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SYSTEMS, SCIENCE AND SOFTWARE

SSS-R-77-3025
Revision 0

EFFICIENCY IMPROVEMENTS
IN
PIPELINE TRANSPORTATION SYSTEMS

by

William F. Banks

James F. Horton

Technical Report - Task 3

Contract EY-76-C-03-1171

ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION

SAN FRANCISCO OPERATIONS OFFICE

1333 BROADWAY

OAKLAND, CA 94612

September 9, 1977

P. O. BOX 1620, LA JOLLA, CALIFORNIA 92038, TELEPHONE (714) 453-0060

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REB

Abstract

The primary purposes of this report are the identification of those potential energy-conservative pipeline innovations which are most energy-effective and cost-effective, and the formulation of recommendations for the R, D, and D programs needed to exploit those opportunities. From a candidate field of over twenty classes of efficiency improvements, eight systems are recommended for pursuit. Most of these possess two highly important attributes : large potential energy savings and broad applicability outside the pipeline industry. The R, D, and D program for each improvement and the recommended immediate next step are described.

Preface

Because of the wide scope of the subject matter of this study, and the consequent bulk of this report, we have attempted to so organize it that it may be conveniently read to three levels of penetration.

For the first level, the reader can rapidly understand from Sections 1.0 and 2.0, in less than a dozen pages, our objectives, conclusions, and recommendations. Many readers will have no particular need to read further.

Section 3.0 presents, in another two dozen pages, the broad outlines of the research, development, and demonstration (R, D, and D) programs which are needed to exploit the energy conservation opportunities which have been identified during the study. A brief summary of the rationale supporting each recommendation is also included, and many readers at that point will have satisfied their purposes and will not need to continue. Thus, the second level of penetration is reached by completion of one or more of the subsections of Section 3.0. To conserve bulk and weight, Sections 1.0, 2.0 and 3.0 may thus be separated from the remainder of the report and bound as a separate summary.

Because Section 3.0 is designed as a stand-alone chapter, it necessarily contains some duplication of material with the later sections. Thus, the reader who wishes to hear everything that we say on any particular improvement area(s) need only read the introduction to Section 3.0 and then proceed directly to the appropriate later chapter(s), as his interest may require.

We have been assisted in this study by many organizations, too numerous to acknowledge individually. However, we are attempting to remember each of them with a copy of this report. Comments, criticisms and additional information are earnestly solicited from these and all other readers, and should be directed to

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We pledge that all such feedback will be coordinated, analyzed, and submitted to our clients, Mr. Richard Alpaugh, Project Manager, Division of Transportation Energy Conservation, ERDA - Washington, and Ms. Estela Romo, ERDA - Oakland.

William F. Banks
James H. Horton

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1.0 OBJECTIVES

1.1 Purpose of the Project

The work reported here is a part of a project which is being conducted by the team of Systems, Science and Software (S^3) of San Diego, and Pipe Line Technologists, Inc. (Pipetech) of Houston, under ERDA contract E(04-03)-1171, "Energy Study of Pipeline Transportation Systems." The basic purpose of the project is to assess the susceptibility of the oil, gas, and other pipeline industries to energy-conservative technological innovations, and to identify the necessary research, development, and demonstrations (R, D, & D) to exploit those opportunities.

The project final report is being published as S^3 report SSS-R-77-3020, "An Energy Study of Pipeline Transportation Systems." That final report will be a summary, combining the results from the seven task reports listed in Table 1.1-1. As will be noted from the table, this present report is one of those task reports.

1.2 Purpose of this Report

This report presents the results of Task 3, which has two primary objectives :

(1) Identification of those potential energy-conservative innovations which are most energy-effective and cost-effective.

(2) Formulation of recommendations for the R, D, and D programs which are needed to exploit those opportunities.

1.3 Scope Limitations

This report does not treat the subject of R and D strategies and the selection of certain of the recommended projects for pursuit in preference to the others. This approach was not taken under any delusions regarding the ultimate necessity to think small, that is, to eliminate some projects, reduce others, and defer yet others, and these questions will

TABLE 1.1-1

Project Reports

SSS-R-77-	Title	Associated Task
3020	An Energy Study of Pipeline Transportation Systems	All
3021	Economic Models of Pipeline Transportation Systems	1 (partial)
3022	Energy Consumption in the Pipeline Industry	1 (partial)
3023	Slurry Pipelines - Economic and Political Issues - A Review	2.1
3024	Federal Regulation of the Pipeline Industry	2.2
3025	Efficiency Improvements in Pipeline Transportation Systems	3
3026	Recommendations for Energy-Conservative Research and Development in Pipeline Transportation (published only in combination with R-3025)	4 (partial)
3027	Energy Conservation Opportunities in the Pipeline Industry	4 (partial)
3069	S^3 Financial Projection Model - Preliminary User's Manual and System Overview	

be addressed in the final project report. For the present, however, we are still thinking big, that is, identifying opportunities to save large amounts of energy, and defining the programs that would permit the exploitation of those opportunities.

2.0 SUMMARY

Most of the energy conservation opportunities which are identified here possess two highly important attributes :

(1) The potential energy savings are large, that is, of the order of several hundredths of a quad.

(2) The technology whereby the energy savings may be realized is broadly applicable outside the pipeline industry.

An example of attribute (1) will be found in Section 4.1 below, where it is seen that bottoming engines and internal cooling of IC engines may each easily aspire, through ultimate industry-wide application, to savings well over a tenth of a quad.

An example of attribute (2) will be found in Section 4.3, where the somewhat startling, though still tentative, conclusion is derived that the pipeline application is likely to be a much more attractive breeding ground for fuel cell development than the one upon which virtually all of the money is currently being spent, that is, the electric utility application. Rather than a commentary upon misdirection of R & D, this observation merely reflects the fact that, for various reasons, the utility industry and its equipment suppliers have been energetic in persuading the government to support their R & D, while the pipeline industry has chosen to go its own way. It is, of course, the identification of just such opportunities as this which is the basic purpose of the present study.

2.1 Recommended Programs

Table 2.1-1 presents the eight programs recommended for pursuit. This study has concluded that each of the listed programs satisfies the first three of the criteria listed below:

1 - Energy-effective

2 - Cost-effective

3 - Technically feasible on a moderate (3-7 year) time scale

4 - Broadly applicable outside the pipeline industry.

Table 2.1-1

RECOMMENDED RESEARCH, DEVELOPMENT AND DEMONSTRATION

	Program	Report Section No.
1.	Gas-fired Combined Cycle Compresser Station	4.1.1.4
2.	Internally Cooled Internal Combustion Engine	4.1.2.2
3.	Methanol-Coal Slurry Pipeline	5.2.2
4.	Methanol-Coal Slurry-Fired and Coal-Fired Engines	6.1.1.2
5.	Indirect-Fired Coal Burning Combined Cycle Pump Station	6.1.1.2
6.	Fuel Cell Pump Station	4.3.6.2
7.	Drag-Reducing Additives in Liquid Pipelines	8.2
8.	Internal Coatings in Pipelines	8.3

Of the eight recommended programs, the first six on Table 2.1-1 additionally satisfy criterion 4, that is, only programs 7 and 8 are pipieline-peculiar.

For the benefit of the reader who does not require a full, detailed explanation, the recommended programs are succintly summarized below. Section 3.0 describes each recommended program in further depth.

It is recommended that the programs discussed below be undertaken. In each case, the program should be coordinated with other active and planned ERDA programs and with other government agencies previously and/or currently engaged in similar or related programs.

