness scores is the same, except for Alternatives II and IV, for the Scenario II scores. In all cases, however, Alternative V retains the highest cost and operational effectiveness score.

From Table 48, column 6, Alternative VI-A ranks second for the short duration operation represented by Scenario I. For the longer duration operation depicted by Scenario II and for the composite evaluation, Alternative IV-A ranks second. Both of these concepts are based on developing a fully automated pipe-laying machine to install aluminum pipe using the ZAP-LOK joining process. Logically, Alternative VI-A proposes using a relatively lightweight schedule 10 pipe to minimize the equipment procurement cost to be amortized during the relative short mission duration of Scenario I. Following the same rationale, the higher cost of the heavier wall schedule 40 pipe can be offset by increases in operating efficiency and reduced maintenance costs over the longer period of operation in Scenario II. Since the length of a potential future Military conflict cannot be projected accurately, the best pipeline system using the ZAP-LOK joining technique is probably a compromise using a pipe wall thickness somewhere between the 0.148 inch evaluated as Alternative VI-A and the 0.322 inch considered in Alternative IV-A. Examination of the basic rating values assigned to the various considerations in evaluating Alternatives IV-A and VI-A, Tables 41 and 45, respectively, does not indicate such a compromised pipeline design concept would result in a cost and operational effectiveness score greater than the score computed for Alternative V, Table 42.

The cost and operational effectiveness data presented heretofore do not consider two important factors, research and development requirements and logistical support. As noted previously in the discussion of technological risks, developing the fully automated pipe-laying equipment proposed by Alternatives IV-A and VI-A would require an extensive research and development program. In contrast, Alternative V is based on manual assembly of the pipeline. Other than the vehicles needed to deliver the pipe to the construction site, there is no requirement for equipment to assemble the pipeline.

---

[28] Negative NSIA trade-off scores must be multiplied by the cost ratios to maintain consistent relationships between the scores.

Peacetime Military pipeline construction requirements, if any, will be associated with personnel training programs. Even under the most severe combat conditions that can be envisioned, the requirements in a theater of operations will never exceed simultaneous construction of more than a few pipelines. Although it is essential for the Army to have a petroleum pipeline construction capability, any specialized equipment used solely for this purpose will be a low-density item with limited utilization. Consequentially, any large expenditure for research and development will represent a significant part of the total life cycle costs of the pipeline system selected.

Following the same rationale, the logistic support costs for a specialized item of pipeline construction or maintenance equipment will add to the life-cycle cost for the pipeline system. Thus, research and development and logistic support requirements reinforce the desirability of the simple, manual pipe-joining technique proposed by Alternative V as opposed to other complex, mechanized pipeline construction procedures.

The proposed mechanical method of joining pipe is an expiedient to rapid pipeline construction with minimal skills. Typically, rapidly assembled joints are inherently less reliable than welded joints. Thus, quick coupling methods should be used only when operational requirements will not allow use of a more reliable joining technique.

As noted in Paragraph 3, the Army cannot develop and maintain the high degree of skills required for construction of welded pipeline. The ZAP-LOK joining process offers a viable alternative to welded pipe joints for applications where high rates of construction are not an essential requirement. Military adaptation of the commercial ZAP-LOK joining process would provide a Military capability to construct high pressure pipelines without highly skilled personnel. An assessment of potential requirements for future Military construction of high-pressure, permanent pipelines is needed to determine if the ZAP-LOK process should be adopted for Army use.

The use of flexible hoseline systems for some fuel transportation applications has frequently received considerable attention. As a result, the Army 4-inch hoseline outfit, FSN 3835-892-5157, and the 6-inch hoseline equipment from the U.S. Marine Corps Amphibious Assault Fuel System (AAFS) were included in the investigation of pipe materials and construction techniques conducted by the Value Engineering Company. These systems are identified as concepts 1234E and 1240E, respectively, in paragraph 8 herein. In both cases, other concepts were found to be better suited to overland transportation of large quantities of fuels.

In comparison to the alternative system designs evaluated in the preceding paragraphs, hoseline systems would have operational characteristics most nearly approximating Alternative I, low-pressure, fiberglass-reinforced, plastic pipe. The low pressure rating of hoseline is the major weakness if the fuel must be transferred more than a few miles. For longer distances, excessive numbers of booster pump stations are required. As the number of pump stations increases, equipment procurement costs rise. More important is the proportionate increase in the number of operator and maintenance personnel required.

Problems associated with low-pressure operations overshadow the advantages of flexible hoseline systems. There have been attempts by Industry to develop lightweight flexible hose suitable for moderate- to high-pressure application. However, the research and development effort needed to solve the problems encountered have not been forthcoming because of lack of funds. Without established requirements for a high-pressure hoseline system, there has been no justification for the Government to fund such a program. Lacking a definitive market potential, Industry will not pursue the matter without Government funding.

Flexibility is the principal advantage of hoseline systems. The ability to traverse extremely uneven terrain and to change directions without special fittings or the problems of bending pipe can significantly decrease the installation effort of some applications. In addition, if the hose is flexible enough to collapse, methods for laying the hose such as flaking and rolling on reels allow more rapid installation than possible with discrete lengths of rigid pipe.

Along with allowing more rapid installation procedures, collapsing the hose for storage and transportation greatly reduces the volume to be handled. For example, in the 4-inch hoseline outfit, 1,000 feet of 4-inch hose is flaked into a container measuring 12 feet in length, 6 feet in width, and 1 foot high. A stack containing 1,000 feet of 4-inch pipe which is 12 feet long and 6 feet wide would be approximately 2 feet high. Thus, the cubage of a 4-inch hose when collapsed is approximately one half the cubage of an equivalent length of 4-inch pipe. Assuming the hose wall thickness does not become too large to allow collapsing the hose tightly, the space saving is even greater with larger diameters.

Weight must also be considered when evaluating the transportability of hose in comparison to pipe. The 150-lb/in$^2$ working pressure of the hose in the 4-inch hoseline outfit weighs approximately 1.65 pounds per foot of length. A 6061-T6 alloy aluminum pipe having a 4-inch inside diameter and weighing 1.65 pounds per foot would have a wall thickness of approximately 0.109 inch and an allowable working pressure of nearly 1,000 lb/in$^2$. Thus, for an equivalent weight, a 4-inch, 6061-T6 aluminum pipe will allow a working pressure more than 6 times

greater than that of the 4-inch hoseline outfit. Similarly, the 100-lb/in$^2$-working-pressure, 6-inch hose in the Marine Corps AAFS weighs approximately 2.3 lb/ft. A 6061-T6 aluminum pipe having a 6-inch inside diameter and weighing 2.3 lb/ft would have a wall thickness of approximately 0.095 inch and a maximum allowable working pressure approaching 600 lb/in$^2$. In this case, use of aluminum pipe allows a six-fold increase in working pressure for the same weight and a corresponding reduction in the number of pump stations required.

The reduced volume of collapsible hose is an advantage for surface transportation where, for most vehicles and watercraft, the amount of pipe that can be carried is a function of available cargo space rather than allowable weight load limit. For overseas shipment, 35,000 pounds is the load limit for C-130 aircraft. A flaking tray from the 4-inch hoseline outfit containing 1,000 feet of hose weighs approximately 2,000 pounds. Within the weight limit, one C-130 aircraft can carry 17 flaking trays or 17,000 feet of collapsible 4-inch hose. An equal quantity of 0.095-inch-wall, 4-inch-inside-diameter, aluminum pipe will fit into the cargo hold of a C-130 aircraft without completely filling the usable space.

The 35,000-pound maximum load limit will allow a C-130 aircraft to carry approximately 14,000 feet of 6-inch hose from the Marine Corps AAFS. Space limitations will allow a maximum of approximately 11,000 feet of 6-inch-inside-diameter by 0.095-inch-wall aluminum pipe to be loaded into a C-130 aircraft. In this case, an aircraft can carry approximately 25 percent more hose than aluminum pipe. This difference becomes inconsequential when one considers that the hoseline will require 2 booster pumps for each aircraft load of hose while the pipeline will need only 1 large booster pump for 5 aircraft loads of pipe.

Insufficient data are available on the physical characteristics of an 8-inch lightweight collapsible hose to make a direct comparison with 8-inch aluminum pipe. However, the relationship is expected to be similar to that of the 6-inch hoseline versus pipeline discussed above.

In June 1970, the Navy laboratories were tasked by the Chief of Naval operations to aid the Marine Corps in developing equipment to satisfy present and future needs. As a part of this program, the Civil Engineering Laboratory (CEL), Port Hueneme, California reviewed the Marine Corps fuel storage and distribution capabilities. The CEL study found the Marine Corps AAFS to be satisfactory for future use when employed in its normal mode of transferring fuel from a shore facility to a Tactical Airfield Fuel Dispensing System (TAFDS) located a maximum of 5 miles away. However, future requirements for resupply of fuel are expected to include delivery of fuel to remote expeditionary sites located more than 25 miles from the typical

TAFDS sites. The CEL study concluded that the present Marine Corps systems cannot effectively supply the required fuel over these long distances.[29] Even if hose is found to be superior to pipe on the basis of operational effectiveness, its use is difficult to justify because of the high cost. The price of the 6-inch lightweight hose in the Marine Corps AAFS is approximately $7.00 per foot of length. The 6-inch, 6061-T6 aluminum pipe to which it has been compared would cost approximately $2.25 per foot of length.

Experience has shown lightweight hoselines to be a highly versatile means for the distribution of bulk fuels in support of assault operations where flexibility, extreme mobility, rapid deployment, and frequent relocation are essential to mission success. As the situation stabilizes, distances increase, and the volume of fuel to be supplied grows, hoselines must be replaced with pipeline facilities to meet operational requirements. The Army 4-inch hoseline outfit is capable of satisfying many of the operational needs where hoselines are practical. A valid mission statement showing that this system will not meet future Military fuel distribution requirements must be defined before development of a larger capacity system can be justified.

15. **Recommended Pipeline System Design Characteristics.** The large number of candidate pipeline components and systems considered by this study has precluded analysis of each alternative in sufficient depth to develop a detailed design specification for any specific component or pipeline system concept. Instead, the purpose of this study has been to identify the pipeline system concept that is most responsive to future Military bulk fuel distribution requirements. To this end, the following paragraphs outline the general design characteristics of a pipeline system that will function effectively when deployed as a subsystem in a total bulk fuel distribution system operating in a theater-of-operations.

The rate of pipeline construction can be increased while reducing the construction manpower requirements by replacing the present Military standard 20-foot lengths of grooved-end steel pipe and split-ring couplings with longer lengths of aluminum pipe joined by a self-latching mechanical coupling. To achieve the maximum rate of construction, it is desirable to use the longest lengths of pipe consistent with human engineering factors for manhandling and transportation limitations.

To meet the required throughput requirements, the most efficient aluminum pipeline will have a nominal diameter of 8 inches and a wall thickness of approximately 0.200 inch. The maximum pipe section length consistent with worldwide transportation limitations is 35 feet. The weight of a 35-foot length of 8.625-inch-outside-diameter by 0.200-inch-wall, 6063-T6 aluminum alloy pipe is 204 pounds.

[29] R. C. Winfrey, et al, *Marine Corps Fuel Systems (1975-1985)*, Technical Note N-1243; Naval Civil Engineering Laboratory; Port Hueneme, California; December 1972.

A section of this pipe, including a lightweight coupling, can be handled by 4 men. For sustained pipeline laying operations, it is recommended that the pipe stringing and joining teams include 6 men to handle 1 section of pipe.

A simple, self-latching mechanical coupling of the type represented by the RACEBILT coupling, manufactured by Race and Race, Inc., Winter Haven, Florida, is the preferred pipe-joining technique. The primary advantages of this type of coupling is that it can be assembled in a few seconds without any tools or training. A disadvantage of the coupling is that the V-type gasket provides a seal in only one direction, against internal pressure. If the pressure outside the pipeline exceeds the internal pressure, leakage past the gasket may occur. This precludes using the RACEBILT couplings in tank farm manifolds and in other applications where the pipe may be in the suction manifold for flood-and-transfer pumps. Use of grooved-end pipe and split-ring mechanical coupling is recommended for all manifolds where the pipe may be a part of a suction manifold.

Any required bends in the pipeline can be formed using conventional pipe forming practices. Making field bends at the job site can be time consuming, particularly if the proper equipment is not readily available. As an alternative, it is recommended that pipe-laying crews be furnished a variety of prefabricated bends of 11, 22½, 45, and 90 degrees to be installed in the pipeline where needed.

Rising fuel costs are continually increasing the cost advantage of using high-speed, medium-duty diesel engines to power all flood-and-transfer and pipeline booster pumps. If possible, pump units should use diesel engines that are common to other high-density items of equipment to reduce logistical support requirements. The potential for improved mission reliability through reduced administrative down time further supports pump units sharing engines with other high-density items of equipment.

Maximum pipeline mission reliability at the lowest cost is achieved using two or more pump units operating in series at each booster pump station. A standby booster pump unit is required at the first booster pump station in each pipeline to maintain an adequate flow rate through the first segment of the pipeline. Improvement in mission reliability resulting from standby pump units at other than the first booster pump station does not merit the additional cost where the pipeline throughput capacity will allow sufficient downtime to perform scheduled maintenance.

All pipeline equipment should be designed with adequate controls and protection devices for safe operation using the tight-line method of pipeline operation which provides the most efficient utilization of personnel and equipment. Fully automated booster pump stations are not practical for Military pipeline operations because

they are too costly and require skilled maintenance personnel.

Pump station manifolds should be prefabricated as modules to eliminate as much on-site assembly work as possible. An improved Military pipeline system should include the following ancillary items:

a. Meters for volumetric measurement of pipeline throughput.

b. Pressure regulation equipment for long down-hill pipeline sections.

c. Product loss reduction equipment providing automatic shutoff due to failure or deliberate rupture of a pipeline.

As noted at the beginning of this chapter, a pipeline is a subsystem of a much larger theater bulk-fuel distribution system. The success of any pipeline in satisfying its assigned mission is dependent on other elements of the total distribution system. Specifically, there must be a constant supply of fuel to the pipeline and adequate storage capacity to receive the pipeline throughput. The current Military capability is deficient in both of these areas.

The problems associated with supplying fuel to a pipeline are examined in Appendix A of this report. This analysis identifies the need for development of an improved tanker mooring and discharge system. More advanced moorings, probably of a single-point type, capable of restraining larger tankers under more severe seastate conditions are required. More important, the tanker discharge capability must be expanded to provide higher flow rates from tankers moored farther off the coastline.

The large-capacity, collapsible, self-supporting fuel-storage tanks now under development at MERADCOM will significantly improve the Army's bulk fuel storage capability. However, a detailed engineering analysis of the entire theater-of-operations requirements for bulk fuel storage is needed to insure that existing and future fuel storage facilities are compatible with the remainder of the theater bulk fuel distribution system.

## V. CONCLUSIONS

16. Conclusions. It is concluded that:

a. The operational effectiveness of Military petroleum pipelines can be improved significantly by using aluminum pipe joined by self-latching mechanical couplings (RACEBILT Industrial fittings or equivalent) in lieu of the present Military standard, lightweight steel, grooved-end pipe and split-ring couplings.

b. All flood-and-transfer and pipeline booster pumps should be powered by high-speed, medium-duty diesel engines that are common to other high-density items of Military equipment.

c. The maximum pipeline reliability at the lowest cost can be achieved using two pumps operating in series at each booster station.

d. The tight-line method of pipeline operation should be employed to achieve the most efficient use of personnel and equipment.

e. Flexible hoselines are not practical as a means for transporting large quantities of fuel except in support of assault operations where flexibility, high mobility, rapid deployment and recovery, and frequent relocation are essential mission requirements.

f. Existing tanker mooring and discharge facilities are not capable of transferring fuel from vessels moored offshore to marine terminals at rates which will maintain a constant supply of fuel to pipeline systems satisfying projected combat support requirements.

## REFERENCES


1. R. Stanley LaValee et al; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase I: 1970-1975, Main Report.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1968.

2. Edward W. King; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase I: 1970-1975, Annex A, Historical and Doctrinal Review.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

3. R. Dean George et al; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase I: 1970-1985, Annex B, Part I: Military Equipment Survey.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

4. R. Dean George et al; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Annex B, Part II, Industry Equipment Survey.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

5. Ray A. Anderson; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase I: 1970-1975, Annex C, Pipeline Simulation Model.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

6. Ray A. Anderson et al; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase I: 1970-1975, Annex E, Cost Effectiveness Analysis.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

7. Gordon B. Page and Richard A. Turker; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase I: 1970-1975, Annex F, Engineer Organization and Equipment.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

8. R. Stanley LaValee and Kenneth R. Simmons; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase I: 1970-1975, Annex G, Synthesized Engineer Bulk Petroleum Facilities System.* Combat Operations Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

9. John M. McCreary et al; *Bulk Petroleum Facilities and Systems (BPFS) – 1970-1985, Phase II: 1975-1985.* Combat Operations Research Group, Technical Operations, Inc.; Alexandria, Virginia; November 1969.

## REFERENCES (Cont'd)


10. A. J. Stepanoff; *Centrifugal and Axial Flow Pumps,* 2nd ed. New York: John Wiley and Sons, Inc.; 1957.

11. *Comparative Analysis of Selected Gas Turbine and Diesel Engines.* Report No. 1080. Naval Ship Systems Command; Washington, D.C.; April 1969.

12. R. C. Winfrey et al; *Marine Corps Fuel Systems (1975-1985).* Naval Civil Engineering Laboratory; Port Hueneme, California; December 1972.

13. *Cost Reference Guide for Construction Equipment.* National Research and Approval Company, Equipment Guide-Book Company; 1975.

14. *Investigation of Pipeline Concepts, Materials and Construction Techniques, Final Report.* Value Engineering Company; Alexandria, Virginia; August 1976.

15. H. N. Johnston; *Portable Pressure-Regulation Station for Critical Downhill Sections of 6-, 8-, and 12-Inch Military Fuel Pipelines.* Report 2017; U.S. Army Mobility Equipment Research and Development Center; Fort Belvoir, Virginia; November 1971.

16. H. N. Johnston; *Potential Methods for Reduction of Product Loss in Military Pipelines.* Report 2034; U.S. Army Mobility Equipment Research and Development Center; Fort Belvoir, Virginia; August 1972.

17. *Introduction to the Oil Pipeline Industry.* Petroleum Extension Service, University of Texas; Austin, Texas; 1966.

18. *Oil Pipeline Pumping Station Operation.* Petroleum Extension Service, University of Texas; Austin Texas; 1956.

19. *Oil Pipeline Construction and Maintenance.* 2nd ed. Petroleum Extension Service, University of Texas; Austin, Texas; April 1973.

20. *Military Petroleum Pipeline Systems.* Headquarters, Department of the Army; Washington, D.C.; TM5-343; February 1969.

21. *Petroleum Supply in Theaters of Operations.* Headquarters, Department of the Army; Washington, D.C.; FM 10-67; October 1968.