(1) Gas-Fired Combined-Cycle Compressor Station

A study and demonstration of a Brayton-Rankine and/or Otto-Rankine combined cycle power plant should be conducted using advanced second or third generation gas turbines with organic Rankine bottoming cycle. The program, including selection of engine type, size selection, should be primarily oriented to pipeline applications but with broad application potential to utilities and other industrial work.

(2) Internally Cooled Internal Combustion Engine

A study and demonstration of an internally cooled reciprocation internal combustion engine, with bottoming cycle, should be conducted. The objective should be to develop the necessary technology to retrofit existing reciprocating pipeline drivers, and at the same time lay the groundwork for development and application of internally cooled internal combustion engines for a broad spectrum of non-pipeline applications.

(3) Methanol-Coal Slurry Pipeline

A study and demonstration of methanol-coal slurry pipeline should be conducted. Their potential for delivering coal over long-distance pipelines at relatively low cost, with low net water requirements, and with a broad spectrum of end use options is extremely attractive. Conversion of existing pipelines to coal-oil slurry transport may offer a convenient transition to the coal-methanol mode, and a brief comparison of the two systems should therefore be made.

(4) Methanol-Coal-Slurry-Fired and Coal-Fired Engines

A study and experimental work should be conducted to assess the potential for operating gas turbines on methanol-coal slurry and on pulverized coal separated from slurry. The program should supplement previous investigations of hot corrosion and erosion problems in turbines and should include work to identify the fouling mechanism, means of inhibiting fouling, and approaches for minimizing the effect of ash deposits on erosion of turbine hot-end components. The work should be closely coordinated and compatible with the study recommended in 3.3 above relating to methanol-coal slurry pipelines.

(5) Indirect-Fired Coal-Burning Combined Cycle Pump Station

A study and demonstration should be conducted of fuel cell power sources in combination with DC motors in a liquid pipeline pump station. The pipeline application offers a unique and extremely attractive application for fuel cell power plants for two reasons. First, the use of DC motors in the pipeline application would avoid the need for inverters to convert the fuel cell output to AC. Freed of this burden, the economics of the pipeline application for fuel cells becomes much more attractive than the electric utility application. Second, the nature of petroleum products pipeline operation is such that the use of DC motors would enable an energy saving in the order of 5-10%. The fuel cell, of course, offers the opportunity to realize that improvement. The combination of these two factors indicates that the pipeline is the preferred application for early commercialization of the fuel cell. Accordingly, it is recommended that continuing ERDA fuel cell programs be reconsidered in this light.

(6) Fuel Cell Pump Station

A study and demonstration should be conducted of high efficiency, indirect-fired gas turbines with Rankine bottoming cycle, using pulverized coal fuel. The program should capitalize on existing technology in closed Brayton cycle engines which have operated successfully on coal in Europe, but at relatively low efficiency because of limited cycle temperatures. Effort should be concentrated on achieving substantial increases in efficiency through the use of advanced,high-temperature materials in the air heater and the addition of an organic Rankine bottoming system.

(7) Drag-Reducing Additives in Liquid Pipelines

Further research should be conducted on drag-reducing additives for liquid pipelines, including: basic research into the mechanism of drag reduction; system studies to identify operating problems and assess economic aspects; and a demonstration to prove the soundness of the concept in practical pipeline operation.

(8) Internal Coatings in Pipelines

Demonstrations should be conducted of internal coatings in both gas and liquid pipelines to determine quantitatively their effect on improving pipeline flow efficiencies and to assess the economic potential of their further use in liquid pipeline. The program should begin with analysis and testing to establish the longevity and dependability of present commercial coatings which are applied in place, followed by research and development if necessary, and a demonstration in a full station-to-station section of an operating pipeline.

2.2 Candidate Improvements

Early in the study the categories of efficiency improvements listed in Table 2.2-1 were identified as candidates for examination. With few exceptions the numbered improvements represent classes of devices rather than a single device. Thus, even though an improvement category may not appear in the earlier list of recommended programs, it may well be that future inventions will justify a program in that category. For example, this study was unable to identify an electric motor improvement which would be appropriate for ERDA support for pipeline application. However, the possibility always exists that a new idea and/or fresh approach, e.g., the Wanless motor, may appear and offer an opportunity for energy savings.

Additionally, it is well to note that some of the improvements in Table 2.2-1, for example, capsule pipelines, were excluded from Table 2.2-1 on the basis of very preliminary analyses. This exclusion could safely be made at this time because, even if that preliminary conclusion is reversed after further study, the realization of the improvement is still well into the future. Accordingly, it is strongly recommended that the list and the associated conclusions be maintained by recurrent review and update, so that future opportunities can be exploited as they appear.

TABLE 2.2-1

Classification of candidate efficiency improvements

Heat Engine Improvements

1 - Bottoming engines
2 - Gas turbine regenerators
3 - Internal cooling of internal combustion engines
4 - Slurry-fired engines
5 - Coal-fired engines
6 - Indirect-fired, coal-burning engines

Non-Heat Engine Energy Conversion

7 - Fuel cells
8 - Electric motor improvements

Flow-Inducer Improvements

9 - Pump improvements
10 - Compressor improvements

Slurry Pipelines

11 - Coal-water system improvements
12 - Coal-methanol systems
13 - Cryogenic systems
14 - Pneumatic slurries

Drag Reduction

15 - Heating
16 - Additives
17 - Internal coatings

Leak Prevention

18 - Internal coatings

Operational Improvements

19 - Automatic control of transients
20 - Computerized optimization of duty cycles
21 - Improved capital utilization

None of the Above

22 - Capsule pipelines

3.0 RECOMMENDED R,D,&D PROGRAMS

This section presents the broad outlines of the programs of research, development, and demonstration which are needed to realize the energy-conservative potential of the technological innovations that are discussed in later sections of this report.

In each case the recommended program consists of the six phases which are described below:

<u>Phase 1 - Identification of the Opportunity</u>

This phase has been in progress under the present study, and with the publication of this report is now complete.

<u>Phase 2 - Concept Validation</u>

In this phase the concept is validated analytically and a definitive program plan is developed.

<u>Phase 3 - Research and Development</u>

The research and development identified under Phase 2 is performed in accordance with the approved R&D plan. The R&D program continues through the design phase (Phase 4) and possibly beyond.

<u>Phase 4 - System Design</u>

A preliminary design is prepared of the demonstrator system on the selected site.

<u>Subphase 4.1 - Site Selection</u> - Several candidate sites are identified and the one best meeting the criteria developed under Phase 2 is selected.

<u>Subphase 4.2 - Preliminary Design</u> - A preliminary design is prepared of the demonstrator system on the selected site.

— - Subphase 4.3 - Detailed Design - The detailed design of the demonstrator, consisting of drawings and specifications suitable for use in construction, are prepared and documented, along with supporting design analyses. This subphase is paced by the progress of the research and development under Phase 3 above, which will have been proceeding concurrently with Phase 4.

Phase 5 - Demonstrator Construction

The demonstrator system is constructed. Data regarding construction costs, design feedback, and technological difficulties are collected, analyzed, and published in accordance with the program plan which was developed under Phase 2.

Phase 6 - Demonstrator Operation

The demonstrator system is operated in accordance with the program plan. Data regarding the operation is collected, analyzed, and published in accordance with the program plan.

The costs of these programs vary widely. The most widely varying component is the research and development, Phase 3. In some cases, for example the viscosity-reducing additives discussed in Section 3.8 below, a few hundred thousand dollars may be adequate. In other cases, for example internal cooling of IC engines, the R&D may run into the millions.

As noted above, Phase 1 has been identified in each case as the opportunity identification under this present study. Thus, the next phase for each of the programs is the concept validation phase, which is expected to consist of major subphases having the following objectives:

1) To validate the concept analytically and, in some cases, experimentally.

2) To prepare the system assessments which establish feasibility from several major viewpoints, e.g., technical, economic, environmental, etc.

3) To prepare any additional supporting studies and assessments which may be needed or desired, e.g., alternate approaches, alternate national policies, etc.

4) To identify the necessary experiments and tests and to formulate the plan for their performance

5) To define the complete R,D,&D program and to formulate the complete detailed program plan.

Figure 3.0-1 presents a generalized Phase 2 schedule, in which the effort to achieve each of these objectives is designated as a subphase. The major second-tier subphases (tasks) under the primary subphases are also shown. The 12-month period indicated for completion may be shorter or longer for any specific program, depending upon its nature and the availability of funds. The magnitude of the total Phase 2 effort depends on the nature of the necessary input and of the laboratory experiments (Subphase 2) and/or supporting assessments (Subphase 3). In some cases, an adequate Phase 2 may be performed within two person-years of effort, while in others, four or five person-years may be necessary. For preliminary budgeting and planning an estimate of approximately three person-years is suggested.

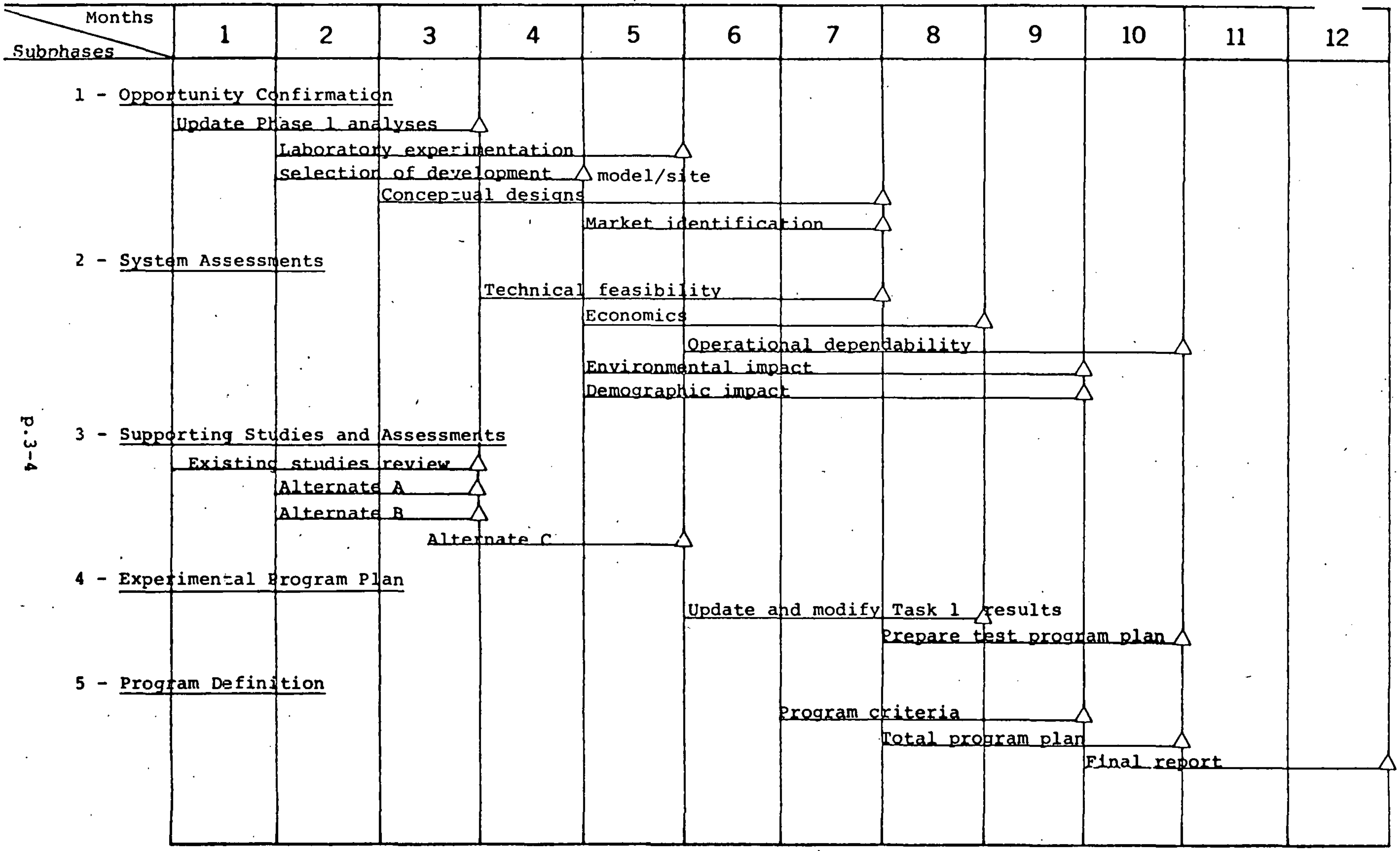

Fig. 3.0-1 - Generic Phase 2 Schedule

3.1 Gas-fired Combined Cycle Compressor Station

It is recommended that a study and demonstration of a Brayton-Rankine and/or Otto/Rankine combined cycle power plant be conducted. An advanced second or third generation gas turbine should be used as the primary power source for the Brayton-Rankine cycle power plant, and a modern gas reciprocating engine, representative of those used in pipeline service, as the primary source for the Otto/Rankine plant. For reasons discussed in Section 4.1.1.4 below, the bottoming system in either case should be of a Rankine type using an organic working fluid. The program, including selection of engine type and size, should be primarily oriented toward pipeline applications. However, in most cases broad application of the newly developed technology to other industrial uses is possible, provided that the R,D,&D program is planned and executed with those applications as secondary objectives.

As is discussed in Section 4.1.1.4, both the organic fluid Rankine cycle and the steam Rankine cycle offer potential for substantial increases in overall thermal efficiency through waste heat recovery. However, the organic Rankine bottoming cycle offers significantly greater improvement and appears to be more cost effective, both for gas turbines and reciprocating engines. With available technology, the organic Rankine system provides over one-third more power than a comparable steam plant, with an organic turbine that is less than one-half the size of a steam turbine.

Of course, it is possible to advance the technology of steam turbines, and at least one manufacturer has a development program in progress to accomplish this objective. However, the discussion of Section 4.1.1.4 shows that the superiority of the organic system derives from the thermodynamic properties that may literally be designed into the fluid, and is independent of the technology of the expander.

For first generation gas turbines, which typically have thermal efficiencies in the 18-22% range, bottoming cycles offer only marginal improvement over regeneration as a means of increasing efficiency, and are therefore not generally cost-effective, even at current and projected fuel costs. However, for second generation turbines, which have efficiencies in the 25-30% range and are commonly used in the gas pipeline industry, the bottoming cycle offers distinct advantages. This is also true for third generation turbines (simple cycle machines with over 30% efficiency), although they are in only limited use at present in pipelines. The primary reason is that turbine inlet temperatures have risen with succeeding generations to such a degree that even though the newer machines are more efficient, the exhaust temperatures have also risen. In these high temperature, high efficiency machines, the pressure ratios continue to increase as well, so that regeneration progressively loses attraction while bottoming gains. Projected efficiencies for the gas turbine-Rankine combined cycle are shown in Fig. 3.1-1. In a typical case, using a second generation simple cycle gas turbine with efficiency of 27%, an efficiency of well over 40% is projected in a combined cycle power plant with an organic Rankine bottoming engine.