## APPENDIX A

## TANKER MOORING AND DISCHARGE SYSTEMS

In the preceding pages, a pipeline system employed as an element of a larger bulk fuel distribution system has been examined in detail as the link between using units and source of bulk fuel supply. Existence of that source has been implicitly assumed; however, its availability is contingent upon the sequential interaction of ocean-going tankers, means to transfer fuel from tankers to a shore-based marine terminal complex, and means to store large volumes of fuel at the marine terminal. That fuel would eventually be transferred forward to using units through pipelines, hoselines, railcars, tank trucks, or a combination of such conduits and vehicles. It must be noted that each conveyance means cited is useful *only* if there is at least as much fuel available at the source as the volume planned to be conveyed forward. For example, if it is planned to pump 10,000 barrels of fuel through an overland pipeline on a given day, the assemblage of elements on the outlet side of the pipeline's first pump station will perform well only if that amount (10,000 barrels) of fuel is available at the marine terminal upstream from that pump station. Thus, the tactical commander will be serviced adequately only if a complete bulk fuel distribution system which extends from tanker to front-line tactical vehicle fuel tank is provided. The relationship of the various elements will become obvious if the supply, demand, and fuel reserve are addressed briefly before proceeding further.

The fuel reserve at any instant is simply the difference between the cumulative volume of fuel delivered and the cumulative volume of fuel consumed. That difference can only be non-negative, since once the fuel reserve is reduced to zero there may be no further deliveries to using units, and for that reason no further decrease in the fuel reserve. The volume of fuel consumed must be loosely construed to include fuel actually consumed in vehicles, stationary equipment, and aircraft, plus fuel lost through leakage, sabotage, and pilferage.

The fuel reserve will normally be stored in the marine terminal and in the forward corps areas. Its level will vary in response to discontinuous changes in fuel delivery and fuel consumption rates. The overland delivery means and that portion of the fuel reserve maintained by the using units in the forward corps area must be capable of accommodating the inevitable fluctuations in demand. The higher the potential throughput and the reliability of the delivery means, the less important the forward area fuel reserve becomes. Conversely, if the delivery means is of low potential throughput or low reliability, the larger the forward area fuel reserve objective must be. At the beginning of the pipeline, the fuel reserve levels within the marine terminal will

vary in response to two factors. The first is the level of combat intensity as manifested in fuel consumption rates. The second factor is a group of uncontrollable variables which collectively dictate whether or not fuel may be delivered to the marine terminal. The principal variables are: (1) availability of tankers which have the required refined petroleum products on board; (2) availability of existing tanker mooring and discharge facilities or coastline characteristics which permit a tanker to approach sufficiently close to the shoreline to permit the use of Military mooring and discharge facilities; (3) near-shore current velocities, sea conditions, and climatology within the operating envelop of the mooring and discharge system; and (4) reliability of the tanker's pumps and mooring and the means used to transfer fuel from tanker to shore (i.e., pipeline, hoseline, or shuttle craft).

The world's coastlines vary substantially in terms of their suitability for near-shore tanker operations. A given coastline may have numerous, few, or no harbors, and those harbors that exist may be either natural or artificial. Even when harbors do exist, their availability is subject to approval of the host nation. If harbors are available, the tactical commander must decide if their convenience is worth the risk which fuel discharge operations pose to other facilities within the harbor area. Turning to the more demanding situation where use of existing harbors has been rejected for some reason, the planner is faced with a fourfold problem: a site must be located which is compatible with Military mooring and discharge systems; a tanker with the required types and volume of fuel must be available; the environmental conditions prevailing at the time the transfer of fuel is to take place must be within the design operating envelopment of the mooring and discharge system; and the system must be functional in a mechanical sense. All four conditions must be satisfied before fuel may be transferred from sea to shore. If the first two conditions have not been satisfied, the tactical commander is left to rely on fuel flown in by aircraft (bladder bird and wet wing); if the two latter conditions have not been satisfied, he may draw on the fuel reserve stored in the marine terminal until the unfavorable conditions subside or the mechanical failure is repaired.

The generic problem of delivering fuel from tanker to marine terminal has been examined in detail in two prior works, the conclusions of which will be summarized below. The reader is referred to the original works[A-1, A-2] for additional information regarding the sources of data and the study methodology used in reaching the conclusions which are:

---

[A-1] F. Cevasco, Multi-Leg Tanker Mooring System and Unloading Facility; System Model and Reliability Analysis, U.S. Army Mobility Equipment Research and Development Command, Fort Belvoir, Virginia, January 1976.

[A-2] F. Cevasco, Coastal Characteristics and Their Affect on Tanker Discharge Operations: A Preliminary Investigation, U.S. Army Mobility Equipment Research and Development Command, Fort Belvoir, Virginia, Pending Publication.

a. As tankers are moored farther offshore, a mooring and discharge system would have to embody increasingly higher mission reliability values if the same level of performance is to be maintained, all other things being equal.

b. Accumulation of a fuel reserve is absolutely essential if numerous weather-induced fuel interruptions are to be avoided.

c. A greater number of tankers, moorings, and unloading lines is required during the first 30 days of a hostility than during the post-day 30 period. This occurs since the fuel consumption plus a contribution to the fuel reserve must be accommodated during the first 30 days, while only consumption must be accommodated afterward.

d. The discharge means connecting the tanker and marine terminal is expected to constitute the limiting bottleneck in virtually any mooring and discharge system used by the Military. While it generally will never be feasible to discharge fuel at a rate even approaching the volumetric capacity of a tanker's pumps, the problem could be ameliorated somewhat by: (1) use of multiple discharge units with each mooring; (2) reducing pipeline friction by application of an internal coating to the unloading line or use of friction reducing fuel additives (both could be thought of as decreasing the roughness coefficient and thereby increasing the flow rate); and (3) use of offshore pumping stations to increase flow rate. The use of multiple conveyance units and internal coatings appear to be the more feasible of the possibilities presented.

e. Weather will periodically and predictively prevent a tanker from initially mooring or from remaining in a mooring; weather factors, therefore, influence the volume of fuel which may be discharged. The degree of influence will vary both from site-to-site and as a function of the month during which operations take place. The current mooring and discharge system (multileg tanker mooring system) has a limitation of seastate 2 or less. Thus, this system would be available only 40 percent of the time during the worst month of the year on a worldwide average and 70 percent on an annual average basis. While this problem may not be overcome totally in any reasonable manner, development of a second-generation mooring system capable of restraining tankers in seastates beyond the seastate 2 limitation of the current system would at least diminish the problem.

f. The current system may only service tankers moored within 5,000 feet of the shore. This implies that the smallest tankers within the Military Sealift Command (MSC) fleet may be safely moored and discharged only 47 percent of the time off coastlines which are otherwise suitable. Attention should be given to developing a second generation unloading line which may be placed further offshore than the current line.

g. Since the 25,000-DWT size tanker is the smallest within the MSC fleet (it is also the largest which the current system may handle), attention should be given to developing a second-generation mooring capable of safely accommodating tankers larger than the 25,000-DWT size.

h. The current system is usable only in locations where the current velocity is 1 knot or less. Only 56 percent of worldwide landing beaches fall in that category.

i. The explosive-embedment anchor development effort consisted largely of innovation rather than of deliberate application of theoretical research findings. While the anchor was subsequently proven to be a useful device, further improvement must await the theoretical findings which a basic and exploratory research effort would be expected to unearth. This problem is further exacerbated by ignorance of the mooring load/time history which the anchors must resist.

j. The probability of delivering fuel from a vessel positioned off a randomly selected landing beach on a randomly selected day using the current mooring and discharge system is relatively low (i.e., the current system lacks universality). While total universality is not attainable in a pragmatic sense, an advanced mooring and discharge system would do much to elevate the degree to which universality is approached. The reader is cautioned that the comparisons to be presented artificially, inflate the current system's worth -- the current mooring will only accommodate a fraction of the MSC tanker fleet while an advanced mooring would conceivably accommodate the entire fleet. The present and advanced discharge system operating envelopes are given in Table A-1.

Table A-1. Operating Envelope Parameters for Present and Advanced Tanker Mooring and Discharge Systems

| Parameter | System Capability: Present | System Capability: Target |
|---|---|---|
| Seastate | 2 | 3 |
| Current Velocity | 1 knot | 2 knots |
| Conduit Length | 5,000 feet | 10,000 feet |
| Tanker Size | 25,000 DWT | 38,000 DWT |

The above capabilities may be transformed into measures of utility by means of methodology developed elsewhere to obtain the probabilities presented in Table A-2.

Table A-2. Probabilities of Being Able to Transfer Fuel for Present and Advanced Mooring and Discharge Systems[a]

| Parameter[b] | Probability of Being Able to Transfer Fuel, Given: Present System | Advanced System |
|---|---|---|
| p[c] | 0.18 – 0.47 | 0.53 – 0.71 |
| p[d] | 0.70 | 0.85 |

[a] The present and advanced systems addressed embody the operating capabilities cited in the preceding Table as present and target, respectively.

[b] For annual occurrence rates of seastates above the upper operational limit, i.e., seastate 2 for the present system and seastate 3 for the advanced system (see the preceding table).

[c] Probability of delivering fuel from a vessel positioned off a randomly selected landing beach on a randomly selected day.

[d] Probability of delivering fuel from a vessel on a randomly selected day utilizing a present or advanced system which has been installed offshore from the objective area.

In summary, the present tanker mooring and discharge system would be responsive to the tactical commander's needs between 18 and 47 percent of the time, while an advanced system would increase this value to between 53 and 71 percent; the latter value jumps to 85 percent once a site has been selected for the system. The present system's indicated utility would be even less if the probabilities in Table A-2 were adjusted downward by a factor corresponding to the percentage of the MSC tanker fleet less than or equal to the current system's limit of 25,000 deadweight tons (less than half the MSC fleet). The 1.5 fold to 3 fold increase in potential coverage of worldwide landing beaches associated with the advanced system would reduce initial site selection constraints substantially. An enhanced seastate tolerance would increase the hypothetical advanced system's usefulness by approximating 150 percent during the worst month of the year and by a lesser 120 percent on an annual basis. An advanced discharge means with a potential throughput double that of the present would halve the number of systems required to support a given magnitude hostility, freeing personnel and equipment for other tasks. While the advanced system's configuration may not be accurately predicted at this time, it is known that the mooring component would undoubtedly be of a single-point type.

## APPENDIX B

# NSIA TRADE-OFF TECHNIQUE

A trade-off technique is a method, procedure, or device used as an aid in decision making. The purpose of trade-offs is to "weigh" two or more alternatives or choices in an objective and systematic manner so as to increase the probability of arriving at a correct decision.

One of the best known trade-off techniques was developed by the National Security Industrial Association (NSIA). This technique involves breaking a complex problem down into a number of smaller problems, the successive solutions of which lead directly to solution of the basic problem. The goal is to objectively express each element of a problem in numerical terms for use in substantiating the optimum decision.

The NSIA technique, when applied objectively, provides reasonable accuracy in decision making without requiring the excessive amounts of time and manpower which often preclude the application of more sophisticated techniques. The principal disadvantage of the NSIA technique is that it does not require examination of all lower order parameters which may impact on the final outcome. Despite this weakness, the NSIA technique is vastly superior to any qualitative judgment of the relative merits of several alternative courses of action.

The evaluator of the effect of a particular alternative should include in the evaluation all aspects of the problem that would possibly be involved. When this is done by trade-offs, it is possible to refine the balance of the favorable and unfavorable effects of each alternative on the overall problem. The total effect of each alternative is expressed as a numerical value and can be incorporated with similar overall measures of the effect on the total problem. The final result obtained becomes an objective basis for judging the desirability of adopting the alternatives that have been so analyzed.

The NSIA trade-off technique produces positive or negative numerical values for the possible effects of a particular parameter (or a change therein) on all the characteristics and other features of a system. As such, it represents an evaluation of the system from one particular point of interest. The evaluator uses numerical values from +1 to +100 for estimated favorable effects and values from -1 to -100 for those found to be unfavorable. An estimate of either +100 or -100 would override all other considerations.

Several precautions should be taken in applying this technique. Evaluation should be made only by individuals fully qualified in the area of the system characteristic being studied. Second, whenever possible, a given evaluation should be made independently by two or more such experts, with the algebraic average of all to be used. Finally, all possible effects of a given alternative should be considered. When this has been done for all the alternatives that have been proposed, a reasonably clear and rather conclusive indication is obtained of the degree of desirability of each. It is evident that every effort must be made to describe clearly and completely any alternative that is proposed so that all the evaluators obtain a uniform and accurate understanding of that alternative.

Procedures for applying the NSIA technique:

(1) Define the problem to be solved clearly and concisely.

(2) List all the alternatives that can be considered as possible solutions to this problem.

(3) For each such alternative, obtain or prepare drawings, schematics, and other materials that define it clearly.

(4) For each alternative, prepare a data sheet similar to the one shown as Figure B1.

NOTE: From this point, this procedure relates solely to the steps taken for one of the alternatives being studied by trade-off.

(5) Determine all of the parameters, such as reliability, safety, cost, and schedule, that could be affected if this alternative were adopted. Enter these by number in the appropriate column of the data sheet for this alternative. Enter special information of significance about any of these characteristics in the column headed "Considerations."

(6) For each characteristic entered in the "Parameters" column, establish and enter in the "Relative Weighting" column a suitable weighting value that represents the relative importance of each characteristic to the system. A value of unity should be assigned to the least important characteristics, with appropriate whole-number values given the others, according to their importance. For example, if the effect on schedule were considered least important, it would be given the factor of 1, and if Safety were considered to be twice as important, it would be weighted by a factor of 2. In some instances, fractional weighting values can be used.

| Parameters | Considerations | Relative Weighting | Basic Rating | | Adjusted Values | |
|---|---|---|---|---|---|---|
| | | | Undesirable | Desirable | Undesirable | Desirable |
| | | | | | | |

**TOTAL**
**Net Adjusted Value =**
**Average Net Value =**

**Figure B1. Sample Evaluation Sheet.**

(7) Evaluation of each alternative in relation to each system characteristic or other parameter should then be made by the individual or group best qualified to judge its desirability. For example, the reliability group would evaluate the feature from the viewpoint of its effect on subassembly or system reliability; the human factors group would do the same from the human engineering viewpoint. Whenever possible, a number of independent evaluations should be made. In every instance, however, utmost care must be taken that each characteristic associated with an alternative is evaluated in isolation, never as influenced by other characteristics. Each evaluator, having made his evaluation, assigns to his findings an appropriate positive or negative number to indicate the degree of desirability or undesirability that has been determined. (See the scale of numerical values given in Figure B-2.) If several evaluations have been made of the alternative in relation to a single system characteristic, the algebraic average of the group is computed and entered, as either undesirable or desirable, in the "Basic Rating" column.

| | |
|---|---|
| +100 | Necessary |
| + 90 | |
| + 80 | |
| + 70 | Very Desirable |
| + 60 | |
| + 50 | |
| + 40 | Desirable |
| + 30 | |
| + 20 | |
| + 10 | |
| 0 | No Effect |
| - 10 | |
| - 20 | |
| - 30 | Undesirable |
| - 40 | |
| - 50 | |
| - 60 | Very Undesirable |
| - 70 | |
| - 80 | |
| - 90 | |
| -100 | Unacceptable |

**Figure B-2. Basic rating scale.**

(8) Multiply the assigned value in the "Basic Rating" column by its corresponding weighting factor, and enter the product, as either undersirable or desirable, in the "Adjusted Values" column.

(9) Having done this for each of the system characteristics or other parameters selected as significant for this alternative, add algebraically all the values entered in the "Adjusted Values" column, establishing thereby a total net value for the alternative.

(10) Obtain a total weighting factor for this design feature by adding all weighting values entered on the data sheet.

(11) To determine an average net value for the design features, divide the total net value by the total weighting factor. The resulting algebraic sign (plus or minus) will indicate whether this alternative is desirable or undesirable, and its absolute value will measure the degree of its desirability or undesirability. The average net value thus determined is the figure of merit for this particular alternative.

When this technique has been applied to all the alternatives under consideration, the average net value determined for each will provide an optimum solution of this particular problem.

## APPENDIX C

### COMPANIES MENTIONED IN BPFS STUDY

Aerojet-General Corporation (AOMC)
9236 East Hall Road
Downey, California 90241

Aeroquip
Gustin-Bacon Division
Post Office Box 927
Lawrence, Kansas 66044

Amercoat Corporation
Ameron Corrosion Control Division
Brea, California 92621

Anbeck Company
Post Office Box 19415
Houston, Texas 77024
(See Zapata)

CIBA-GEIGY Corporation
Pipe Systems Department
9900-T Northwest Freeway
Houston, Texas 77018

CIBA Products Company
556 Morris Avenue
Summit, New Jersey 07901
(See CIBA-GEIGY)

CRC-Crose International, Inc.
Post Office Box 3227
Houston, Texas 77024

Frieberg and Fonnsbeck Associates
Post Office Box 2127
Fullerton, California 92633

Gustin-Bacon Division
Certain-Teed Products Corporation
Post Office Box 15079-S
Kansas City, Kansas 66115
(See Aeroquip) For Overhaul[c]

Mobile Pipe Constructors, Inc.
16 Edgewater Drive
Belvedere, California 94920

Mohr, Glen
Post Office Box 52
Linthicum, Maryland 21090
(See Mobile Pipe Constructors)

Race and Race, Incorporated
Post Office Box 1400
Winter Haven, Florida 33880

Reynolds Aluminum Company
Post Office Box 27003-ZA
Richmond, Virginia 23261

Rockwell International
North American Aviation Group
1700 East Imperial Highway
El Segundo, California 90245

Smith, A. O., Corporation
Reinforced Plastics Division
2700 West 65th Street
Little Rock, Arkansas 72209

Victaulic Company of America
3102 Hamilton Boulevard
South Plainfield, New Jersey 07080

Westinghouse Electric Corporation
Industrial Equipment Division
Post Office Box 300
Sykesville, Maryland 21784

Zapata Pipeline Technology, Inc.
2521 Fairway Park Drive
Suite 420
Houston, Texas 77018

# APPENDIX D

## COST ESTIMATING GUIDANCE
## TRANSPORTATION COSTS

**1. Basic Factors.**

a. **Budget Factors.** (Source: Mrs. June Stacey, Prog & Budget Div, S&M Div, ODCSLOG, 26 Jun 75). These figures represent summary rates for all cargo, as reflected in the FY 76 Second Destination Transportation budget.

| | Budget Factors | Converted to S TON |
|---|---|---|
| CONUS Line Haul | \$47.88/S TON | \$47.88 |
| CONUS Port Handling | 13.02/M TON | 32.55 |
| Mil Sealift Cmd | 61.81/M TON | 154.53 |
| O/S Port Handling | 5.90/M TON | 14.75 |
| O/S Line Haul | 10.93/S TON | 10.93 |

b. **Conversion Factors.**

1 M TON = 40 $ft^3$
1 S TON = 100 $ft^3$
2.5 M TON = 1 S TON (General Cargo)
1 M TON = 1 S TON (Ammunition)

c. **Packing and Crating Weights.** Guidance has been requested from ODCSLOG. In the interim, a factor of 10 percent will be added for general cargo and ammunition only.