Reciprocating engines, which constitute approximately half of the installed pipeline compressor power, are also strong

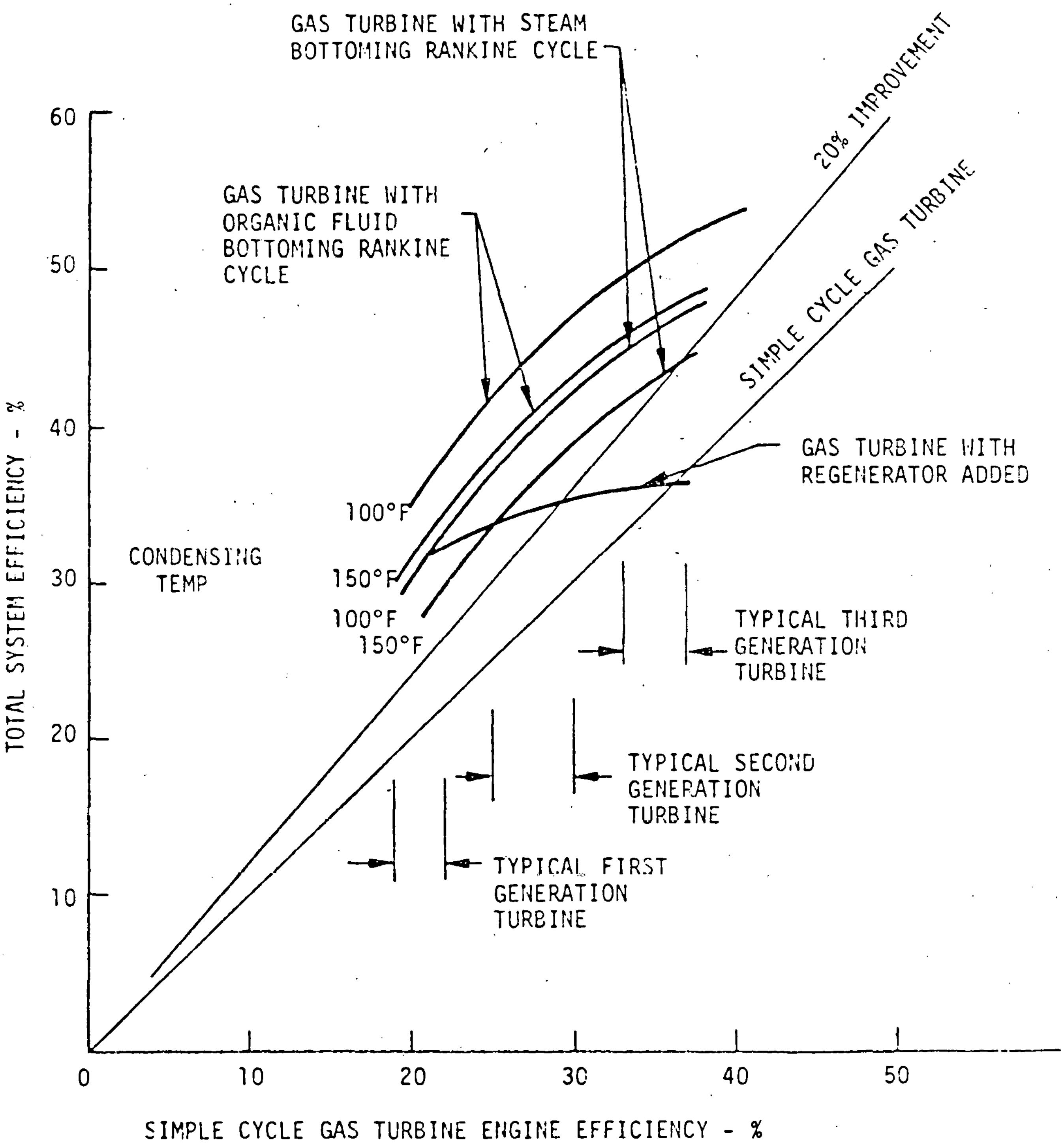

Fig. 3.1-1 Combined System Efficiency with Gas Efficiency

candidates for bottoming. Even with the relatively high efficiency reciprocators, organic Rankine bottoming cycles can realize performance improvements above 20%, so that large, nationwide energy savings are possible. This point is illustrated in Fig. 3.1-2.

In this connection, it is important to recognize that the relative heat rate improvement that may be achieved on a particular engine is not necessarily of great importance. Rather, total potential nationwide savings should be the primary objective, and the size of this total may be unrelated to relative heat rate improvement. For example, one might be inclined to conclude that because a 35% heat rate improvement can be achieved by bottoming a turbine as against, say, 20% for a reciprocator, that the turbine is a far better application. However, this is not necessarily true because it may be only a question of whether to add a 2000-hp bottomer to the turbine or a 1000-hp one to the reciprocator. And it may well be that more energy can be saved by putting more, though smaller, bottomers on reciprocators. These are the kinds of questions to be addressed in the system assessments under Phase 2.

A central feature of the study phase of this program should be a careful, balanced appraisal of the primary engine options. There are several major factors to consider in this appraisal. One is the early definition of criteria for the entire demonstration program. As has just been discussed, efficiency criteria should include not only the percentage improvement in heat rate of the engine to which the bottoming unit is applied, but the overall system efficiency and its effect on total energy savings across the pipeline industry. Broad applicability of the engine is another important consideration. The engine size and type should be representative of a broad range of pipeline applications and should also consider other applications to which the technology can be applied. System analysis and com-