**2. Computations.** (Source: RAC Study: Selected Uniform Cost Factors; A Manual for the Army Materiel Command, Jun 72).

a. Determine weight of equipment to be transported in terms of S TONS. Vehicles and large volume items should be computed from volume (cube); general cargo and ammunition, directly from weight. Source reference for Military vehicles and selected organizational equipment currently in the inventory is TB 55-46-2.

b. Add weight of packing and crating for general cargo and ammunition.

c. Apply the following composite factors to total tonnage (S TON):

| | 1st Dest. | 2nd Destination | |
|---|---|---|---|
| | | To User | For Overhaul[c] |
| Total Tonnage | \$47.88 | | |
| Inventory Positioned in CONUS[a] | | \$47.88 | \$95.76[d] |
| Inventory Positioned O/S[a] | | \$271.57[b] | \$543.14[d] |

[a] If distribution unknown, assume 50-50.

[b] Sum of all factors, plus double weighting of O/S line haul because of intermediate back-up depot.

[c] This is transportation cost for each overhaul. Multiply by number of overhauls as determined in calculation of depot overhaul costs.

[d] Twice one-way transportation cost to user.

3. **Models.**

a. **First Destination Transportation.**

$$\left.\begin{matrix}(\text{S TON} \times 1.1)\\ \text{or}\\ (\text{ft}^3/100)\end{matrix}\right\} \times \ \$47.88$$

b. **Second Destination Transportation.**

$$\left.\begin{matrix}(\text{S TON} \times 1.1)\\ \text{or}\\ (\text{ft}^3/100)\end{matrix}\right\} \times \ \left[(\%\ \text{Conus} \times \$47.88) + (\%\ \text{O/S} \times \$271.57)\right]$$

c. **Transportation for Overhaul.**

Second Dest x 2 x No. of Overhauls
Trans Costs per Unit

4. **Rationale.** Should include:

a. The models.

b. The statement that: Cost factors were obtained from Program and Budget Division, S&M Directorate, ODCSLOG. Cost models were derived from RAC Study: Selected Uniform Cost Factors; A Manual for the Army Materiel Command, Jun 72.

## APPENDIX E

## DESIGN OF ALTERNATIVE PIPELINE SYSTEMS

Assessment of the cost and operational effectiveness of a pipeline concept requires definition of the principal hydraulic design characteristics of the actual system to be evaluated. The design procedure for each alternative pipeline concept evaluated in this report is summarized in this appendix.

Military pipeline design criteria states the throughput of different types of fuels to be pumped must be considered, and the heaviest fuel making up 24 percent or more of the total is to be taken as the design fuel.[E-1] Diesel fuel is the heaviest of all fuels likely to be pumped through Military pipelines. The evaluation criteria established in paragraph 5 of the basic report states diesel fuel represents 30 percent of the total throughput. Therefore, all pipeline design calculations are based on diesel fuel at 60°F having a 0.8448 specific gravity[E-2] and a kinematic viscosity of 3.85 centistokes.[E-3]

The friction head, or loss of head, due to fuel flowing through the pipeline is computed using the Darcy-Weisbach equation and resistance coefficients from Figure 40 of the basic report.

From Table 2 of the report, the maximum daily througput requirement for Scenario I is 27,620 barrels per day. A design rate of 950 gal/min is selected for Scenario I. This flow rate will allow the maximum daily throughput requirement to be delivered in approximately 20 hours of operation.

For Scenario II, the design rate of flow is specified as 35,000 barrels in 23 hours of operation. This is equivalent to a throughput of 1,065 gal/min.

a. **Alternative I.** The pipeline is constructed using fiberglass-reinforced epoxy resin pipe with PRONTO-LOCK mechanical joints manufactured by CIBA-GEIGY Corporation. The maximum safe working pressure for an 8-inch-diameter pipeline is 150 lb/in$^2$. Assuming 20 lb/in$^2$ suction pressure is required at the pump inlet, the pressure loss between pump stations cannot exceed (150 - 20) = 130 lb/in$^2$.

---

E-1 Department of the Army Technical Manual, *Military Petroleum Pipeline Systems*, TM5-343, February 1969, p. 6-1.

E-2 *Ibid*, p. 6-2.

E-3 *Ibid*, p. C-4.

(1) **Scenario I.** For diesel fuel having a specific gravity of 0.8448 at 60°F, 130 lb/in² is equal to 355 feet of fuel. At the design rate of flow of 950 gal/min, the fluid friction loss through the pipe is computed to be 71.2 feet per mile. The total dynamic head losses from the marine terminal to the highest point in the pipeline at mile 60[E-4] is (71.2) (60) = 4272 feet of fuel plus 3000 feet increase in elevation (reference Figure 2 of the basic report) or 7272 feet total head. Dividing the total head of 7272 feet for 60 miles of pipeline by the 355 feet maximum total dynamic head per pump station gives a value of 20.5. Thus, 21 booster pump stations are required for the first 60 miles of pipeline. The design head for each pump station will be 7272/21 or 347 feet total dynamic head. The hydraulic gradient shown in Figure E-1 for pump stations 1 through 21 is constructed using these flow characteristics.

In the downhill run between miles 60 and 80, Figure E-1, the slope of the pipeline profile is steeper than the hydraulic gradient. Under these conditions, the static head exceeds the fluid friction losses; therefore, no pump stations are required in this section of the pipeline. The critical pressure in this section of the pipeline occurs under no-flow conditions where the static head must be maintained at or below the maximum safe working pressure of 150 lb/in² or 410 feet of fuel. The total drop in elevation from mile 60 to mile 90 is 3000 - 400 = 2600 feet. Dividing this total static head by the maximum allowable head, a value of 2600/410 = 6.34 is obtained. Thus, 6 pressure regulation stations must be used on the downhill run to prevent over pressurization of the pipeline under static conditions.

When pressure regulation stations are used at locations R1 through R6 as shown in Figure E-1, the resulting static head is shown by the stepped hydraulic gradient. At the design rate of flow, the static head below pressure regulation station R6 will push the fuel to mile 81.3. Four pump stations, each developing a total dynamic head of 311 feet of fuel, are required to push the fuel on to the end of the 100-mile pipeline. This results in the hydraulic profile shown in Figure E-1 for booster pump stations 22 through 25.

The locations for the booster pump stations and pressure regulating stations are listed in Table E-1.

The operating conditions for booster pump stations 1 through 21 of 347 feet total dynamic head and 950 gal/min equate to 83.2 water horsepower. From Figure E-2, a booster pump of this size will have an efficiency of approximately 0.797. The brake horsepower required to drive the pump is 104.4 brake horsepower.

---

E-4 All locations along the pipeline are designated by the distance, in miles, from the first booster pump station; i.e., mile 60 is 60 miles from the first booster pump station.

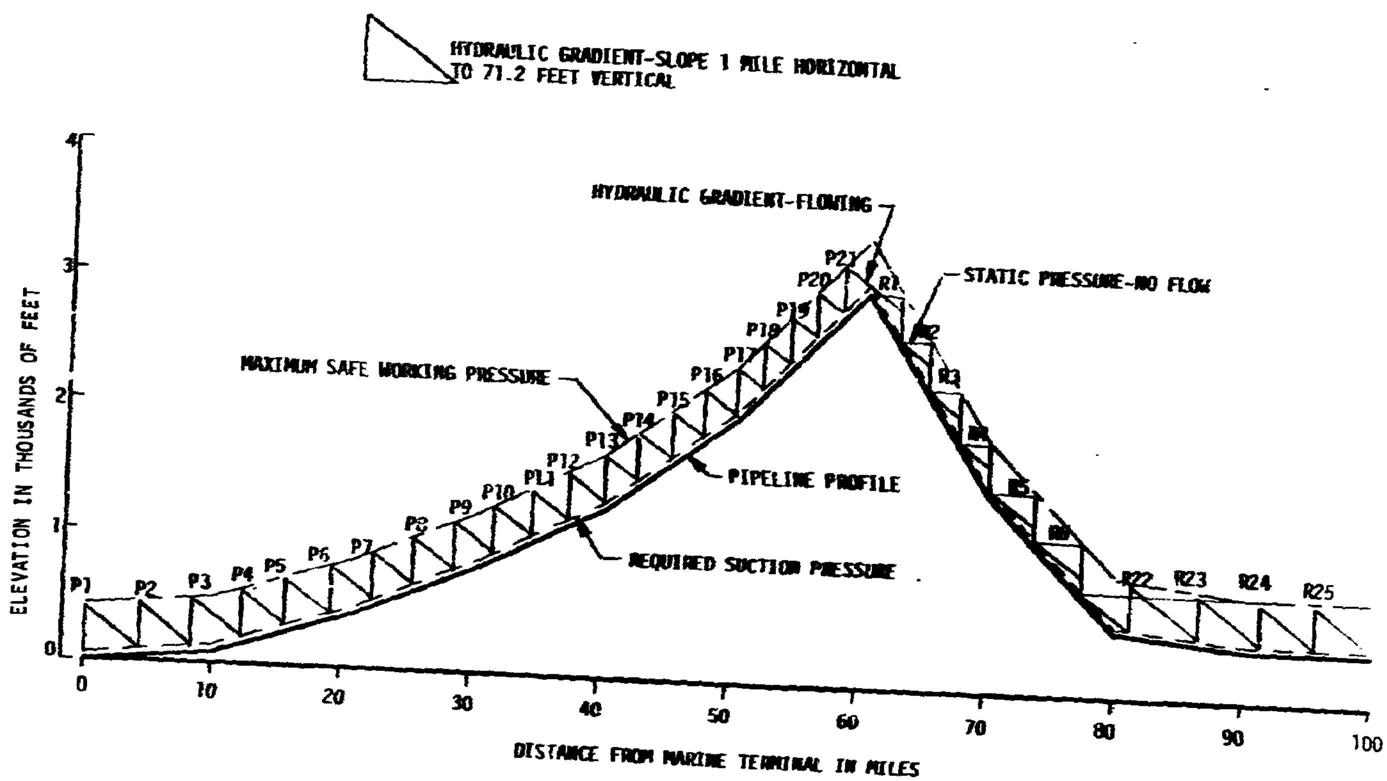



Figure E-1. Hydraulic gradient for Alternative I – Scenario L

Table E-1. Location of Pipeline Booster Pump Stations and Pressure Reduction Stations for Alternative I Scenario I.

| Station | Type | Location* |
|---|---|---|
| P1 | Booster Pump | 0 |
| P2 | Booster Pump | 4.27 |
| P3 | Booster Pump | 8.54 |
| P4 | Booster Pump | 12.26 |
| P5 | Booster Pump | 15.69 |
| P6 | Booster Pump | 19.12 |
| P7 | Booster Pump | 22.32 |
| P8 | Booster Pump | 25.44 |
| P9 | Booster Pump | 28.56 |
| P10 | Booster Pump | 31.54 |
| P11 | Booster Pump | 34.40 |
| P12 | Booster Pump | 37.26 |
| P13 | Booster Pump | 40.11 |
| P14 | Booster Pump | 42.59 |
| P15 | Booster Pump | 45.07 |
| P16 | Booster Pump | 47.55 |
| P17 | Booster Pump | 50.03 |
| P18 | Booster Pump | 52.05 |
| P19 | Booster Pump | 54.07 |
| P20 | Booster Pump | 56.09 |
| P21 | Booster Pump | 58.11 |
| R1 | Pressure Regulating | 62.45 |
| R2 | Pressure Regulating | 64.93 |
| R3 | Pressure Regulating | 67.41 |
| R4 | Pressure Regulating | 69.89 |
| R5 | Pressure Regulating | 73.56 |
| R6 | Pressure Regulating | 77.29 |
| P22 | Booster Pump | 81.30 |
| P23 | Booster Pump | 86.38 |
| P24 | Booster Pump | 91.26 |
| P25 | Booster Pump | 95.63 |

* Location shown as miles from the marine terminal.

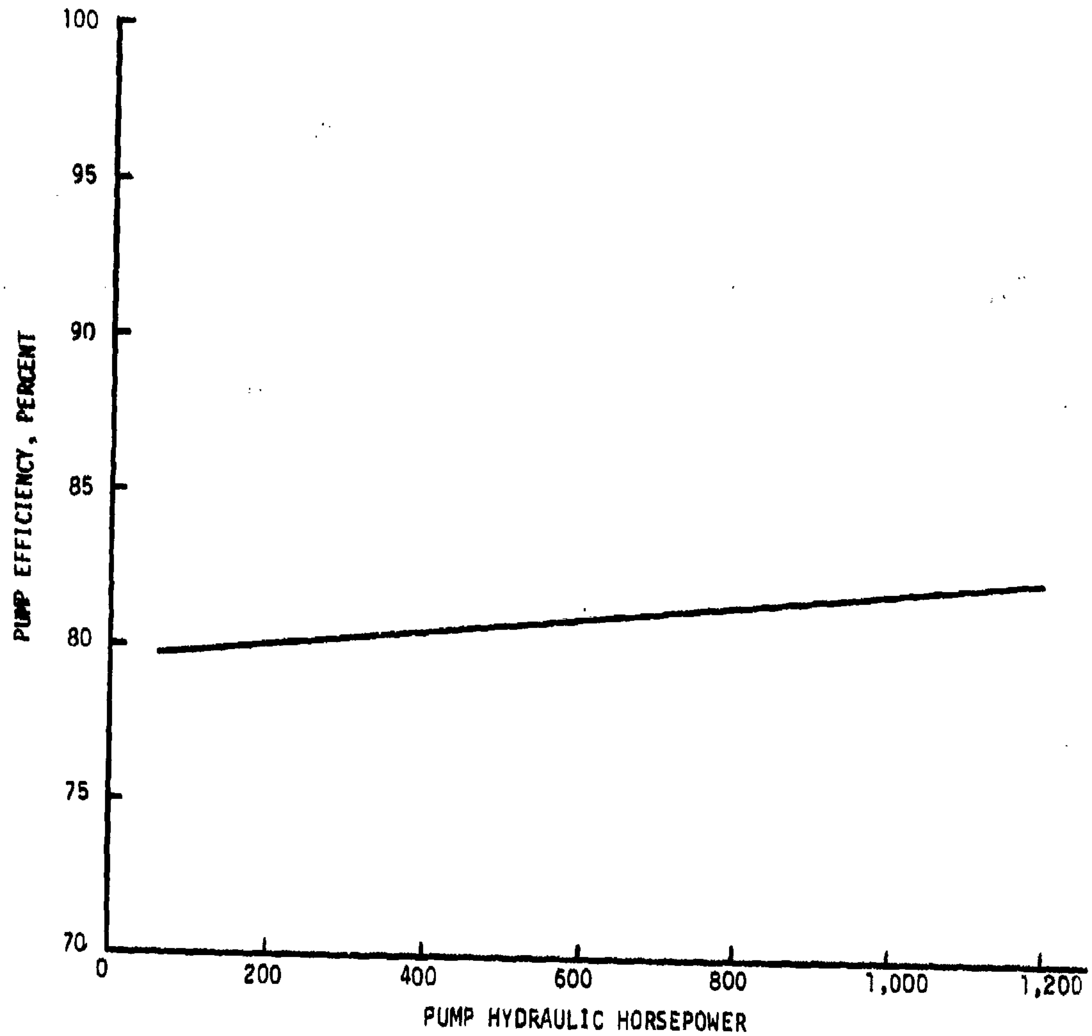



**Figure E-2. Efficiency of pipeline booster pumps.**

(2) **Scenario II.** As in Scenario I, the 130 lb/in$^2$ maximum pressure rise across each pump station equates to a head of 355 feet of fuel. At the specific design throughput of 35,000 barrels in 23 hours, or 1,065 gal/min, the fluid function losses for diesel fuel will be 78.0 feet of fuel per mile of pipeline. Adding the specified 5 feet per mile rise in elevation of the pipeline profile yields a total dynamic head of 83 feet per mile of 8,300 feet over the 100-mile length of the pipeline. Dividing this value by the 355 feet maximum head for each pump station yields a value of 23.88. Using 24 booster pump stations, the total dynamic head of each station is 8300/24 or 346 feet. The resulting hydraulic gradient for the pipeline is shown in Figure E-3 with the pumps located 4.17 miles apart.

Booster pump station design conditions of 1,065 gal/min at 346 feet total dynamic head equal 93.1 water horsepower. Using an efficiency of 0.797 from Figure E-2, 116.8 brake horsepower are required to drive the pump.

b. **Alternative II.** Using schedule 40, 6061-T6 aluminum pipe, this pipeline is installed using aluminum mechanical couplings for grooved-end pipe. The maximum safe working pressure for the 7.981-inch-inside-diameter pipe is limited to 800 lb/in$^2$ by the pressure rating of the couplings. Using 20 lb/in$^2$ suction pressure, the effective pressure loss between pump stations is limited to (800 - 20) = 780 lb/in$^2$. Based on diesel fuel at 60°F having a specific gravity of 0.8448, 780 lb/in$^2$ is equivalent to 2133 feet of fuel.

(1) **Scenario I.** Diesel fuel flowing at the 950 gal/min design rate of flow will produce 82.1 feet per mile fluid friction losses. The total head requirements for the 100-mile pipeline, including 400 feet increase in elevation is (82.1) (100) + 400 = 8610 feet. Four pump stations operating at the maximum safe discharge pressure will develop (4 x 2133) = 8532 feet of head. This is just 78 feet or 19.5 feet per pump station less than necessary to meet the design conditions. It is not practical to increase the number of pump stations from 4 to 5 to obtain this small amount of additional head. Possible alternatives include: (a) increasing the maximum operating pressure by 19.5 feet, which reduces the factor of safety slightly; (b) reducing the suction pressure by 19.5 feet; or (c) reducing the design rate of flow. For this analysis, reducing the design flow rate is assumed to be the best approach.

A total effective head of 8532 feet less 400 feet static head from change in elevation results in 8132 feet of head available to overcome dynamic flow losses. Using the Darcy-Weisbach equation to compute the rate of flow corresponding to a fluid friction loss of (8132/100) = 81.32 feet per mile yields a new design rate of flow of 945 gal/min. The hydraulic gradient for the pipeline system with four pump stations operating at 945 gal/min and 2133 feet of head is shown in Figure E-4.