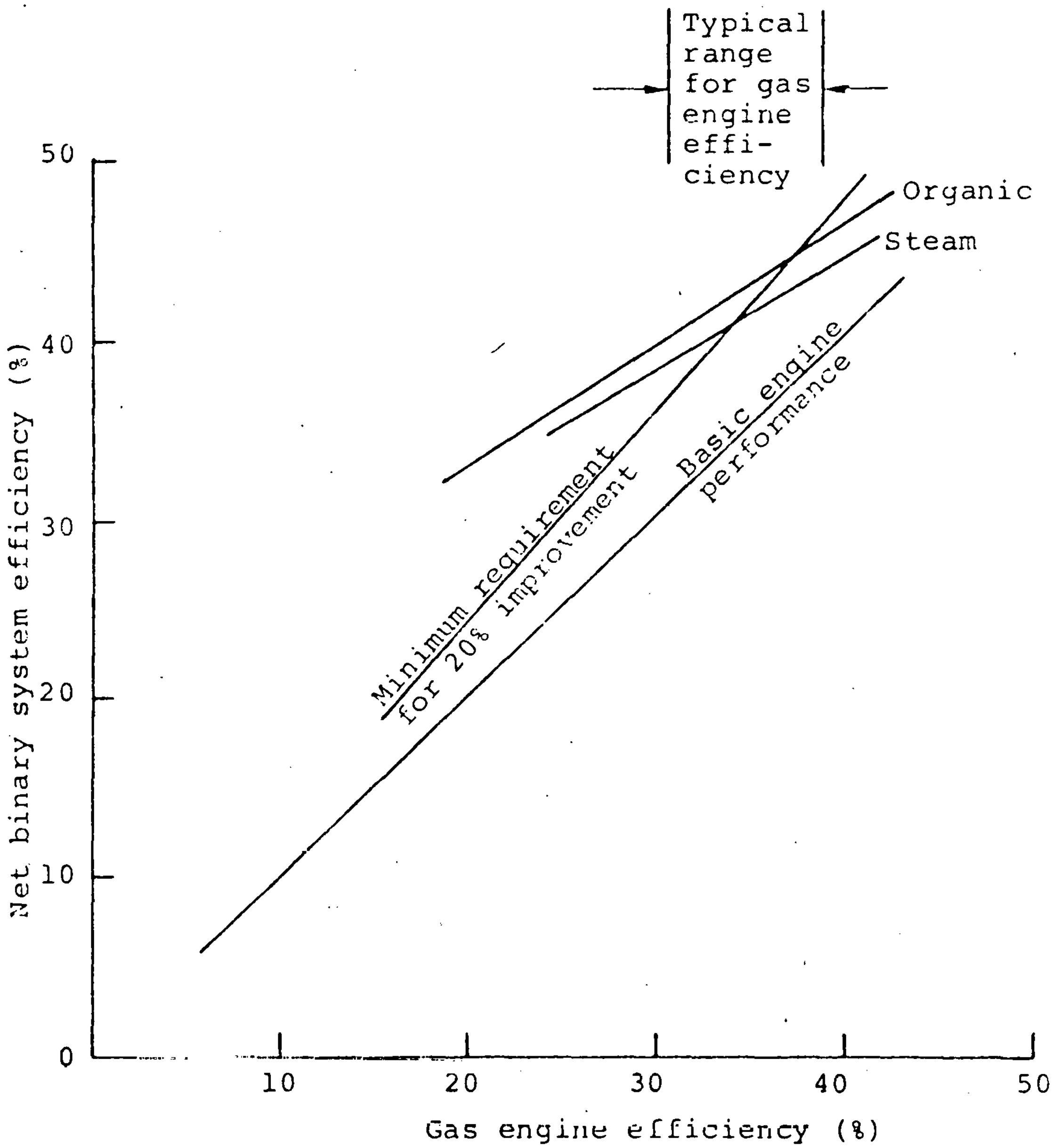

Fig. 3.1-2 Combined-cycle efficiency with gas reciprocators

ponent design and cost evaluations should include upsize and downsize variations around the baseline engine configuration, with a view of encompassing a product line to exploit a large portion of the total market.

For example, it might be concluded that the baseline for development should be a 1000-hp engine for bottoming large reciprocators, while the upsize model should be a 2500-hp version for large turbines and the downsize model should be a 300-hp version which could be applied to both reciprocating gas engines and to diesels. It is not suggested that a program to initially develop three engines be undertaken, but recognition should be given to the ultimate need for the other models so that the groundwork can be properly laid for their subsequent development.

Another important feature is site selection. It will be governed principally by the characteristics and commercial applicability of the engine which is selected for the bottoming cycle development. The site selection must also be dictated by the long-term needs of the program.

The risk areas in this program involve primarily the selection of the organic working fluid and the components in the bottoming system. The working fluid selection involves such major considerations as thermodynamic characteristics, long-term thermal stability, compatibility with materials used in structural components, safety characteristics (flammability, toxicity, etc.), freezing temperature, and cost. Other related risk areas are rotating machinery bearing and seal performance, turbine aerodynamic design, and boiler operation. All of these factors should be carefully assessed through analysis and through laboratory experimentation as necessary during the study phase of the program.

3.2 Internally Cooled Internal Combustion Engine

It is recommended that a study and demonstration be conducted of an internally cooled (adiabatic) reciprocating internal combustion engine for pipeline driver applications. The engine should also accept the bottoming engine being developed under recommendation 3.1 above. The objective should be to develop the necessary technology to retrofit existing reciprocating pipeline drivers, and at the same time to lay the groundwork for development and application of internally cooled internal combustion engines for a broad spectrum of nonpipeline applications.

This program, although similar in some respects to that described in Section 3.1 and related in that the bottoming engine should be installed on the internally cooled engine, is a separate and distinct effort and is recommended on its own merits as an additional program. Because of the fundamental nature of the engine development involved in the uncooled engine concept, it will require a very considerable research and development effort. The potential payoff in terms of increased energy savings, however, is so great that the investment is well justified. It should be possible, through a well planned and executed program, to develop a technology which will be applicable to the millions of diesel engines in service outside the pipeline industry. Thus it is concluded that the program is entirely justified in view of the tremendous potential for energy conservation.

As discussed in Section 4.1.2 below, the best potential for achieving significant gains in fuel economy of the diesel engine is to apply the engine in a combined cycle system utilizing either a Brayton or Rankine bottoming cycle to recover part of the exhaust energy.

The feasibility of the diesel-Brayton combined cycle (turbocompound) in an internally cooled 2-stroke diesel engine has been proved in research and exploratory development programs at Engine Systems, Inc., supported by the Army Tank-Automotive Research and Development Command (TARADCOM). Studies have indicated a potential reduction of 27-45% in specific fuel consumption, together with a substantial reduction in specific weight over comparable commercial diesel engines. A similar research program at Cummins Engine Company (also partially supported by TARADCOM) on a 4-stroke diesel adiabatic turbocompound engine with high-temperature ceramic components in the cylinders and no external cooling has indicated potential for similar large reductions in fuel consumption.

In natural gas reciprocating engines, the principle of using exhaust waste heat energy for space heating, air conditioning, and so on, has been applied in a number of on-site power (total energy) installations. Only limited effort to date has been directed toward adapting bottoming cycles to gas reciprocating engines to produce additional shaft power; however, the same principles as used in the diesel applications mentioned above can be adapted to gas engines, whether they are operated as dual fuel type (pilot injection of diesel fuel plus main change of gas) or as straight Otto cycle spark ignited engines.

As is discussed in Section 4.1.3 below, the heat balance data on a typical 4-stroke, naturally aspirated gas engine shows that approximately 25-30% of the input energy is rejected in jacket water cooling. If it were feasible to cool the engine internally in a manner similar to that described above for the internally cooled diesel engine, much of the heat could be recovered from the engine exhaust by a bottoming engine to produce additional shaft output. This of course is equivalent to turbocompounding the gas engine in the same manner as that discussed for the diesel engine. The expander for the bottoming cycle could in principle be either an open cycle gas power turbine such as used in the diesel programs described above, or a vapor expander (turbine or reciprocator) in a closed Rankine system such as that being developed under recommendation 3.1 above.

The internal cooling concept has not been explored in any depth for gas engines. However, from what has been said, it is seen that if internal cooling development can be conducted in a way that is applicable to both gas and diesel engines, the benefits would extend far beyond the pipeline industry. They would accrue in vehicular and marine transportation and in many stationary applications as well. As indicated in Section 4.3.1, if it were possible to retrofit half of the present population of reciprocating gas engines used in pipelines and if the economic value of natural gas escalates of over $2/Mcf, a saving of over $20 million per year could conceivably be realized for pipeline transportation alone.

As is indicated in Fig. 3.0-1 (Subphase 3), one of the first steps in the concept validation phase of the program is to conduct a detailed study of the presently used pipeline engines in order to select a suitable size and type of engine which could possibly be modified to the internally cooled principle. Recent statistics from the American Gas Assn. show that over 3500 reciprocating gas engines ranging in size from 1000 to 13,500 hp, are presently installed in gas pipelines. Over 70% of these engines fall in the 1000-2000 hp range. Of the total population of these engines, almost 80% are of the 2-stroke type. This suggests that the most representative engine to select for the R&D effort would be a 2-stroke engine in the 1000-2000 hp range. However, there are other considerations which may weigh in favor of the 4-stroke engine.