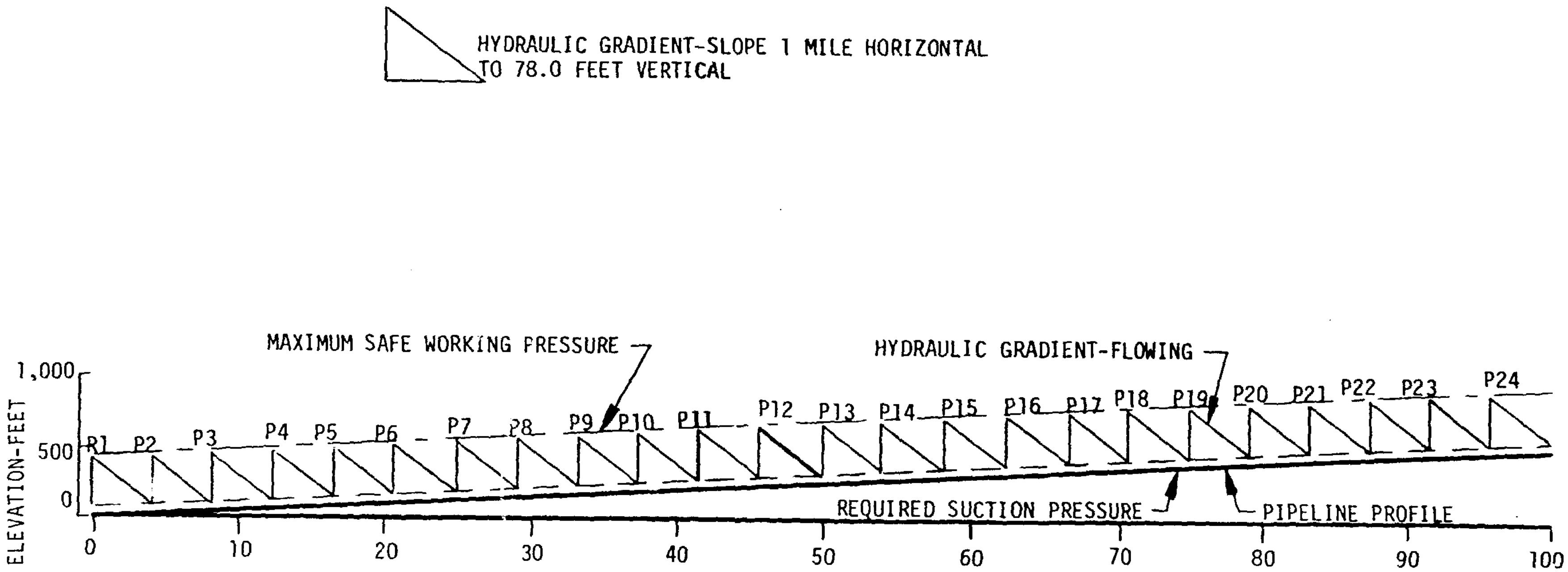



**Figure E-3.** Hydraulic gradient for Alternative I – Scenario II.

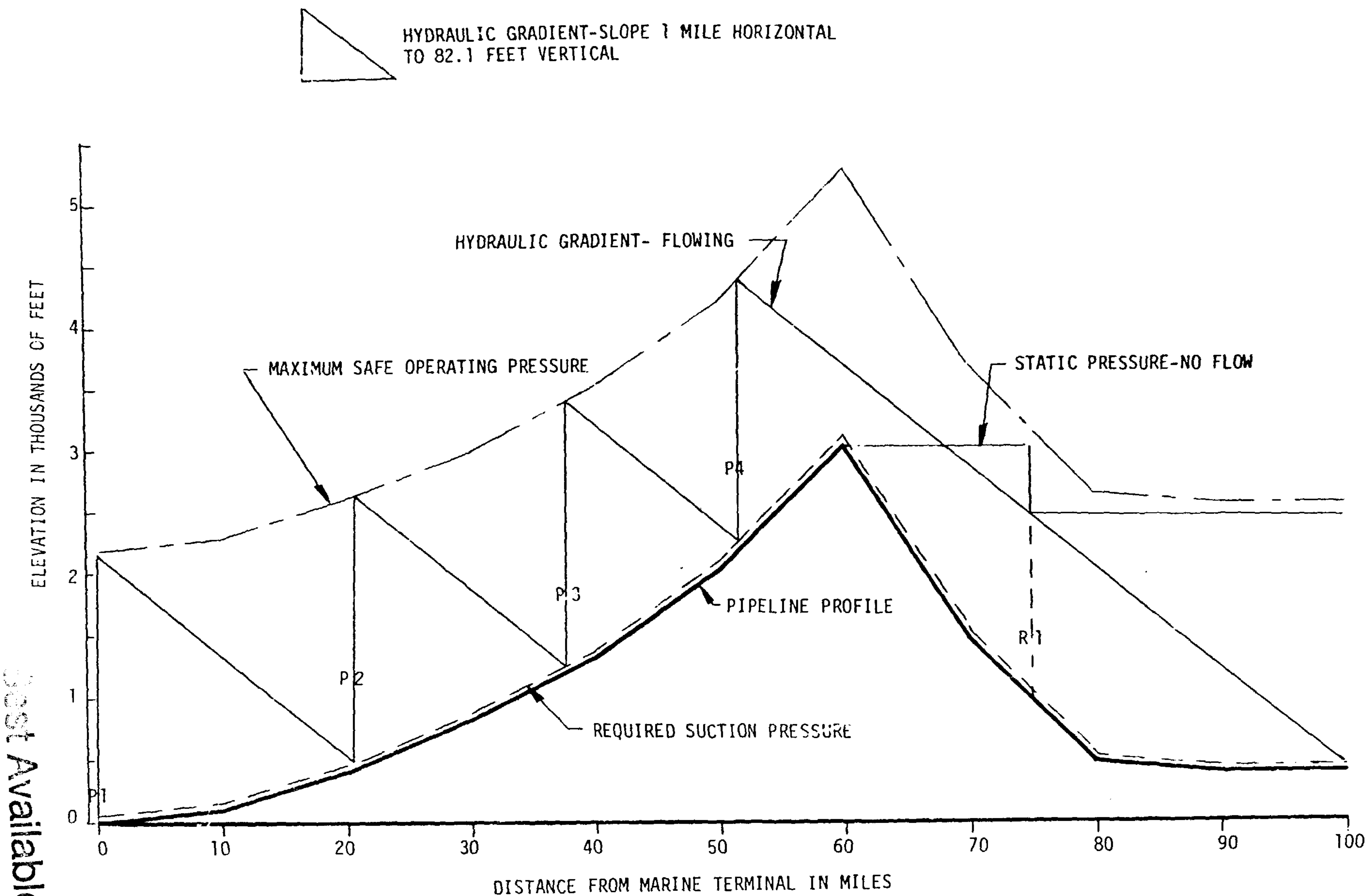



Figure E-4. Hydraulic gradient for Alternative II – Scenario I.

The downhill run from mile 60 to mile 90 presents no problems under dynamic conditions since, as shown in Figure E-4, the hydraulic gradient from pump station P4 to the end of the pipeline is below the curve for the maximum safe working pressure. The pipeline profile falls 2188 feet, equivalent to the maximum safe working pressure, between mile 60 and mile 76.88. Thus, without a pressure regulation station, the pipeline would be overpressured from mile 76.88 to the end (mile 100).

This set of design conditions presents an ideal situation for employing a pressure reducing station. A horizontal static gradient line drawn 2088 feet above the profile at mile 100 intersects the dynamic gradient at mile 75 as shown in Figure E-4. By positioning the pressure reduction station at this point and adjusting the pressure setting to limit the downstream pressure to 1488 feet, the pressure reduction station will not restrict the flow at design flow conditions. Under static conditions the pressure regulation station will limit the downstream pressure to 1488 feet of head. Adding the 600 feet difference in elevation from the pressure regulation station at mile 75 to the lowest section of the pipeline from mile 90 to 100, the maximum static pressure downstream from the pressure regulation station is (1488 + 600) = 2088 feet. The difference in elevation from the highest point on the pipeline at mile 60 to the pressure regulation station at mile 75 is 2,000 feet. This is the highest static pressure in the pipeline above the pressure regulation station occurring at the pressure regulation station inlet.

The locations for the pipeline booster pump stations and the pressure regulation stations are shown in Table E-2.

Table E-2. Location of Pipeline Booster Pump Stations and Pressure Regulation Stations for Alternative II – Scenario 1.

| Station | Type | Location* |
|---|---|---|
| P1 | Booster Pump | 0 |
| P2 | Booster Pump | 20.88 |
| P3 | Booster Pump | 37.82 |
| P4 | Booster Pump | 51.84 |
| R1 | Pressure Regulation | 75 |

* Location shown as miles from the marine terminal.

The booster pump stations operating at 945 gal/min and 2,133 feet total dynamic head develop 509 water horsepower. From Figure E-2, a booster pump of this size would have an efficiency of approximately 0.807. The power required to drive the pump would be 631 brake horsepower.

When turbine-engine-driven pumps are used, one pump unit can handle the entire pumping operation at each booster pump station. The size and weight of a 631-brake horsepower diesel-engine-driven pump unit would exceed Military transportability limits. Thus, two diesel-engine-driven units rated at 315 brake horsepower each would be required at each booster pump station. Units of this size slightly exceed the limit of 304 brake horsepower listed in Table 6 of the basic report. However, this limit is based on the average weight of pump units. By judicious selection of components, design of a 315-brake horsepower diesel-engine-driven pump of acceptable size and weight is possible.

Based on the foregoing, the minimum number of pumps required at each booster station for Alternative II, Scenario I will be one turbine-engine-driven pump or two diesel-engine-driven pumps.

(2) **Scenario II.** The same pressure characteristics used in Scenario I apply. Thus, the maximum operating pressure is 2188 feet of head and the maximum total dynamic head developed at each pump station is limited to 2133 feet. For diesel fuel flowing at the design rate of flow of 1,065 gal/min, the fluid friction losses through the pipe are computed to be 9,632 feet for the 100-mile pipeline. Adding the increase in elevation of 500 feet, the total head requirement at the design rate of flow is 10,132 feet. When five booster pump stations are used, each pump station must develop 10,132/5 = 2026.4 feet of head. Figure E-5 shows the hydraulic gradient for the pipeline with the pump stations located 20 miles apart.

The hydraulic horsepower developed by a pumping station delivering 1,065 gal/min at 2026.4 feet of head is 545 water horsepower. Applying a pump efficiency of 0.808 from Figure E-2, the power required is 675 brake horsepower.

As was the case in Scenario I, one turbine-engine-driven pump can be used at each pump station. Transportability limitations require the use of three 225-brake horsepower diesel-engine-driven pumps at each booster station.

c. **Alternative III.** This alternative uses 6063-T6 aluminum pipe joined by mechanical couplings manufactured by Race and Race, Inc. The maximum safe working pressure recommended by the manufacturer for 8-inch-diameter, 0.150-inch-wall pipe is 359 lb/in$^2$. Again, using 20 lb/in$^2$ suction pressure, the maximum pressure loss between pump stations is (359 - 20) = 339 lb/in$^2$. This maximum working pressure is equal to 982 feet of diesel fuel. The maximum total dynamic head for each pump station is limited to 927 feet with 55 feet suction pressure.

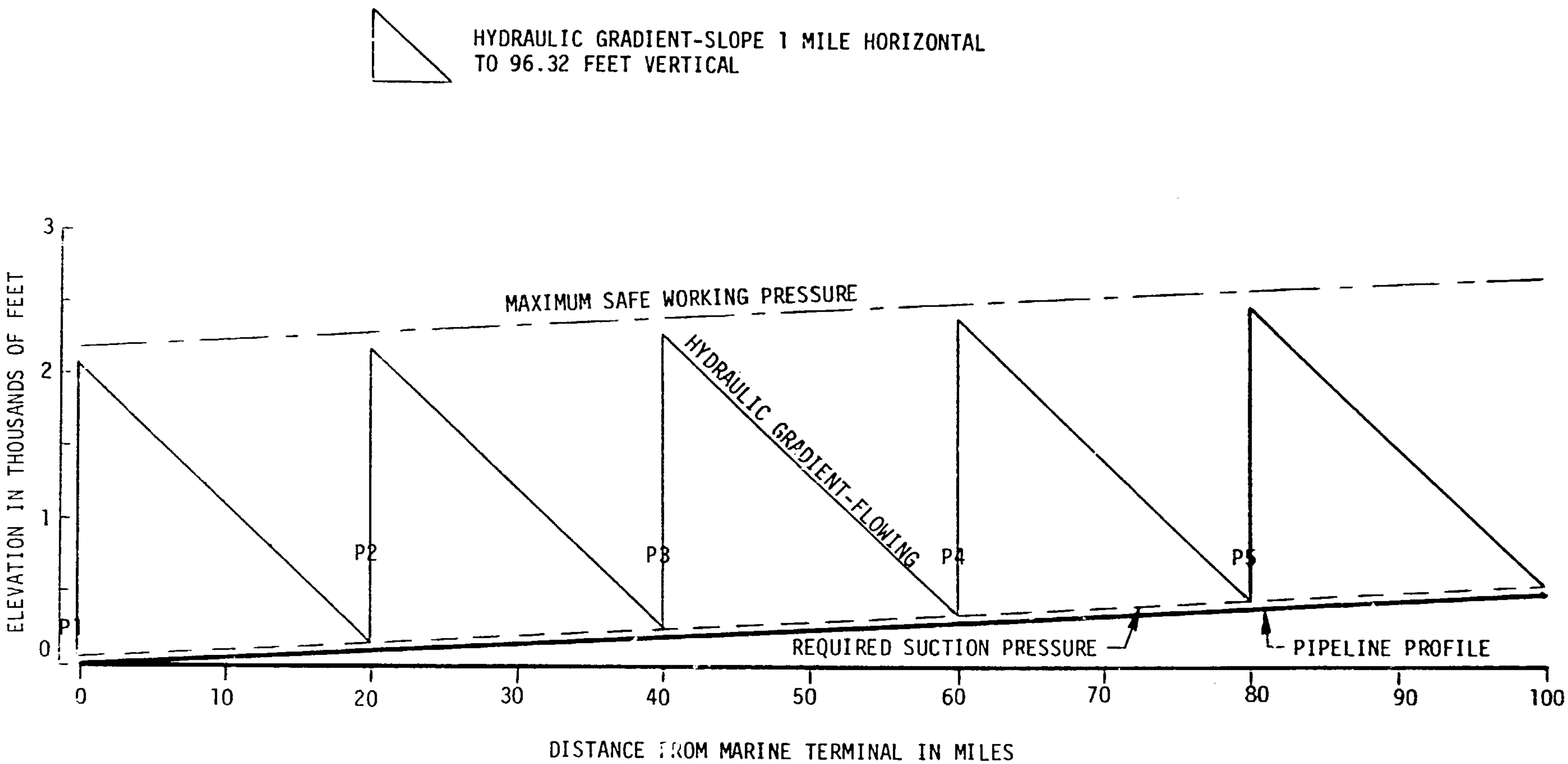



Figure E-5. Hydraulic gradient for Alternative II – Scenario II.

(1) **Scenario I.** The design flow rate of 950 gal/min translates to 67.4 feet per mile fluid friction losses for the 8.239-inch-inside-diameter pipe. The dynamic flow losses from the marine terminal to mile 60 are (60) (67.4) = 4,044 feet. Adding the 3000 feet static head due to change in elevation, the pump stations in the first 60 miles of the pipeline must develop a total head of (4,044 + 3,000) = 7,044 feet. Dividing the total head required by the maximum head per station, a value of (7044/927) = 7.60 is obtained. Therefore, 8 pump stations are required to develop 7,044 feet of head or (7044/8) = 880.5 feet total dynamic head per pump station. Figure E-6 shows the hydraulic gradient for this pipeline design.

On the downhill run from mile 60 to mile 90 both the dynamic and static gradients would exceed the maximum safe working pressure without the use of pressure regulation stations. In this case the pipeline designer has an option on how the line is to be designed. Since the total change in elevation of 2600 feet is less than three times the safe working pressure of the pipe, only two pressure regulation stations would be required to maintain safe static pressure conditions. However, the static head below the last pressure regulation station would not be sufficient to push the fuel all the way to pump station P9 in Figure E-6. This would require two pump stations in the pipeline between mile 80 and mile 100. By using three regulation stations on the downhill run as shown in Figure E-6 only one pump station, P9, is required in this segment of the pipeline. The use of three pressure regulation stations and one pump station is a superior choice over two pressure regulation stations and two pump stations.

The final system design is as illustrated by the hydraulic gradient in Figure E-6. The locations for all booster pump stations and pressure regulation stations are shown in Table E-3.

Table E-3. Location of Pipeline Booster Pump Stations and Pressure Regulation Stations for Alternative III -- Scenario I.

| Station | Type | Location* |
|---|---|---|
| P1 | Booster Pump | 0 |
| P2 | Booster Pump | 11.1 |
| P3 | Booster Pump | 20.1 |
| P4 | Booster Pump | 28.3 |
| P5 | Booster Pump | 35.9 |
| P6 | Booster Pump | 42.9 |
| P7 | Booster Pump | 49.3 |
| P8 | Booster Pump | 54.7 |
| R1 | Pressure Regulation | 64 |
| R2 | Pressure Regulation | 68 |
| R3 | Pressure Regulation | 74 |
| P9 | Booster Pump | 86.4 |

* Location shown as miles from marine terminal.

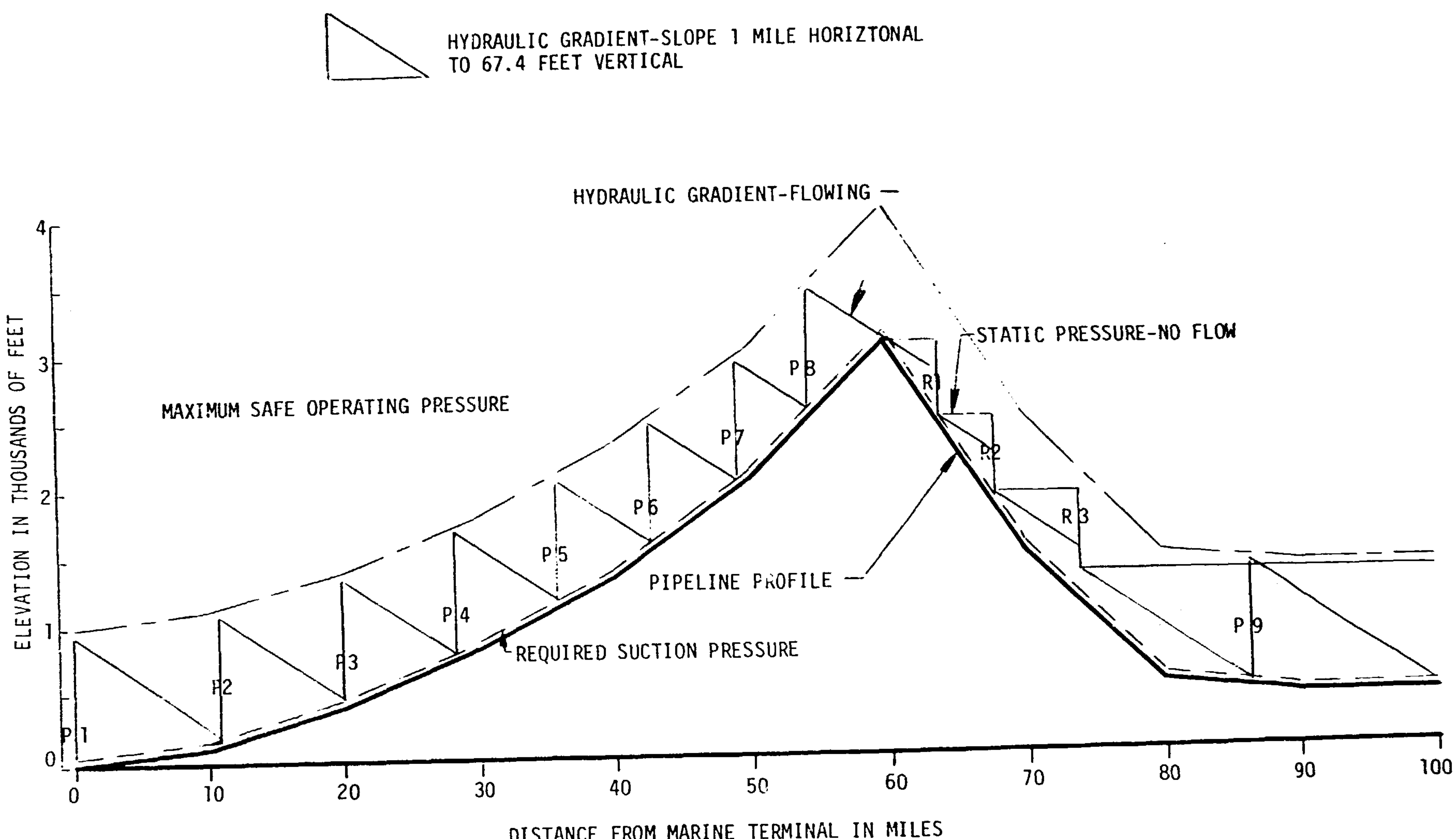



**Figure E-6. Hydraulic gradient for Alternative III – Scenario I.**

A pump station operating at 950 gal/min and 880.5 feet total dynamic head develops 211 water horsepower.

From Figure E-2, a booster pump of this size will have an efficiency of 0.800. The power required to drive a pump of this size is 264 brake horsepower.

(2) **Scenario II.** Under the specified design conditions, the fluid friction losses when flowing at 1,065 gal/min are 82.9 feet per mile. Adding the 5 feet per mile static gradient yields a total head requirement of 87.9 feet per mile or 8790 feet for the entire 100-mile pipeline. Ten booster pump stations, each developing 879 feet total dynamic head, are required to maintain the pump station discharge pressure below the maximum safe working pressure of 957 feet. Operating at 55 feet (20 lb/in$^2$) suction pressure and 897 feet total dynamic head produces a working pressure of 934 feet. The corresponding hydraulic gradient is shown in Figure E-7 with 10 miles between pump stations.

The hydraulic horsepower equivalent to 1,065 gal/min and 879 feet total dynamic head is 236 water horsepower. From Figure E-2, the pump efficiency is 0.801 with the pump power requirements being equal to 294 brake horsepower.

For both diesel-engine-driven pumps and turbine-engine-driven pumps, the total dynamic head can be developed by a single pump at each booster station.

d. **Alternative IV.** This pipeline concept joins 6061-T6 aluminum pipe by the ZAP-LOK mechanical swaging process. Using 8-inch schedule 40 pipe, having an inside diameter of 7.981 inches, the pipeline has a maximum safe operating pressure of 1,000 lb/in$^2$. With 20 lb/in$^2$ suction pressure, the maximum effective pressure loss between pump stations cannot exceed (1,000 - 20) = 980 lb/in$^2$. When pumping diesel fuel, equivalent heads are: 1000 lb/in$^2$ = 2735 feet of fuel, 980 lb/in$^2$ = 2,680 feet of fuel, and 20 lb/in$^2$ = 55 feet of fuel.

(1) **Scenario I.** Diesel fuel flowing at the design rate of flow of 950 gal/min incurs fluid friction losses of 82.1 feet per mile of pipeline. The total head requirements for the 100-mile pipeline, including the static head of 400 feet due to the net rise in elevation is (82.1) (100) + 400 = 8610 feet of fuel. The minimum number of pump stations required to develop the total head without exceeding the 2735 feet maximum safe working pressure is four. With each pump station developing 2,152.5 feet total dynamic head and having a suction pressure of 55 feet, the working pressure is 2207.5 feet of fuel. The hydraulic gradient for this pipeline system is shown in Figure E-8.

The dynamic hydraulic gradient is well below the maximum safe working pressure at all points along the pipeline. The pipeline profile falls 2600 feet from mile 60 to

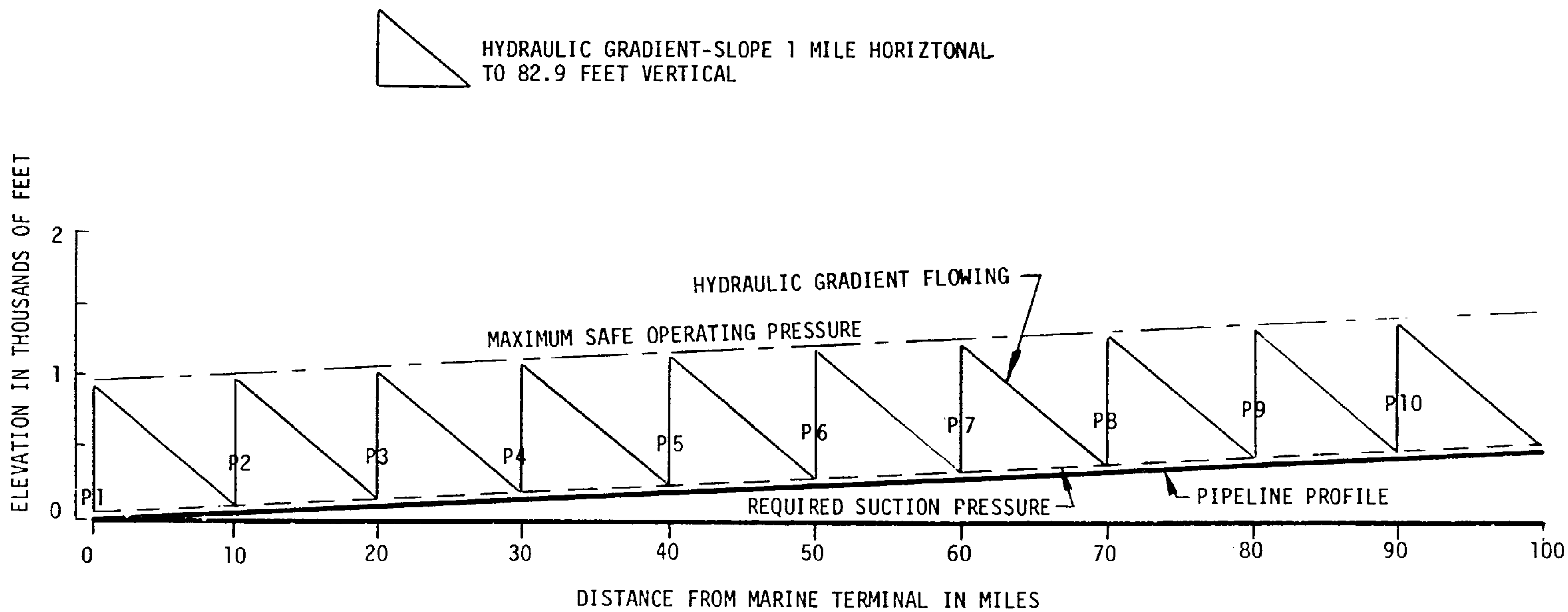



**Figure E-7. Hydraulic gradient for Alternative III – Scenario II.**

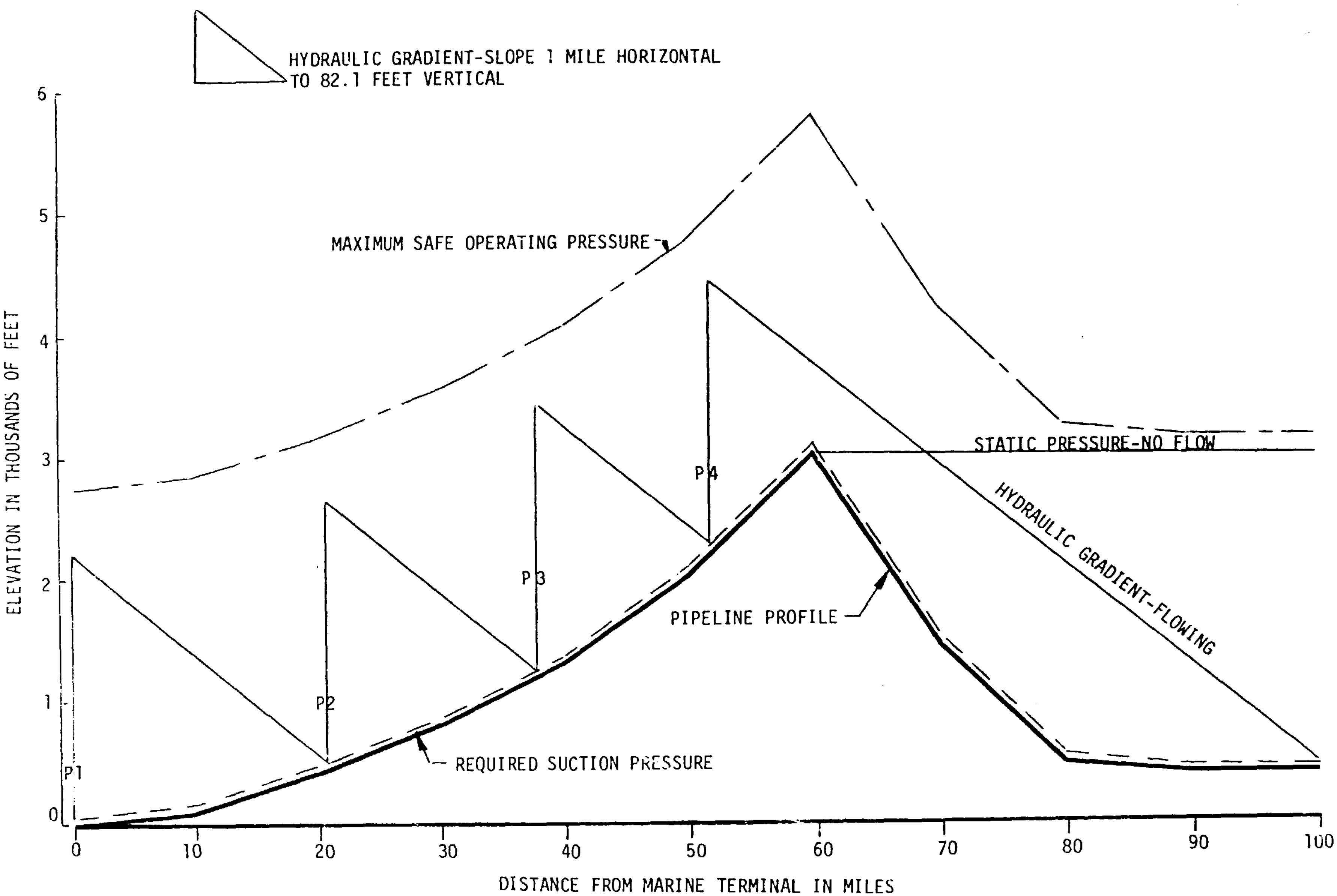



Figure E-8. Hydraulic gradient for Alternative IV – Scenario I.

mile 90. The resulting maximum static head of 2,600 feet is also less than the 2,735 feet maximum safe working pressure. Therefore, no pressure regulation stations are required for this pipeline design.

The booster pump station location are shown in Table E-4.

Table E-4. Location of Pipeline Booster Pump Stations for Alternative IV -- Scenario I.

| Station | Location* |
|---|---|
| P1 | 0 |
| P2 | 20.9 |
| P3 | 37.9 |
| P4 | 51.9 |

*Location shown as miles from the marine terminal.

The booster pump station performance requirements of 950 gal/min and 2,152.5 feet of head correspond to 516 water horsepower.

Based on pump efficiency of 0.807 from Figure E-2, the power required to drive the pump would be 611 brake horsepower.

A single turbine-engine-driven pump is capable of delivering the required horsepower. Two diesel-engine-driven pump units, each rated at approximately 305 brake horsepower are required. Otherwise the weight and size of the pump units would exceed Military transportability limits.

(2) **Scenario II.** As in Scenario I, the maximum safe working pressure, maximum loss between pump stations and pump station suction pressure are 2,734, 2,680, and 55 feet, respectively. At the specified flow rate of 1,065 gal/min the fluid friction losses are equal to 96.32 feet per mile of pipeline. The 500 feet rise in elevation along the length of the pipeline added to 9632 feet dynamic flow losses creates a total head requirement of 10,132 feet. When four pump stations are used, each pump station must develop (10,132/4) = 2,533 feet pump suction pressure. The hydraulic gradient is shown in Figure E-9 with the pump stations located 25 miles apart.

The hydraulic horsepower of a pump operating at 1,065 gal/min and 2,533 feet of head is 681 water horsepower. From Figure E-2, a pump of this size would have an efficiency of 0.811. The required brake horsepower is computed to be 840 brake horsepower.

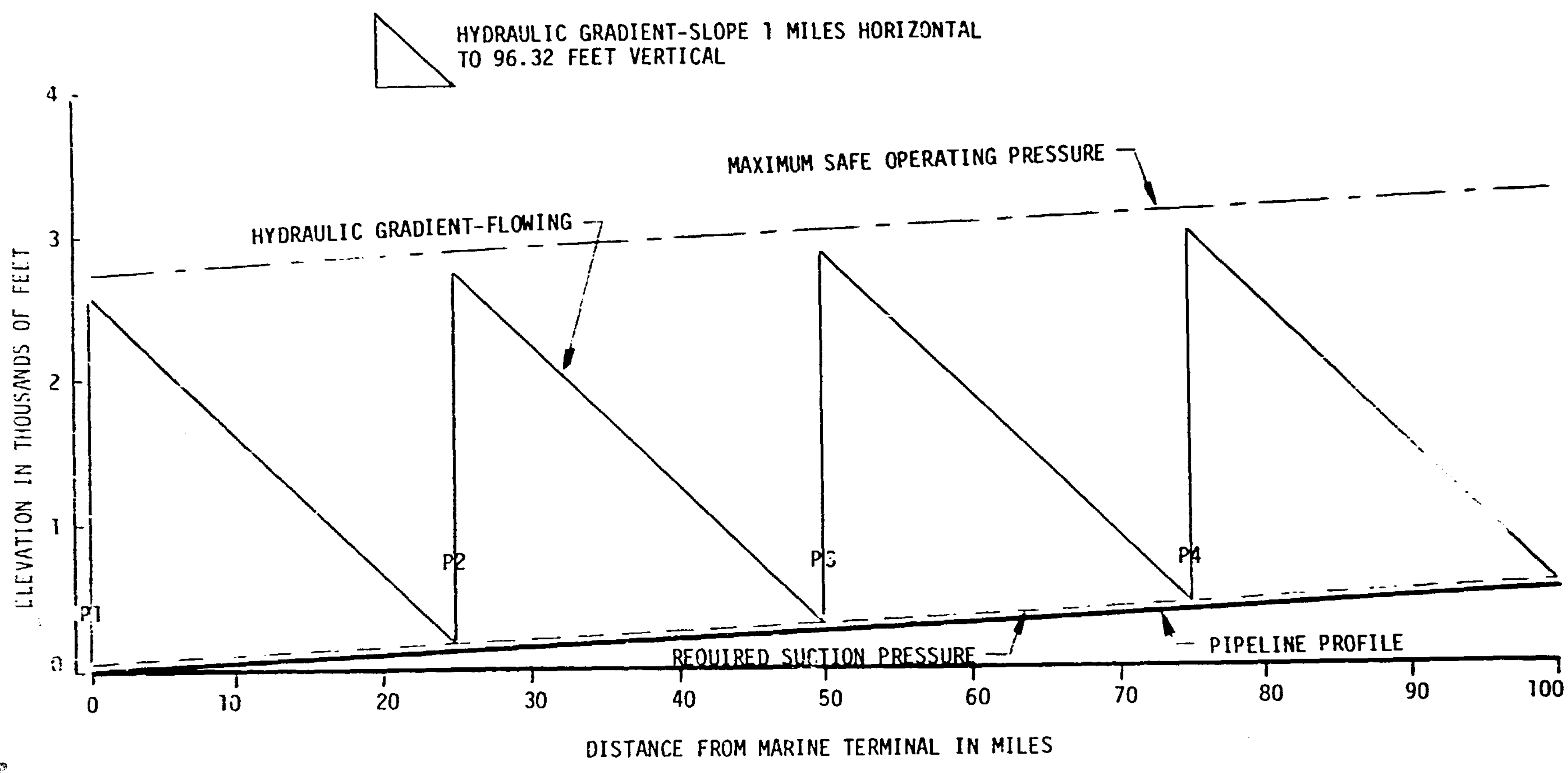



Figure E-9. Hydraulic gradient for Alternative IV – Scenario II.

One turbine-engine-driven pump can satisfy the total pump station performance requirements. Three diesel-engine-driven pumps, each rated at approximately 280 brake horsepower, are required at each pump station to prevent the weight and size of the pumps from exceeding Military transportability limitations.

e. **Alternative V.** This alternative employs 6063-T6, 8-inch, 0.200-inch-wall, aluminum pipe joined by mechanical couplings manufactured by Raco and Race, Inc. The maximum safe working pressure recommended by the manufacturer for this pipe is 482 lb/in$^2$. With 20 lb/in$^2$ suction pressure required the maximum pressure loss between pump stations is (482 - 20) = 462 lb/in$^2$. Expressed in feet of head using 0.8448 specific gravity diesel fuel, pressures of 482, 462, and 20 lb/in$^2$ correspond to static heads of 1,318, 1,263, and 55 feet of fuel, respectively.

(1) **Scenario I.** At the design rate of flow, the fluid friction loss through the 8.225-inch-inside-diameter pipe is computed to be 71.4 feet of fuel per mile. The total flow losses from the marine terminal to mile 60 are (60) (71.4) = 4,284 feet plus 3000 feet increase in elevation, or 7,284 feet. Dividing the total required head by the maximum allowable total dynamic head per pump station yields a value of (7,284/1,263) = 5.77. Therefore, six booster pump stations are required to develop 7,284 feet of head or (7,284/6) = 1,214 feet of head per station. Figure E-10 shows the hydraulic gradient for this pipeline design.

On the downhill run from mile 60 to mile 90 the hydraulic gradient at design flow conditions would not exceed the maximum safe working pressure for the pipe. However, under no-flow conditions the static head would be 1,318 feet at mile 68.79 resulting in overpressuring of the line from that point to mile 100. Locating a pressure regulation station at mile 68.67 adjusted to maintain the discharge head at atmospheric pressure will limit the maximum static head to 1300 feet of fuel at the inlet to the pressure regulation station and between mile 90 and mile 100. Below the pressure regulation station, the static head will push the fuel to mile 84.5 at the design rate of flow. A pump station developing 1062 feet of head is required at this point to push the fuel on to the end of the pipeline.

A pumping station delivering 950 gal/min at 1,214 feet total dynamic develops 291 water horsepower.

Applying a pump efficiency of 0.802 from Figure E-2, the required engine power rating is 362 horsepower. Using turbine-engine-driven pumps, one pump unit can handle the entire pumping operation at each booster pump station. The size and weight of a diesel-engine-driven pump of this capacity would exceed the transportability limits established herein. Therefore, at least two diesel-engine-driven pumps would be required at each pump station.