The 2-stroke engine compressors are manufactured by two major suppliers and are predominantly of the integral type, i.e., engines of in-line or V-angle configuration with integrally connected horizontal compressor cylinders. This could present some difficulties in converting the engine to a bottoming system wherein the compressor would be a separately driven unit.

Many of the 4-stroke engine-compressors in service are also of the integral type. Overall, however, there are more sources of separable engine-compressors in the 4-stroke category from which to select. The Phase 2 effort should determine the desirability, technical feasibility, and practicality of modifying any of the engine compressor types presently in service so that the additional power can be absorbed as additional compression work. If this approach is not practicable, an alternate approach would be to select one of the later design engines with better performance characteristics and utilize it as the vehicle for an internally cooled engine development to drive a separable compressor as well as other load devices.

The principal problem area in this program, once a suitable engine is selected for modification, lies in the adaptation of the internal cooling principle to the engine. Considerable research and exploratory development effort, preferably starting on a single cylinder version of the engine, will be required to determine the proper design and high temperature materials combination required to achieve performance objectives. A full scale engineering development and extensive reliability testing must follow in order to demonstrate a viable engine design for the commercial market. For those reasons, the program must be regarded as intermediate to long term in nature, with the first practical application likely in the late 1980's.

3.3 Methanol-Coal Slurry Pipeline

It is recommended that a study and demonstration of a methanol-coal slurry pipeline be conducted. The potential of this concept for delivering coal over long-distance pipelines at relatively low cost, with low net water requirements and with a broad spectrum of end use options, is extremely attractive.

Leonard Keller, President of the Methacoal Corp., Dallas, Texas, holds patents and disclosures relating to the methanol-coal slurry, which he calls Methacoal. He describes Methacoal as a pseudo-thixotropic, or shear-thinning, mechanically stabilized suspensoid. At rest it appears to be a moist, solid mass of black mud, but when stirred or agitated, it becomes thinner and flows easily. The flow characteristics are such that it is typified by viscous or laminar flow, rather than the turbulent flow which is typical of coal-water slurries.

The attractiveness of the Methacoal concept derives from its potential advantages in two broad areas:

1) Potential advantages of methanol over water as the carrier fluid of a coal slurry, and

2) Potential advantages of the methanol slurry as a fuel, or of the partially separated components of the slurry, over straight coal as a fuel.

3.3.1 Advantages of Methanol as Carrier

Methanol possesses two strong advantages, in principle at least, plus a lesser one, over water as the coal slurry carrier fluid:

1) For equivalent amounts of energy transported, the methanol slurry requires less water as input to the system than a water slurry system;

2) For equivalent amounts of energy transported, the methanol slurry consumes less energy than the water slurry;

3) Depending upon the methanol conversion process, most of the sulphur may not only be removed from the coal but may be recovered in usable commercial form. Thus, the sulphur content of a 2 to 1 slurry is almost one-third less than that of the feed coal, or almost one-fourth less for a 3 to 1 slurry. What would otherwise be a troublesome pollutant may be recovered as a marketable by-product.

These advantages result from several characteristics which are described in Sections 5.2.1.2 and 5.2.1.3 below, and are summarized briefly here.

1. High efficiency of coal to slurry conversion. While the direct conversion of coal to methanol is at most 50% efficient, the equivalent efficiency of conversion for the entire slurry system potentially approaches 90% when the slurry itself is considered as a fuel.

2. Reduction of system water requirement. For a methanol-coal slurry which is two-thirds coal, the carrying capacity advantage over coal-water slurry is calculated to be 3.2. For a three-fourths coal-one-fouth methanol system, this advantage becomes 4.35. Thus, the water requirement for the methanol slurry may be three to four times less than that of water slurry.

3. Benefits of dry coal. If the coal entering the slurrifier at the head of the methanol slurry pipeline is completely dry, several additional benefits would accrue, as indicated below. Some of these would be realized in the pipeline operation and some in the power plant.

a. The first benefit derives from the large increase in pipeline efficiency because it is no longer necessary to transport the water. For Western coals, which generally have high moisture content, this benefit can be very large. For example, with a coal having a one third moisture content, the efficiency of the pipeline is increased by 50% if the coal is dried at the head of the line. This increase is realized as a direct percentage increase in the number of Btu transported per Btu consumed in the transportation process.

b. The second benefit is the further reduction of the system water requirement. For carrying ratios in the range of two to three and moisture content in the range of 20% and greater, the net advantage of methanol over water as a carrier of dried coal is calculated to be in the range of four to five.

c. The third potential benefit from drying the coal is the possibility of recovering the drying energy from the methanol conversion process as electric power at what is equivalent to zero transportation cost.

3.3.2 Fuel-form Options

At the pipeline terminal many potential options are available for realizing the value of the methanol. All represent some advantage over plain coal, and a strong advantage over the wet coal that is separated from a water slurry. The principal options are listed below:

1) The slurry may be burned directly as a fuel in power plant boilers.

2) The slurry may be separated into powdered coal and alcohols which, in turn, provide fuel for several applications. The suboptions include:

a) After separation from the slurry, powdered coal may be used as feed stock for low-Btu gas plants in areas where water for the gasification is available, or for synthetic natural gas plants or ammonia plants.

b) The alcohols may be returned via a second pipe laid alongside the main line, to the head of the pipeline. The return line for methanol needs only a fifth or a sixth of the capacity that would be required for a water slurry.

c) The alcohols may be marketed as fuel-grade methanol for stationary engines. The market could include natural gas supplement, replacement for propane or butane, gas turbine fuel, additive to gasoline fuel, or be used directly as fuel in engines for automotive and industrial applications.

d) The alcohols may be marketed as vehicular fuel. If one looks ahead to the next century when petroleum can no longer supply most of the vehicular fuel, there appear to be two preeminent candidates for liquid, vehicular (ultimate) fuels: methanol and hydrogen. Of course there are many problems and obstacles to the adoption of either of these, which means that a great deal of research and development will be necessary to bring either concept to fruition. In the near term, the use of methanol in the pipeline offers the opportunity to find early answers to many of the questions relative to its potential as the ultimate vehicular fuel. That is, the principal objectives of the two R&D programs can be accomplished by funding only the smaller, more immediate of the two. Moreover, if a methanol pipeline is built, it would constitute a part of the demonstration program for the ultimate fuel.

e) The alcohols separated from slurry may be further separated into the basic constituents for subsequent marketing, including methanol, ethanol, n-propanol, and i-butanol.

3. The slurry can potentially be used directly as pipeline fuel. If it were feasible to burn the slurry in a gas turbine with a bottoming engine, the overall efficiency of the pumping process would then be approximately 50% greater than that of the electrically driven prime movers. The direct use of the slurry as prime mover fuel would render the slurry pipeline the most energy efficient of all coal transportation modes insofar as the consumption of mechanical energy of movement is concerned. When these two factors are combined in a system design and subjected to economic analysis, it may well be that the methanol-coal slurry is overall the most energy efficient mode of long-distance coal transport.