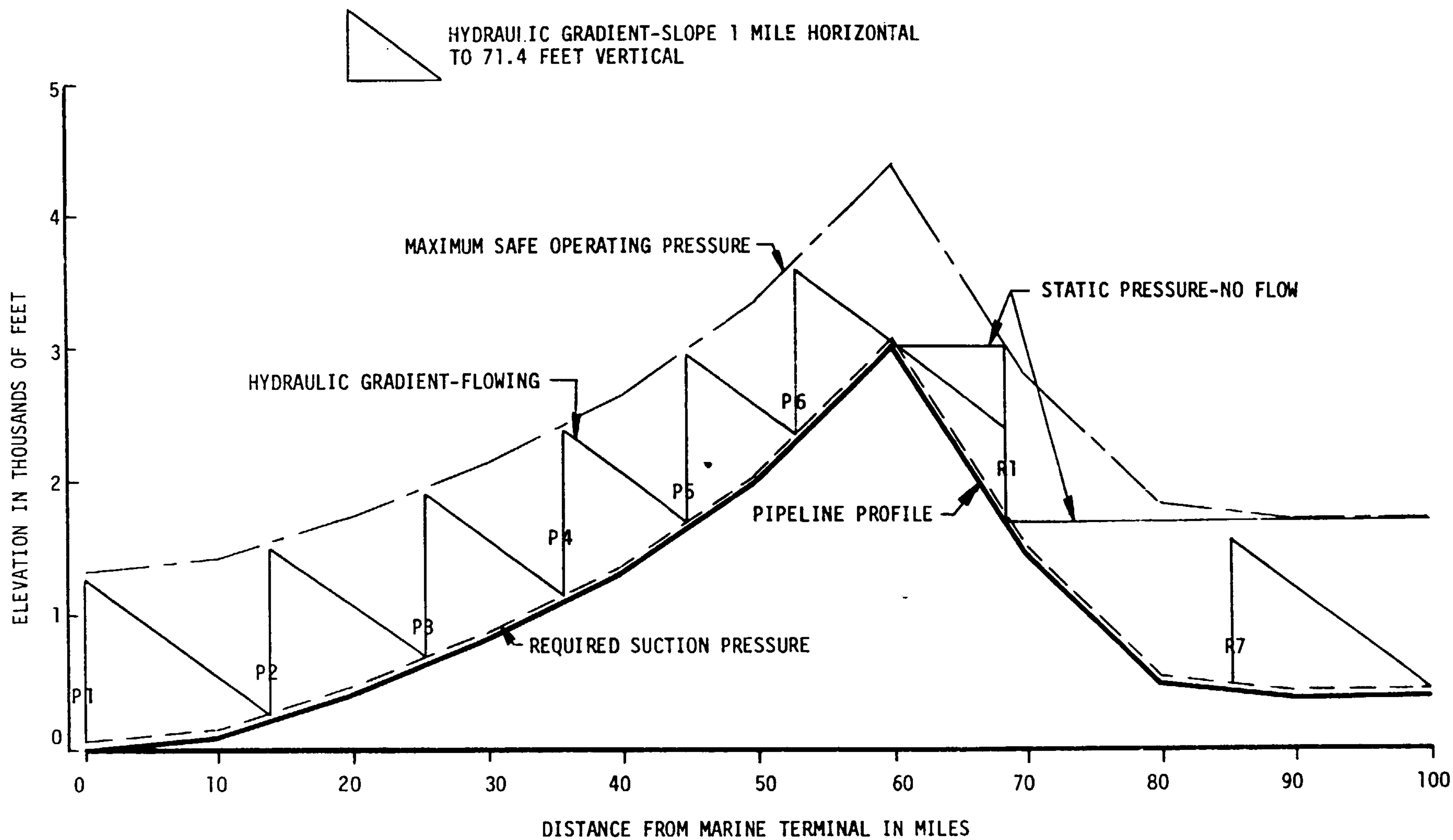



Figure E-10. Hydraulic gradient for Alternative V – Scenario I.

The locations of the pump stations and pressure regulation stations are shown in Table E-5.

Table E-5. Location of Pipeline Booster Pump Stations and Pressure Regulation Stations for Alternative V – Scenario I.

| Station | Type | Location* |
|---|---|---|
| P1 | Booster Pump | 0 |
| P2 | Booster Pump | 13.94 |
| P3 | Booster Pump | 25.38 |
| P4 | Booster Pump | 35.76 |
| P5 | Booster Pump | 44.95 |
| P6 | Booster Pump | 52.91 |
| R1 | Pressure Regulation | 68.67 |
| P7 | Booster Pump | 84.5 |

* Location shown as miles from marine terminal.

(2) **Scenario II.** The specified design conditions result in fluid friction losses of 88.8 feet per mile. Adding the 5-feet-per-mile static gradient of the pipeline profile yields a total head requirement of (88.8 + 5) = 93.8 feet per mile or 9,380 feet for the 100-mile pipeline. Eight booster pump stations, each developing 1172 feet total dynamic head, will develop the required head within the limits of the 1,318 feet maximum safe working pressure. Operating with 55 feet (20 $lb/in^2$) suction pressure, the pump station discharge pressure, or working pressure, at design conditions is (55 + 1,172) = 1,227 feet of fuel. The resultant hydraulic gradient is shown in Figure E-11.

The hydraulic horsepower equivalent to 1,065 gal/min and 1,172 feet total dynamic head is 315 water horsepower. From Figure E-2, the corresponding pump efficiency is 0.801 used to compute the required engine rating of 393 brake horsepower. As in Scenario I, a single turbine-engine-driven pump unit can be used at each pump station. Two diesel-engine-driven pumps will be required at each station because of transportability limits on size and weight.

f. **Alternative VI.** In this pipeline design, 6061-T6 alloy schedule 10 aluminum pipe is joined by the ZAP-LOK mechanical swaging process. The 8.329-inch-inside-diameter pipe has a maximum safe operating pressure of 661 $lb/in^2$, equivalent to 1,807 feet of diesel fuel. Operating with 20 $lb/in^2$ (55 feet) pump suction pressure, the maximum pressure loss between pump stations is limited to (1,807 - 55) = 1,752 feet of fuel.

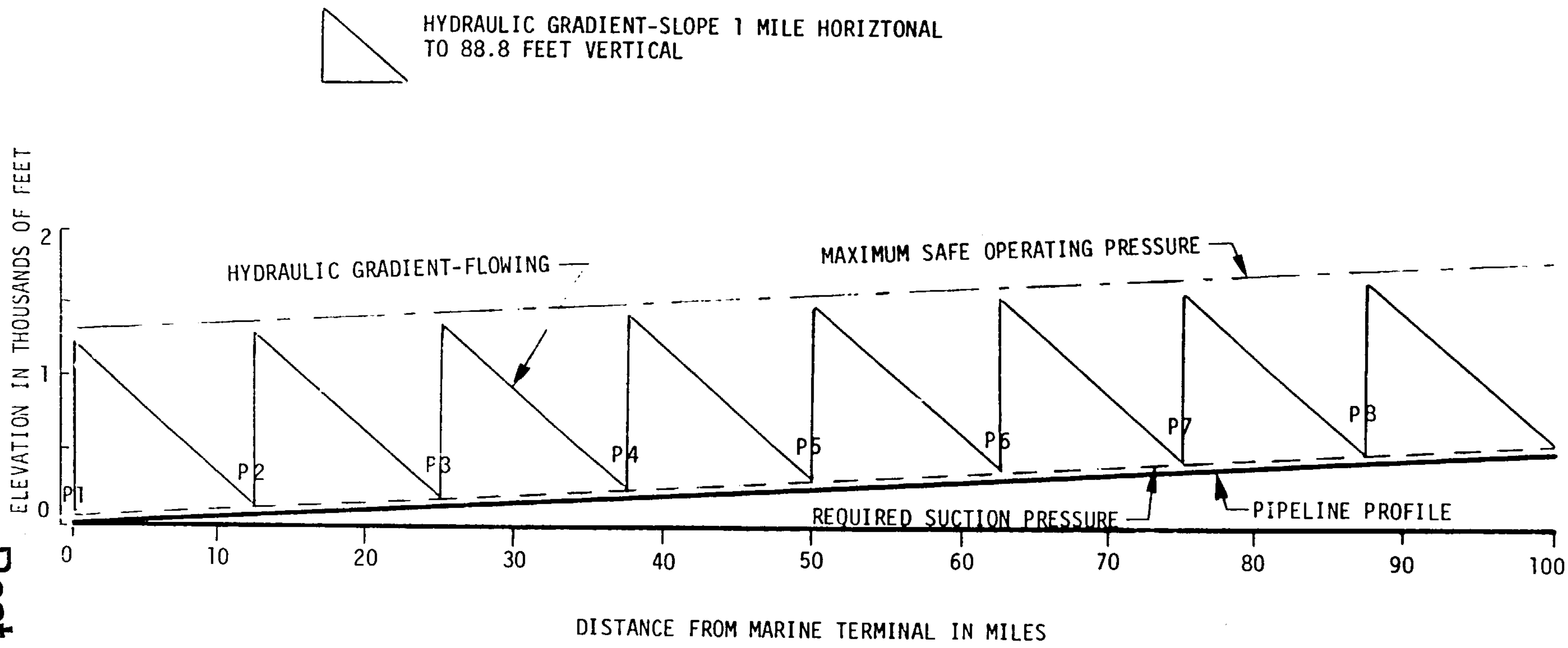



**Figure E-11. Hydraulic gradient for Alternative V – Scenario II.**

(1) **Scenario I.** Diesel fuel flowing at the design rate of flow of 950 gal/min will lose 67.1 feet of head per mile due to fluid friction. The total head requirement for the first 60 miles of pipeline is (67.1) (60) + 3,000 = 7,026 feet including the 3,000 feet rise in elevation. Four pump stations discharging at the maximum safe operating pressure will develop (4) (1752) = 7,008 feet of head. Adding the 55 feet suction head available at station 1, the total dynamic head is (7,008 + 55 - 7,026) = 36 feet at mile 60.

The hydraulic gradient when flowing at design conditions, shown in Figure E-12, would be below the maximum safe working pressure at all points from pump station P4 to the end of the pipeline without a pressure regulation station. However, a pressure regulation station must be used on the downhill slope from mile 60 to mile 80 to prevent overpressuring the pipeline under static conditions.

By locating a pressure regulation station at mile 67, adjusted to maintain the discharge pressure at 55 feet, the fuel will flow to mile 90 by gravity due to the drop in elevation. A pump station at mile 90 developing 614 feet total dynamic head will provide the pressure necessary to maintain the design rate of flow to the end of the pipeline. The resulting hydraulic gradient is shown in Figure E-12. Locations of the booster pump stations and pressure regulation stations are shown in Table E-6.

Table E-6. Location of Pipeline Booster Pump Stations and Pressure Regulation Stations for Alternative VI – Scenario I.

| Station | Type | Location* |
|---|---|---|
| P1 | Booster Pump | 0 |
| P2 | Booster Pump | 20.09 |
| P3 | Booster Pump | 35.90 |
| P4 | Booster Pump | 49.28 |
| R1 | Pressure Regulation | 67 |
| P5 | Booster Pump | 90 |

* Location shown as miles from marine terminal.

The booster pump performance requirements for station P1 through P4 correspond to 420 water horsepower. Based on a pump efficiency of 0.805 from Figure E-2, the power required to drive the pump is 522 brake horsepower. A single turbine-engine-driven pump is capable of delivering the required pump performance. In order to maintain the pump unit weight and size within the transportability limits, two diesel-engine-driven pumps will be required at pump station P1, P2, P3, and P4. A single pump of the same capacity would be adequate at pump station P5.

(2) **Scenario II.** As in Scenario I, the suction pressure, maximum total dynamic head at each pump station, and maximum safe operating pressure are 55, 1,752, and 1,807 feet of fuel, respectively. At the specified rate of flow of 1,065

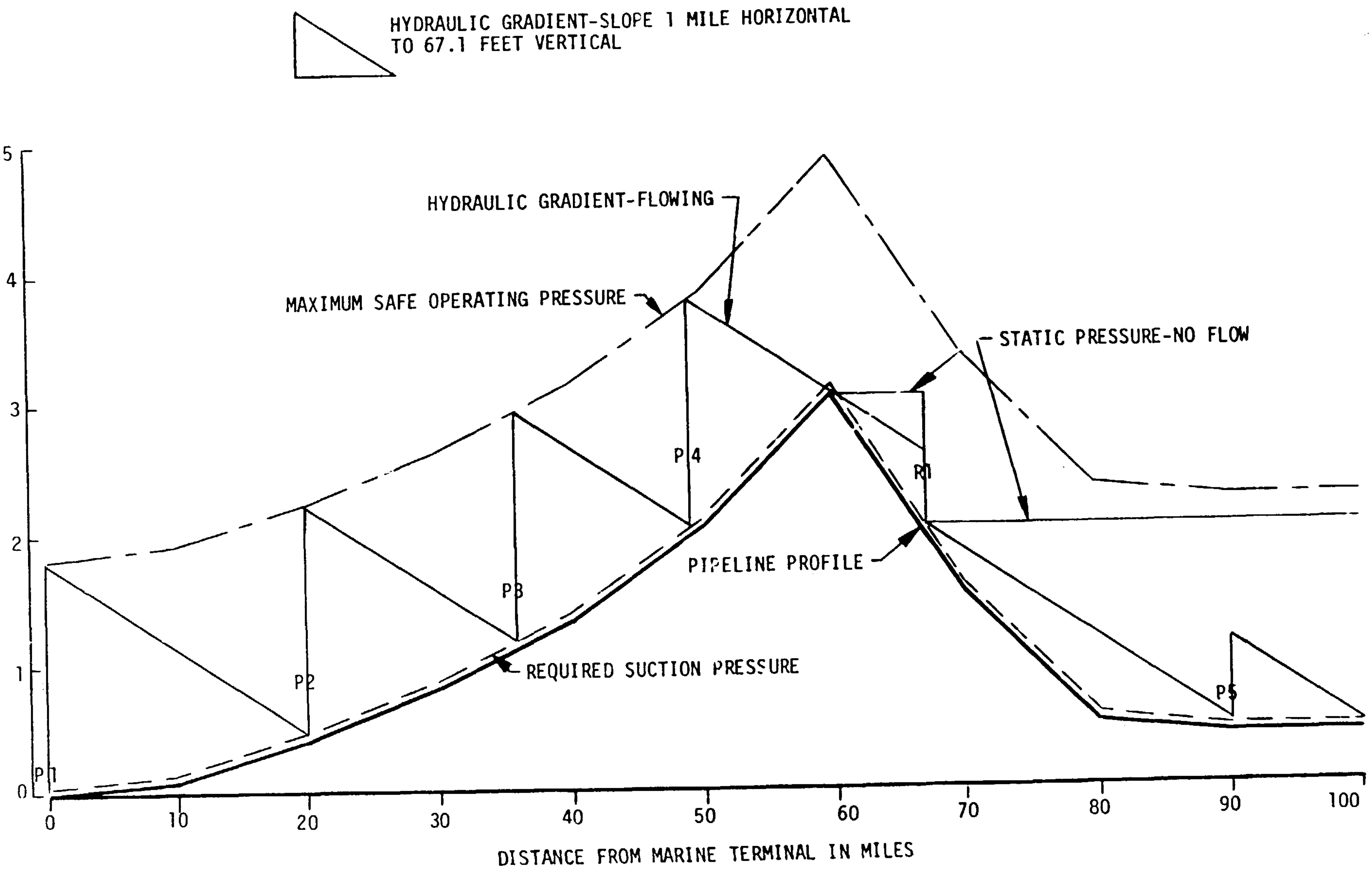



Figure E-12. Hydraulic gradient for Alternative VI – Scenario I.

gal/min, the fluid friction losses are equal to 82.4 feet per mile. Adding the 5-feet-per-mile rise in elevation creates a total head requirement for the 100 miles of pipeline or (82.4 + 5) (100) = 8,740 feet. Five booster pump stations, each developing 1,748 feet total dynamic head, will provide the required hydraulic horsepower. The hydraulic gradient is shown in Figure E-13 with the pump stations located 20 miles apart.

The hydraulic horsepower produced by a pump operating at 1,065 gal/min and 1,748 feet total dynamic head is 470 water horsepower. Based on a pump efficiency of 0.806 from Figure E-2, the required engine power rating is 583 brake horsepower. One turbine-engine-driven pump can satisfy the total pump station power requirements. Transportability limits on pump size and weight will require two diesel-engine-driven pumps at each booster station.

g. **Alternative VII.** Selected to evaluate the possibility of using two 6-inch-diameter pipelines, this alternative uses 6063-T6 aluminum pipe joined by Race and Race, Inc., mechanical coupling. The maximum safe working pressure recommended by the manufacturer for the 6.625-inch-outside-diameter, 0.134-inch wall pipe is 410 lb/in$^2$. As with the 8-inch-diameter pipelines, the minimum acceptable pump station suction pressure is assumed to be 20 lb/in$^2$. The maximum pressure loss between pump stations is 410 - 20 = 390 lb/in$^2$ which is equivalent to 1,066 feet of fuel.

(1) **Scenario I.** The design flow for each 6-inch pipeline is assumed to be one half the 950 gal/min flow rate used for 8-inch pipelines, or (950/2) = 475 gal/min. At this rate of flow, the fluid friction losses will be 69.9 feet of fuel per mile of pipeline length. Adding the 3,000 feet static head due to the rise in elevation, the pump stations in the first 60 miles of the pipeline must develop a total head of (4,194 + 3,000) = 7,194 feet of fuel. Dividing the total required head by the maximum allowable pressure rise at each station yields a value of (7,194/1,006) = 6.74. Therefore, seven pump stations are required, developing (7,194/7) = 1,028 feet of dynamic head. Figure E-14 shows the hydraulic gradient for this pipeline design.

On the downhill run from mile 60 to 90, both the static and dynamic gradients would exceed the maximum safe working pressure for the pipe without the use of pressure regulation stations. The optimum system design would use two pressure regulation stations located at mile 67 and mile 76 as shown in Figure E-14. By proper adjustment of the discharge pressure at the mile 76 pressure regulation station, the available static head will maintain the desired rate of flow to mile 89.2. Another pump station is requested at that point to push the fuel to the end of the pipeline at the required flow rate.

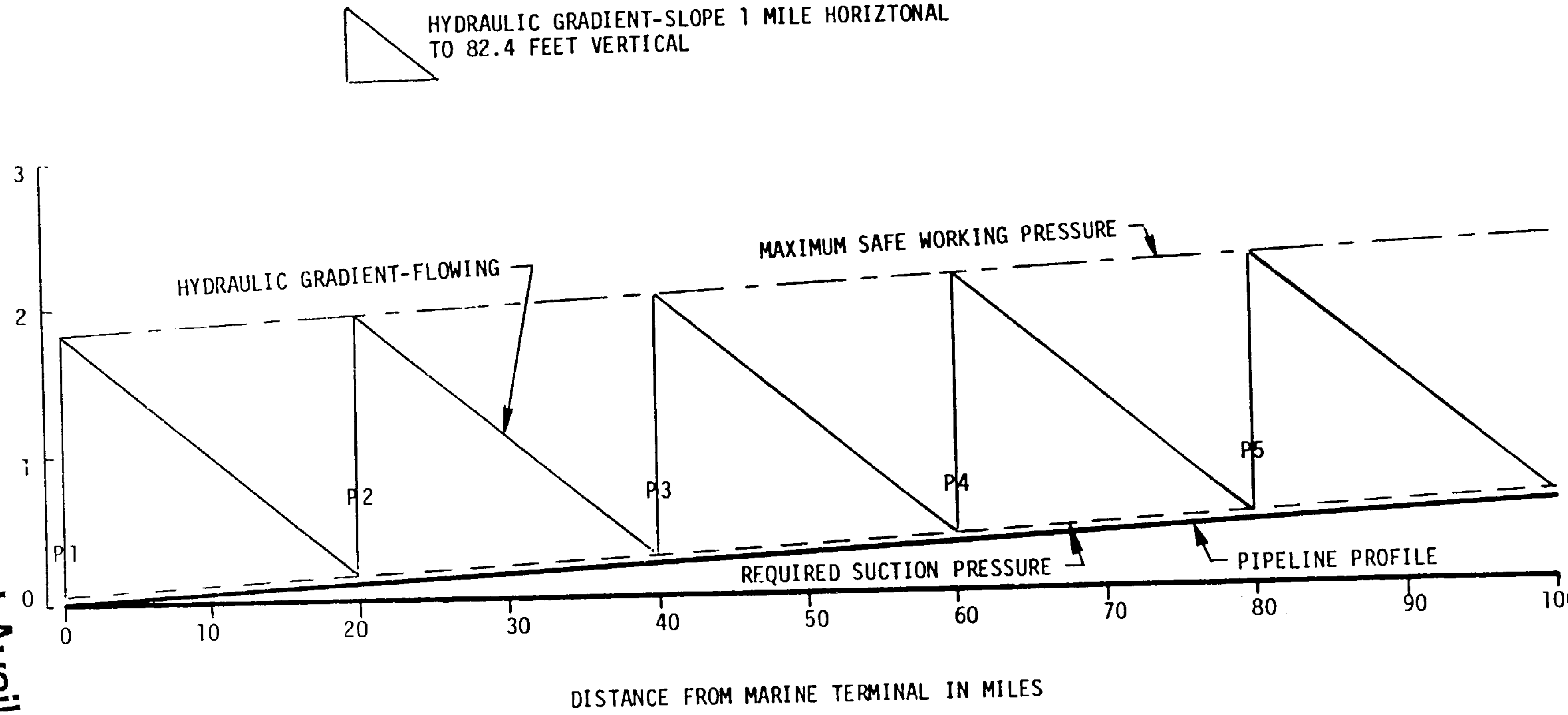



**Figure E-13. Hydraulic gradient for Alternative VI – Scenario II.**

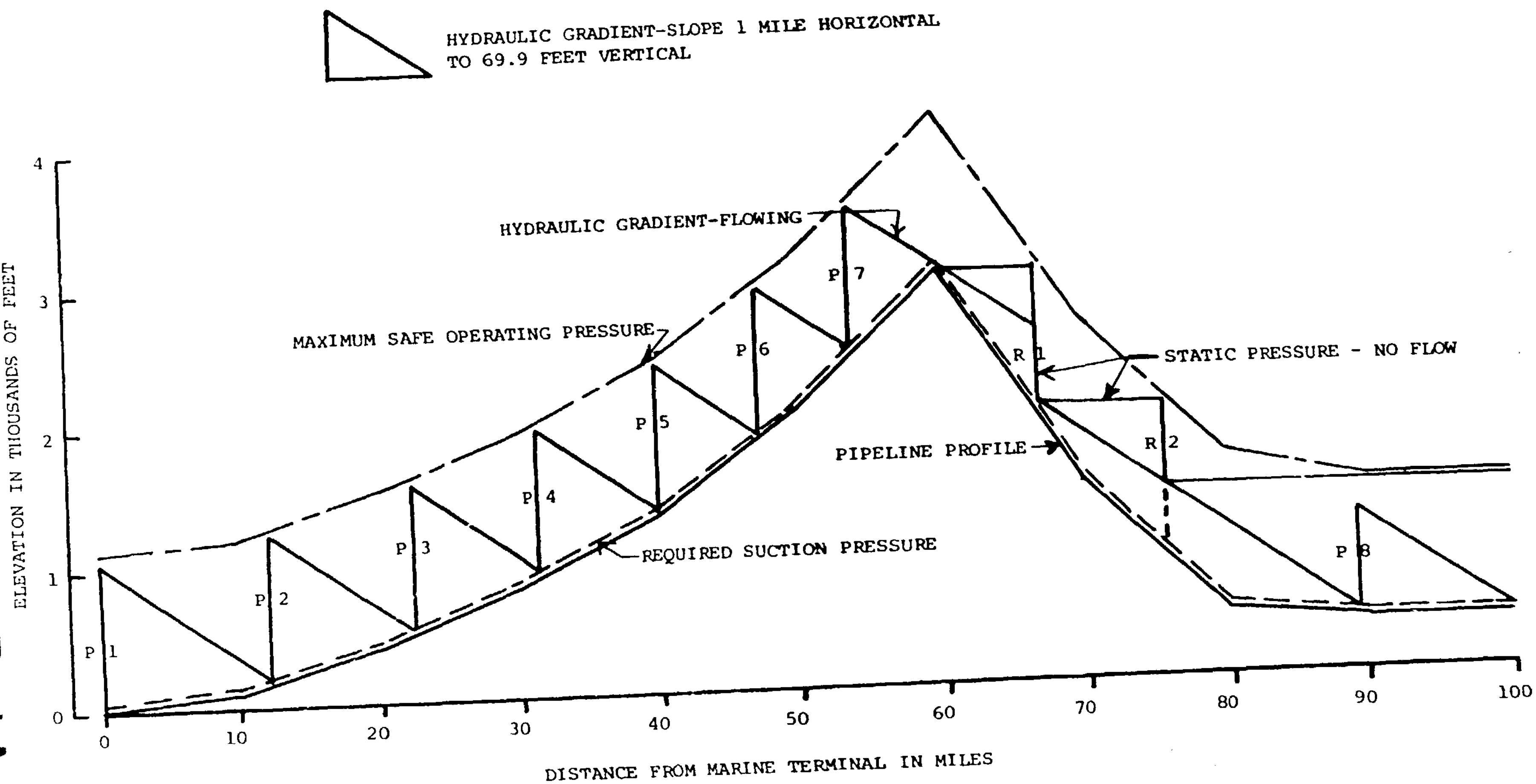



Figure E-14. Hydraulic gradient for Alternative VII – Scenario I.

The locations for all booster pump stations and pressure regulation stations are listed in Table E-7.

Table E-7. Location of Pipeline Booster Pump Stations and Pressure Regulation Stations for Alternative VII – Scenario I.

| Station | Type | Location* |
|---|---|---|
| P1 | Booster Pump | 0 |
| P2 | Booster Pump | 12.3 |
| P3 | Booster Pump | 22.4 |
| P4 | Booster Pump | 31.6 |
| P5 | Booster Pump | 40.1 |
| P6 | Booster Pump | 47.5 |
| P7 | Booster Pump | 53.9 |
| R1 | Pressure Regulation | 67.0 |
| R2 | Pressure Regulation | 76.0 |
| P8 | Booster Pump | 89.2 |

* Location shown as miles from marine terminal.

A pump station operating at 475 gal/min and 1,028 feet total dynamic head develops 123 water horsepower. Using a pump efficiency of 0.798 from Figure E-2, the power required to drive the pump will be 155 brake horsepower.

(2) **Scenario II.** Under the specified design conditions with each 6-inch pipeline carrying one-half the required 1,065 gal/min rate of flow, or 532.5 gal/min, the fluid friction losses are computed to be 92.3 feet of fuel per mile. Adding the 5-feet-per-mile static gradient yields a total head of 97.3 feet per mile or 9,730 feet of fuel for the entire 100-mile pipeline. Ten booster stations, each developing 973 feet total dynamic head, are required to maintain the pump station discharge pressure below the maximum safe working pressure of 1,121 feet of diesel fuel. The resulting hydraulic gradient is shown in Figure E-15.

The hydraulic horsepower equivalent to 532.5 gal/min and 973 feet of fuel is 135 water horsepower. From Figure E-2, the pump efficiency will be 0.798. The pump power requirement is 135/0.78 = 169 brake horsepower.

h. **Alternative VIII.** This pipeline design is based on using two parallel 6-inch-diameter, 6061-T6 aluminum alloy, schedule 10 pipe joined by the ZAP-LOK mechanical swaging process. The 6.625-inch-outside-diameter pipe has a maximum safe working pressure of 780 lb/in$^2$, equivalent to 2,123 feet of diesel fuel. Operating with 20 lb/in$^2$ (55 feet of fuel) pump suction pressure, the maximum pressure loss between pump stations is (2,123 - 55) = 2068 feet of fuel.

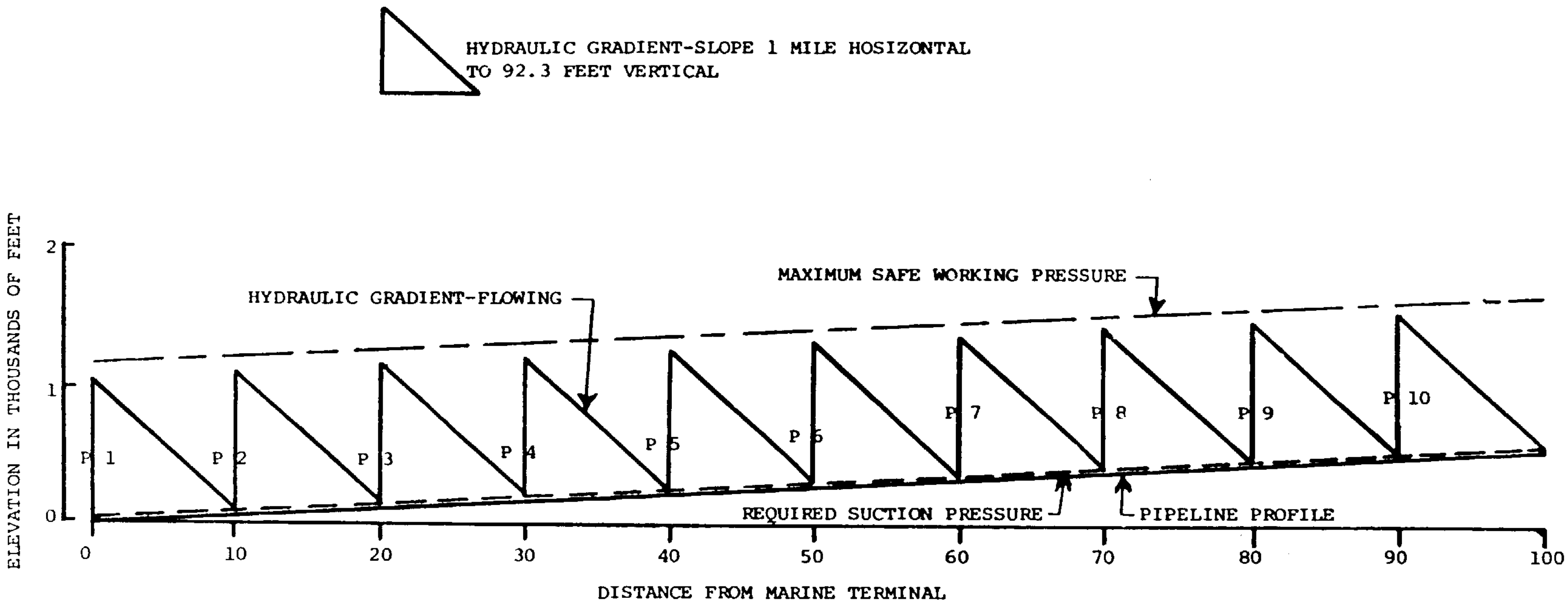



Figure E-15. Hydraulic gradient for Alternative VII – Scenario II.

(1) **Scenario I.** Diesel fuel flowing at the design rate of flow of (950/2) = 475 gal/min will lose 69.9 feet of head per mile due to fluid friction. The total head requirement for the first 60 miles of pipeline is (69.9) (60) + 3,000 = 7,194 feet of fuel. Using four booster pump stations, the total dynamic head developed at each station is (7,194/4) = 1,799 feet of fuel.

The hydraulic gradient, shown in Figure E-16, when flowing at design conditions is below the maximum safe working pressure at all points along the pipeline without the use of a pressure regulation station. However, a pressure regulation station must be used on the downhill run from mile 60 to mile 80 to prevent overpressuring the pipeline under static conditions. The resulting static gradient is as shown in Figure E-16 with the pipeline booster pump station's and pressure regulation station's locations as listed in Table E-8.

Table E-8. Location of Pipeline Booster Pump Stations and Pressure Regulation Stations for Alternative VIII – Scenario I.

| Station | Type | Location* |
|---|---|---|
| P1 | Booster Pump | 0 |
| P2 | Booster Pump | 20.0 |
| P3 | Booster Pump | 35.9 |
| P4 | Booster Pump | 49.3 |
| R1 | Pressure Regulation | 73.0 |

* Location shown as miles from marine terminal.

The power requirement for a pump station operating at 475 gal/min and 1799 feet total dynamic head is 222 water horsepower. Based on a pump efficiency of 0.800 from Figure E-2, the pump engine must have a continuous power rating of 278 brake horsepower

(2) **Scenario II.** As in Scenario I, the suction pressure, maximum total dynamic head at each booster pump station and maximum safe operating pressure are 55, 2,068, and 2,123 feet of diesel fuel, respectively. At one-half the required throughput rate of (1,065/2) = 532.5 gal/min, the fluid friction losses are equal to 92.3 feet per mile. Adding the 5-feet-per-mile rise in elevation gives a total dynamic head of (92.3 + 5) (100) = 9,730 feet of fuel for the 100 miles of pipeline. Five booster pump stations, each developing 1,946 feet total dynamic head, will provide the required hydraulic horsepower. The hydraulic gradient is shown in Figure E-17.

A flow rate of 932.5 gal/min and 1,946 feet total dynamic head is equal to 270 water horsepower. Based on a pump efficiency of 0.801 from Figure E-2, the required pump engine power rating is 337 brake horsepower. In order to maintain

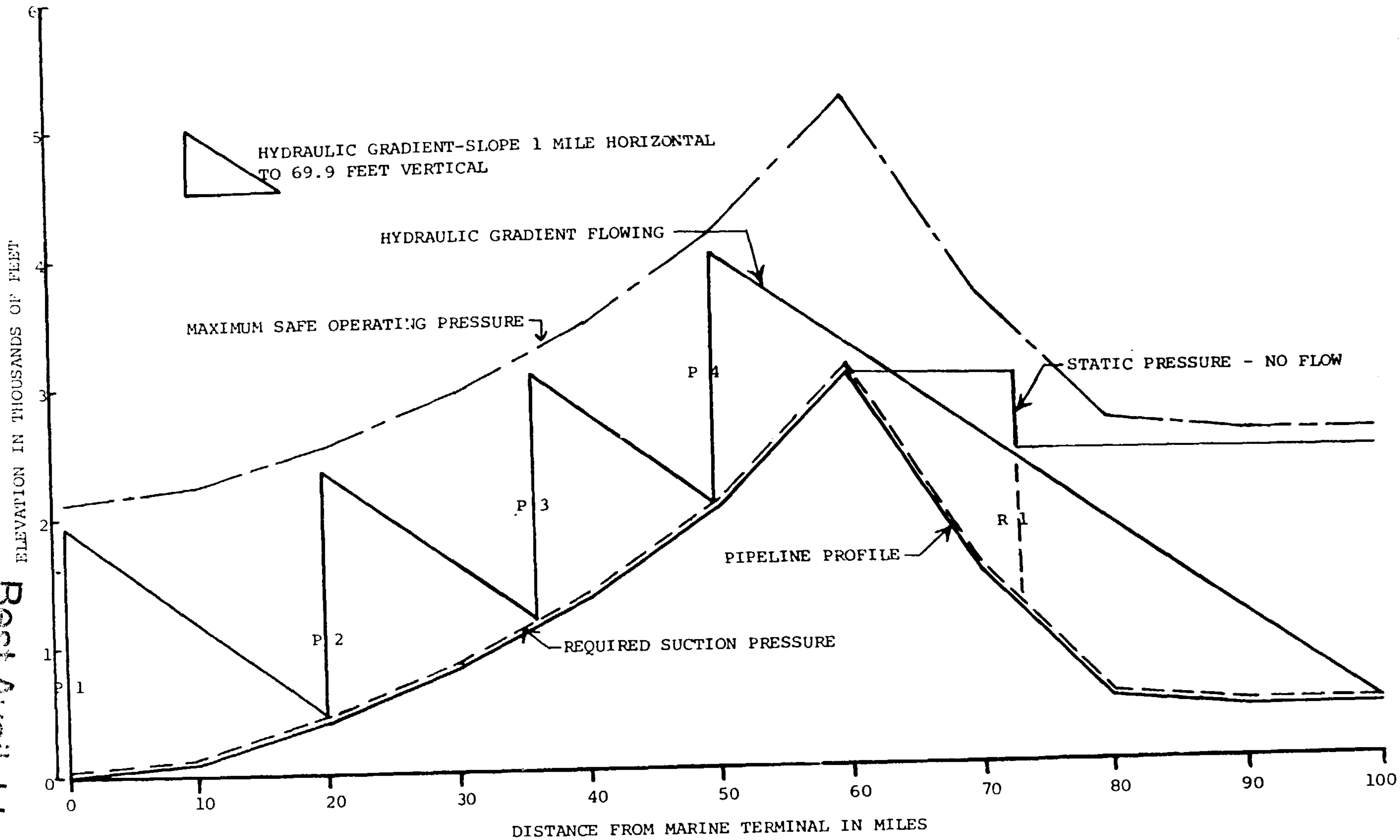



Figure E-16. Hydraulic gradient for Alternative VIII – Scenario I.