3.3.3 Problems and Limitations

It is recognized that there are several potential problem areas associated with this program. Three particularly sensitive questions associated with methanol-coal slurries will be discussed.

a) Engine fuel unknowns. Although the use of the slurry as boiler fuel does not appear to present any fundamental difficulties, the actual demonstration remains to be done. Its use in reciprocating engines may not be practicable because of the inherent problems of burning coal in these engines. Use of the slurry in open cycle gas turbines appears somewhat more promising but raises obvious questions regarding corrosion and erosion effects on turbine hot end components resulting from the coal constituent of the fuel. The problems of burning methanol-coal slurry and pulverized coal in gas turbines are addressed in another recommended program resulting from the initial study. It is important to recognize that even if the slurry cannot be burned directly as fuel, the concept is not necessarily invalidated.

b) Safety. Because the flammability of methanol-coal slurry is far greater than that of water-coal slurry, the safety implications of this new application must be examined. However, since the flammability of the methanol-coal is still much less than that of other pipeline fluids, it seems clear that safety considerations could not invalidate the concept.

c) Environmental impact. The environmental disruption resulting from a methanol-coal slurry pipeline spill is almost certain to be more undesirable than that from a coal-water slurry. Although this is unlikely to be a decisive negative factor considering that other flammable fluids are moved extensively by pipeline, the impact must nevertheless be examined.

d) Rheology of methanol-coal slurries. The very brief experimental work which was performed as part of this study yielded very encouraging results. However, they are far from conclusive, and the rheology of the slurry requires further work. The processes of coal communition and slurrification also require experimental exploration.

3.4 Slurry-fired and Coal-fired Engines

It is recommended that study and experimental work be conducted to assess the potential for operating gas turbines on methanol-coal slurry and on pulverized coal separated from the slurry. The program should supplement previous investigations of hot corrosion and erosion in turbines and should include work to identify the fouling mechanism, means of inhibiting the fouling and approaches.

As is discussed in Section 6.0 below, the conversion and use of coal has high priority in the President's National Energy Plan. Several large demonstration projects for converting coal to clean fuels are under way. Coal utilization programs are directed toward development of processes to permit increased use of coal by direct combustion, with the objective of developing and demonstrating on a commercial scale the direct combustion of high sulfur coal without exceeding pollution standards. These processes are fluidized bed combustors containing sulfur oxide sorbents. While the coal conversion and utilization projects in ERDA's Fossil Energy program have potential benefits in the overall spectrum of power generation, there are alternative approaches which could have significant benefits more directly related to the pipeline industry. One of these involves the direct utilization of pulverized coal in liquid pipeline drivers with the coal brought to the pumping station in the form of slurries. Section 3.3 above presented a recommendation for a study and demonstration program on a methanol-coal slurry pipeline, pointing out as one of the advantages the possible use of the slurry as a fuel in gas turbine drivers for pipeline pumps.

The problems involved in burning pulverized coal in gas turbines are discussed in Section 6.1.1.2 below. Mention was made also of a recent investigation at Solar Division of International Harvester to identify the hot corrosion and erosion problems of a gas turbine burning coal in a fluidized bed combustor, and of an ERDA-sponsored program at Curtiss-Wright to demonstrate the feasibility of a gas turbine to burn high sulfur coal economically in utility service.

The work at Solar has indicated fouling, resulting from volitalization of fly ash at temperatures above about 470°C, to be the principal deterrent to the use of coal-fired gas turbines, although erosion may also become significant in the lower temperature turbine stages. It was concluded that further work to improve the potential of operating gas turbines on coal must identify the principal contributor to the fouling mechanism, and then investigate strategies for mitigating the ash deposits. A continuation of the work is recommended in view of its potential benefits for pipelines as well as many other applications.

The investigation of methanol-coal slurry as a fuel for open cycle gas turbines would provide a very useful supplement to the methanol-coal pipeline study recommended in Section 3.3 above. As discussed in Section 5.3.2 below, it has already been demonstrated that methanol has excellent characteristics as a gas turbine fuel. One approach in a methanol-coal pipeline might therefore be to separate the coal from the slurry and use plain methanol as fuel. A far more desirable approach, however, would be to use the methanol-coal slurry directly as fuel. This latter approach remains to be investigated.

It is anticipated that the objectives of this program could be largely accomplished through an analytical study, followed by laboratory rig tests and then a controlled laboratory test on a gas turbine engine of a suitable size and type for liquid pipeline use. If the results are favorable, the demonstration phase could be accomplished in conjunction with the methanol-coal slurry pipeline demonstration recommended in Section 3.3 above.

3.5 Fuel Cell Pump Station

It is recommended that a study and demonstration be conducted of fuel cell power sources in combination with DC motors in a liquid pipeline pump station. The pipeline application offers a unique and extremely attractive application for fuel cell power plants for three reasons.

First, the use of DC motors in the pipeline application would avoid the need for inverters to convert the fuel cell output to AC. Freed of this burden, the economics of the pipeline application for fuel cells becomes much more attractive than that of the electric utility application. Second, the nature of petroleum products pipeline operation is such that the use of DC motors in and of itself would enable an energy saving on the order of 5-10%. The fuel cell, of course, offers the opportunity to realize that improvement. Third, the potential efficiecy of the fuel cells is much higher than that of the generating and transmission systems which currently supply pipeline pumping power.

The combination of these factors indicates that the pipeline is the preferred application for early commercialization of the fuel cell. Accordingly, it is recommended that continuing ERDA fuel cell programs be reconsidered in this light.

Fuel cells have some unique characteristics which make them attractive for power generation. As discussed in Section 4.4.2, these include high efficiency at all ratings, 25 kw and above; good part-load efficiency; low noise, thermal, and chemical pollution; and considerable freedom in site selection. Waste heat recovery potential is another advantage which can be exploited in some sites for such thermal uses as process heat/steam absorption chillers, space heating, and water heating. Second generation fuel cells of the high temperature type, which are targeted for commercial introduction in the mid-1980's, may lend themselves to utilizaiton of waste heat in the form of bottoming cycles which can further enhance overall plant efficiency.

The primary problems associated with fuel cells, as indicated in Section 4.4.3, are cost and durability. Expensive electrocatalysts are required in the fuel cell stacks. Fuel processing equipment is complex and expensive, and subject to degradation with fuels which contain any contaminants. Inverters, which are required to convert fuel cell system output from DC to AC, are also expensive and do not yet exist in the sizes needed for large-scale power conversion. While the feasibility of attaining high efficiencies in medium to large size fuel cell power plants has been proved, extensive engineering development and demonstration remain to be done before economics, reliability, and durability are established.

Fuel cell development status is discussed in Section 4.4.4 below. The major development effort at present is being directed toward first generation fuel cell power plants, which are targeted for commercial introduction in 1980. These will be low-temperature systems of the phosphoric acid type with a heat rate of 9000-9300 Btu/kwh (36.7-37.8 thermal efficiency) and a projected capital cost (in 1975 dollars) of $250/kw if produced in meaningful quantities.

The major program around which the first generation effort is centered is the FCG-1 program, which is being accomplished by scaling up from kilowatts to megawatts in steps: (a) a 1-MW pilot plant, (b) a 4.8 MW demonstration plant, and (c) 26 MW demonstration plants, all to precede the manufacture of 26 MW FCG-1 power plants on a production basis. The 1-MW pilot plant has been built and tested successfully. The 4.8-MW demonstration plant is under contract, jointly funded by ERDA ($25 million), the Electric Power Research Institute (EPRI) ($5 million), and United Technologies Corp. ($12 million). Delivery is scheduled for mid-1978 and testing by early 1979.