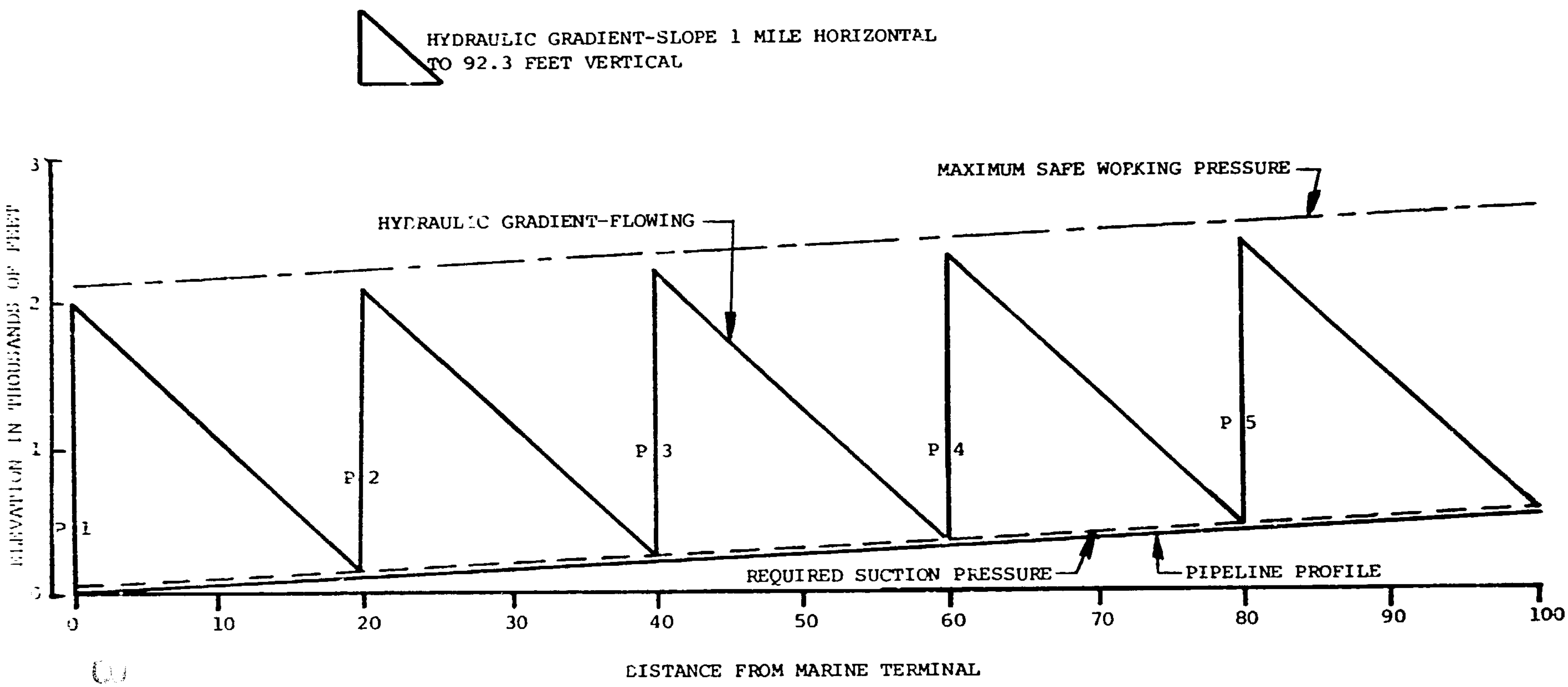



Figure E-17. Hydraulic gradient for Alternative VIII – Scenario II.

the pump unit weight and size within the transportability limits, two diesel-engine-driven pumps will be required at each booster pump station.

i. **Military Standard System.** To satisfy the scenario requirements with Military standard equipment, 8-inch, lightweight, steel tubing joined by grooved-end mechanical couplings would be used. The maximum safe working pressure of 500 lb/in$^2$ is equal to a head of 1,367 feet of diesel fuel. The Military standard 6-inch, 4-stage, diesel-engine-driven pump, conforming to MIL-P-53375A is designed to operate the 20 lb/in$^2$, or 55 feet, of pressure at the inlet. Thus, the maximum pressure rise at each pump station is limited to (1,367 - 55) = 1,312 feet total dynamic head. One pump unit is capable of developing this head at the design rates of flow for Scenarios I and II.

(1) **Scenario I.** At the design rate of flow of 950 gal/min, the fluid friction loss for diesel fuel is computed to be 63.7 feet per mile. The total head required in the pipeline segment from the marine terminal to the highest point in the pipeline at mile 60 is (63.7) (60) + 3,000 = 6,822 feet of fuel. Six pump stations each developing 1,137 feet of head will achieve the design rate of flow to mile 60. The actual working pressure will be 1,137 feet total dynamic head plus 55 feet suction pressure or 1,192 feet of fuel.

The drop in elevation of 2,600 feet between mile 60 and mile 90 exceeds the maximum safe working pressure of the lightweight steel tubing. As a result, a pressure regulation station must be used in this downhill run. When the pressure regulation station is located at mile 69 with the discharge pressure adjusted to 55 feet of fuel, the satic head at the inlet to the pressure regulation station under no-flow conditions will be 1350 feet. At the same time the maximum static head in the lowest section of the pipeline from mile 90 to mile 100 will be 1305 feet of fuel.

At design flow conditions, the static head will be adequate to move the fuel to mile 88. At that point a pump station adding 725 feet total dynamic head will be required to maintain flow. The resulting hydraulic gradient is shown in Figure E-18. The locations of the pump stations and pressure regulation stations are shown in Table E-9.

(2) **Scenario II.** At the specified design rate of flow of 1,065 gal/min, the fluid friction losses for diesel fuel will be 83.6 feet of fuel per mile. Adding 5 feet per mile rise in elevation gives a total head requirement of 88.6 feet per mile or 8,860 feet through 100 miles of pipeline. Seven booster pump stations, located 14.3 miles apart, will deliver the required flow when each booster station develops 1,266 feet total dynamic head. The hydraulic gradient for the pipeline is shown in Figure E-19.

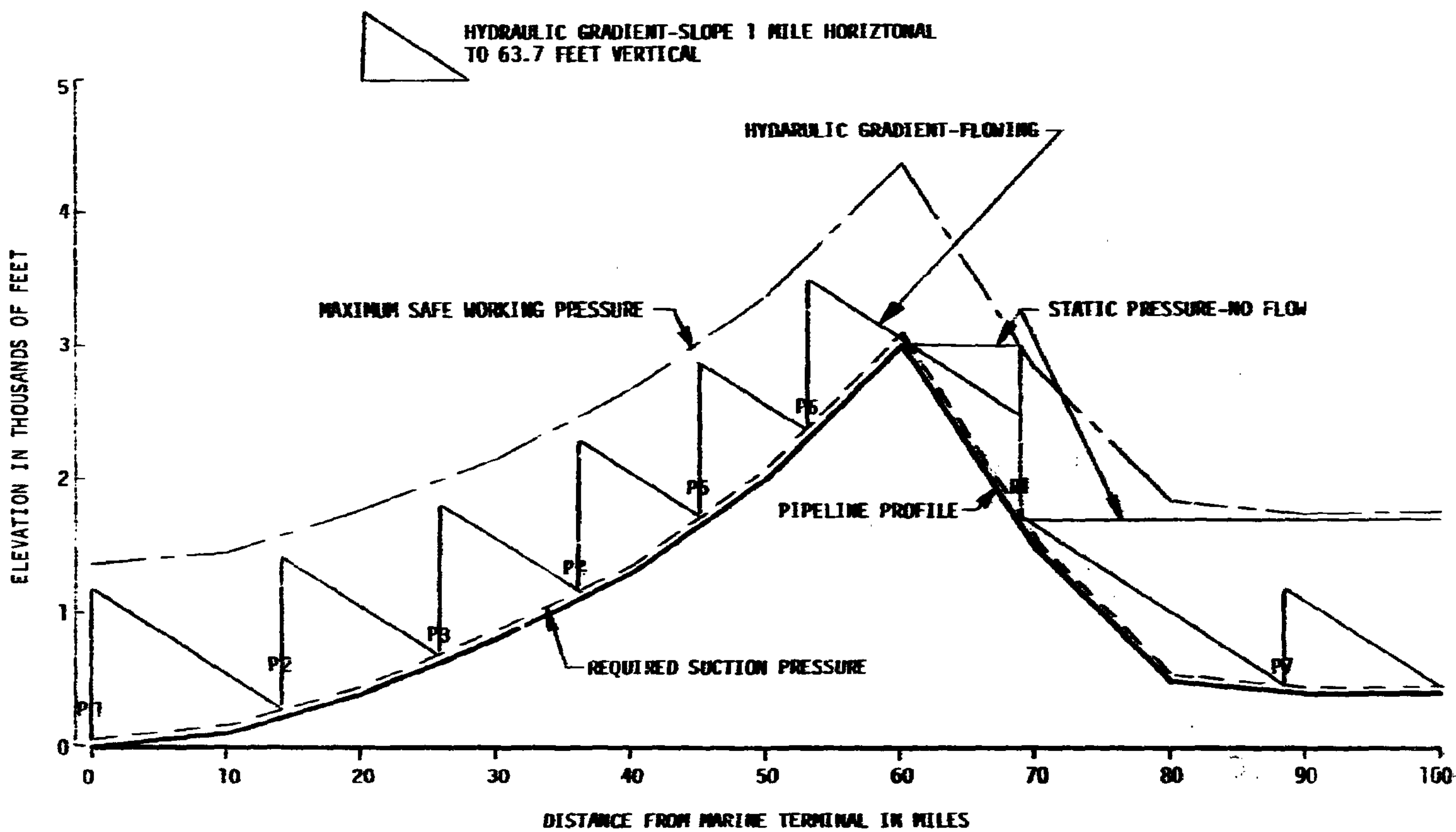



Figure E-18. Hydraulic gradient for Military standard pipeline, lightweight steel – Scenario I.

Table E-9. Location of Pipeline Booster Pump Stations and Pressure Regulation Stations for Military Standard Pipeline - Scenario I.

| Station | Type | Location* |
|---|---|---|
| P1 | Booster Pump | 0 |
| P2 | Booster Pump | 14.27 |
| P3 | Booster Pump | 25.79 |
| P4 | Booster Pump | 36.16 |
| P5 | Booster Pump | 45.24 |
| P6 | Booster Pump | 53.06 |
| R1 | Pressure Regulation | 69 |
| P7 | Booster Pump | 88 |

* Location shown in miles from the marine terminal.

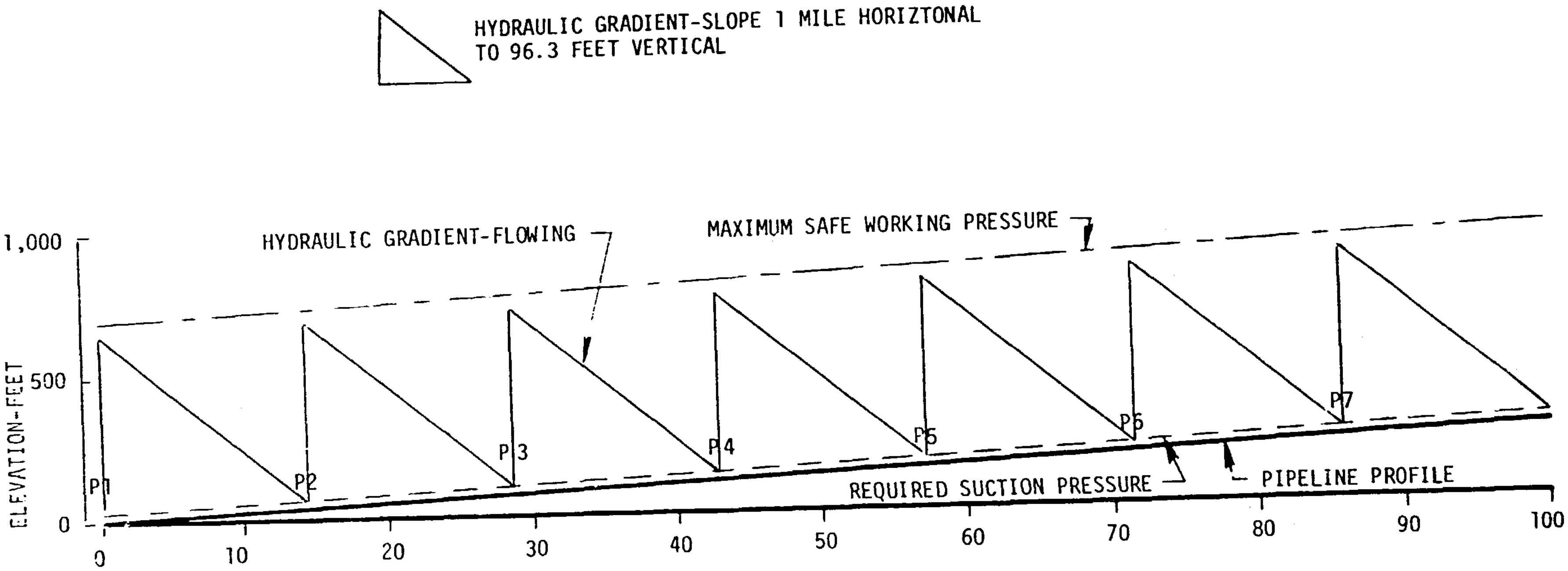



**Figure E-19. Hydraulic gradient for Military standard pipeline, lightweight steel — Scenario II.**

## DISTRIBUTION FOR MERADCOM REPORT 2249

| No. Copies | Addressee |
|---|---|
| | **Department of Defense** |
| 1 | Director, Technical Information<br>Defense Advanced Research<br>Projects Agency<br>1400 Wilson Blvd<br>Arlington, VA 22209 |
| 1 | Director<br>Defense Nuclear Agency<br>ATTN: STTL<br>Washington, DC 20305 |
| 12 | Defense Documentation Center<br>Cameron Station<br>Alexandria, VA 22314 |
| | **Department of the Army** |
| 1 | Commander<br>US Army Development and<br>Readiness Command<br>ATTN: DRCMM-SP<br>5001 Eisenhower Avenue<br>Alexandria, VA 22333 |
| 1 | Commander, HQ TRADOC<br>ATTN: ATEN-ME<br>Fort Monroe, VA 23651 |
| 1 | Commander, HQ TRADOC<br>ATTN: ATCD-SP-L<br>Fort Monroe, VA 23651 |
| 1 | HQDA (DAME-AOA-M)<br>Washington, DC 20310 |
| 1 | HQDA (DALO-TSE)<br>Washington, DC 20310 |
| 1 | HQDA (DAMA-CSS)<br>Washington, DC 20310 |
| 1 | HQDA (DAEN-RDL)<br>Washington, DC 20314 |
| 1 | HQDA (DAEN-MCE-D)<br>Washington, DC 20314 |
| 1 | Director<br>Army Materials and Mechanics<br>Research Center<br>ATTN: DRXMR-STL, Tech Library<br>Watertown, MA 02172 |
| 1 | Commander<br>US Army Aberdeen Proving Ground<br>ATTN: STEAP-MT-U (GE Branch)<br>Aberdeen Proving Ground, MD 21005 |
| 1 | Director<br>US Army Materiel Systems Analysis<br>Agency<br>ATTN: DRAMXSY-CM<br>Aberdeen Proving Ground, MD 21005 |
| 1 | Director<br>US Army Engineer Waterways<br>Experiment Station<br>ATTN: Chief, Library Branch<br>Technical Information Ctr<br>Vicksburg, MS 39180 |
| 1 | Commander<br>US Army Troop Support & Aviation<br>Materiel Readiness Command<br>ATTN: DRSTS-WJ<br>4300 Goodfellow Blvd<br>St. Louis, MO 63120 |
| 1 | Commander<br>US Army Troop Support & Aviation<br>Materiel Readiness Command<br>ATTN: DRSTS-KTE<br>4300 Goodfellow Blvd<br>St. Louis, MO 63120 |
| 2 | Director<br>Petrol & Fld Svc Dept<br>US Army Quartermaster School<br>Fort Lee, VA 23801 |
| 2 | Commandant<br>US Army Quartermaster School<br>ATTN: ATSM-CD-M<br>Fort Lee, VA 23801 |
| 1 | Commandant<br>US Army Engineer School<br>ATTN: ATSE-CD-CS<br>Fort Belvoir, VA 22060 |
| 1 | Commander<br>172nd Infantry Brigade<br>ATTN: DIO (AFZT-DI-L)<br>Fort Richardson, Alaska 99505 |

| No. Copies | Addressee | No. Copies | Addressee |
|---|---|---|---|
| 1 | HQ, USAREUR & Seventh Army<br>DCSENGR, ATTN: AEAEN-MO<br>ATTN: Mil Ops Div<br>APO New York 09403 | 1 | Commander<br>US Army Logistics Center<br>ATTN: ATCL-MS<br>Fort Lee, VA 23801 |
| 2 | Engineer Representative<br>US Army Standardization Group, UK<br>Box 65, FPO New York 09510 | | **MERADCOM** |
| 1 | HQ, DA, ODCSLOG<br>Directorate for Transportation and Services<br>Army Energy Office<br>ATTN: DALO-TSE<br>Washington, DC 20310 | 1 | Commander, DRDME-Z<br>Technical Director, DRDME-ZT<br>Assoc Tech Dir/R&D, DRDME-ZN<br>Assoc Tech Dir/Engrg & Acq, DRDME-ZE<br>Spec Asst/Matl Asmt, DRDME-ZG<br>Spec Asst/Tech Asmt, DRDME-ZK<br>CIRCULATE |
| 1 | Plastics Technical Evaluation Center<br>Picatinny Arsenal, Bldg 176<br>ATTN: A. M. Anzalone SARPA-FR-M-D<br>Dover, NJ 07801 | 1 | Chief, Ctrmine Lab, DRDME-N<br>Chief, Elec Pwr Lab, DRDME-E<br>Chief, Cam & Topo Lab, DRDME-R<br>Chief, Mar & Br Lab, DRDME-M<br>Chief, Mech & Constr Eqpt Lab, DRDME-H<br>Chief, Ctr Intrus Lab, DRDME-X<br>Chief, Matl Tech Lab, DRDME-V<br>Director, Product A&T Directorate, DRDME-T<br>CIRCULATE |
| 1 | Learning Resources Center<br>US Army Engineer School<br>Bldg 270<br>Fort Belvoir, VA 22060 | | |
| 1 | Commander<br>Headquarters, 39th Engineer Battalion (Cbt)<br>Fort Devens, MA 01433 | 1 | Engy & Water Res Lab, DRDME-G |
| | | 30 | Petrol & Envrion Tech Div, DRDME-GS |
| 1 | President<br>US Army Armor and Engineer Board<br>ATTN: ATZK-AE-TD-E<br>Fort Knox, KY 40121 | 1 | Engineering Division, DRDME-GE |
| | | 3 | Tech Report Ofc, DRDME-WP |
| | | 3 | Security Ofc (for liaison officers), DRDME-S |
| 1 | Commander<br>2nd Engineer Group<br>ATTN: S4<br>APO San Francisco 96301 | 2 | Tech Library, DRDME-WC |
| | | 1 | Plans, Programs & Ops Ofc, DRDME-U |
| 1 | Commander and Director<br>USAFESA<br>ATTN: FESA-RTD<br>Fort Belvoir, VA 22060 | 1 | Pub Affairs Ofc, DRDME-I |
| | | 1 | Ofc of Chief Counsel, DRDME-L |
| 1 | Director<br>US Army TRADOC Systems Analysis Activity<br>ATTN: ATAA-SL (Tech Lib)<br>White Sands Missile Range, NM 88002 | | **Department of the Navy** |
| | | 1 | Commander, Naval Facilities Engineering Command<br>Department of the Navy<br>ATTN: Code 032-A<br>200 Stovall St.<br>Alexandria, VA 22332 |
| 2 | Commander<br>US Army Logistics Management Ctr<br>ATTN: DRXMC-D<br>Fort Lee, VA 23801 | | |

| No. Copies | Addressee |
|---|---|
| 1 | Officer-in-Charge (Code L31)<br>Civil Engineering Laboratory<br>Naval Construction Battalion<br>Center<br>Port Hueneme, CA 93043 |
| 1 | Naval Training Equipment Center<br>ATTN: Technical Library<br>Orlando, FL 32813 |
| | **Department of the Air Force** |
| 1 | HQ USAF/RDPS (Mr. Allan Eaffy)<br>Washington, DC 20330 |
| 1 | Mr. William J. Engle<br>Chief, Utilities Branch<br>HQ USAF/PREEU<br>Washington, DC 20332 |
| 1 | HQ USAF/PREES<br>ATTN: Mr. Edwin B. Mixon<br>Bolling AFB-Bldg 626<br>Washington, DC 20332 |
| 1 | AFAPL/SFL<br>Wright-Patterson AFB, OH 45433 |
| 1 | Department of Transportation<br>Library, FOB 10A, TAD-494.6<br>800 Independence Avenue, SW<br>Washington, DC 20591 |
| 1 | Professor Raymond R. Fox<br>School of Engineering and<br>Applied Science<br>The George Washington University<br>Washington, DC 20052 |





1 2 3 82,384 — AG — Ft Belvoir