Concurrent with the FCG-1 program, another major effort, called the RP114 program is under way, intended to broaden the application of fuel cells beyond near-term needs to result in a second generation power plant for the utility industry to be used in a wide variety of applications. Major goals for the second generation include commercial introduction by 1985, capital cost of $200/kw and a heat rate of 7300-7500 Btu/kwh (thermal efficiency 45.5-46.8%) using natural gas and clean liquid fuels. The second generation fuel cells will probably use molten carbonate electrolyte and operate at relatively high temperatures (500-700°C).

Pipeline application of fuel cells is discussed in Section 4.3.6 below, where it is identified as an attractive R&D opportunity. From analysis of duty cycles in product pipelines, it is seen that, depending on the characteristics of the pipeline and of the product being moved, throttling may be necessary at both maximum flow and a number of intermediate flows below the maximum. These throttling losses could be avoided if an economical, efficient, variable-speed motor were available.

The principal conclusion which emerges from the analysis just described is that the pipeline application is much more attractive for demonstration of megawatt-scale fuel cells than the electric utility application. There are two principal reasons for this. First, in the utility application, it is necessary to provide inverters because utility systems in this country all operate on 60-cycle AC, whereas pipeline pumps can be driven by DC motors. These inverters do not yet exist, so the requirement for them constitutes a heavy additional burden on the fuel cell development. Moreover, the inverters, when developed, are certain to be expensive, possibly costing as much as the fuel cell itself. Thus, fuel cells for the utility application will always be more expensive than for the pipeline applications.

Second, the amenability of the pipeline to DC motors also offers an energy saving of the order of 5-10% in and of itself by avoidance of the throttling action described above. An accurate, quantitative analysis to yield an estimate of energy wasted in the pipeline industry through such throttling would require writing a computer program with extensive input data from the various pipelines. Such detailed study is clearly impractical under the scope of the present investigation and in fact is unnecessary. However, an approximation, derived from the fuel and power costs of the 105 interstate pipeline companies (see Fig. 4.3-3 of Report R-3022 of this series) indicates a total wastage of about $10 million annually. This order of potential saving is clearly enough to justify an R&D program of several million dollars.

Another attraction of fuel cells for pipeline applications lies in the comparative efficiency of fuel cell power plants <u>vs</u>

electric utility power. The latter seldom reaches 25% after all power generation and transmission losses are taken into account. If the expected fuel cell efficiencies (up to 47% in second generation plants) are realized in commercial applications, the net efficiency of the fuel cell power source, even after deducting losses in conversion and transport of fuel, will still far exceed that of the electric utility power source.

It is not suggested that the ready adaptability of fuel cell-DC motors to pipeline applications is an established conclusion. Achievement of the goals set forth for large fuel cell power plants in the near term, particularly those related to cost and durability, will be a difficult and costly task. In addition, the availability of DC motors which will meet the pipelines' requirement of long, unattended, maintenance-free operation, must be recognized as a problem area which may require a significant research and development effort to resolve. However, the potential that clearly exists for significant energy saving justifies a strong recommendation that further work be done to define an appropriate R&D program for the application of fuel cells in pipeline service.

3.6 Indirect-fired Coal Burning Combined Cycle Pump Station

It is recommended that a study and demonstration be conducted on a high efficiency indirect-fired gas turbine with Rankine bottoming cycle, using pulverized coal fuel. The program should capitalize on existing technology in closed Brayton cycle engines which have operated successfully on coal in Europe, but at relatively low efficiency because of limited cycle temperatures. Effort should be concentrated on achieving substantial increases in efficiency through the use of advanced high-temperature materials in the air heater and the addition of an organic Rankine bottoming system. The program is further discussed in Section 6.1.1.2 below.

Development of closed Brayton cycle systems first began in 1939. The first power plant of this type was an oil burning plant manufactured by Escher-Wyss in Switzerland, and was placed in service in 1940. The first coal burning, closed Brayton system was an Escher-Wyss 2300-kw power plant, commissioned in 1956 in Germany, and this plant had accumulated 120,000 hours by June 1976. Altogether, some 14 plants ranging in size from 2 to 50 MW have been built in Europe, Great Britain, Russia, and Japan. A number of these plants, including several which use coal as fuel, have accumulated over 100,000 hours of operation. Turbine inlet temperatures have ranged from 650 to 750°C and plant efficiencies have been in the general range of 25 to 32%.

Although these closed cycle plants have demonstrated economic viability, utilities have been reluctant to install them on a broad basis, primarily, it would seem, because of the hitherto wide availability of clean fuel whose combustion gas can be passed directly through the turbine. With the present emphasis upon use of coal and with the advent in recent years of high-temperature ceramic materials, these plants deserve further exploration.

By using new high-temperature materials, such as silicon nitride, silicon carbide, or other ceramics in the heat exchanger, it may be feasible to increase turbine inlet temperatures by several hundred degrees and thereby increase the overall thermal efficiency by a substantial amount. The addition of a Rankine bottoming system using waste heat from the air heater would result in a further increase in the overall plant efficiency. It is estimated that with a turbine inlet temperature of 1000°C (which should be feasible with ceramic materials in the air heater) and an organic Rankine bottoming cycle, an overall efficiency of over 40% can be achieved in a plant of 2-MW capacity or higher. Another advantage of ceramic materials would be increased resistance to erosion and corrosion effects from combustion of pulverized coal.

This program is recommended independent of the program on methanol-coal-slurry fired and coal-fired engines described in Section 3.4, although they are complementary. With the strong emphasis on coal in the Nation's energy plan, it is important that other options be explored for possible use in the event that practical solutions are not developed for coal-burning, open cycle gas turbines.

Aside from problem areas associated with the organic Rankine bottoming cycle, which would be as described in Section 3.1, the principal risk in the indirect fired gas turbine would appear to lie in the air heater. The application of ceramics in this stationary heat exchanger should be less of a problem than in the hot parts of a gas turbine (nozzles and particularly the rotary parts), such as are being pursued by several of the leading turbine manufacturers. However, there are potential problems, such as differential expansion between the metal housing and ceramic core and possible fouling of the air heater surfaces from slag, which will require careful design and development to overcome.

3.7 Internal Coatings in Pipelines

It is recommended that a program be conducted of analysis, experiments, testa, and demonstrations of internal coatings in both gas and liquid pipelines to determine with good precision their enhancement of flow efficiencies, to demonstrate their longevity, and to assess the economic potential of their further use in liquid pipelines. The program should begin with analysis and testing to establish the longevity and dependability of present commercial coatings which are applied in place, followed by research and development if necessary, and a demonstration in a full station-to-station section of an operating pipeline.

Results of tests and applications over the past 30 years have proved that internal coating of pipelines is an effective method of increasing pipeline flow efficiency. In addition to increased throughput, internal coatings provide other advantages including: protection against corrosion; reduced cost of scrubbers, strainers, pigs, and other types of pipeline cleaning equipment; prevention of contamination from corrosive products; reduced maintenance and labor costs; protection of pipe interiors against accumulation of foreign materials; and reduction of leakage.

Epoxy-type internal pipe coatings are currently being used in pipe for transmitting dehydrated natural gas, wet gas, crude oil, sour crude oil, salt water, fresh water, and petroleum products. Thousands of miles of internally coated pipelines are in service, with pipe sizes ranging from 2 to 42 inches.

There are two principal methods of application that have been used for internal coatings: in-place, or *in situ*, and spraying. *In situ* coating is applied to lines already laid, thereby avoiding the welding problem, i.e., the burnoff of the coating that occurs at the joints when new sections of pipe are welded together. Spray application of internal coatings is accomplished on individual sections of pipe at the pipe mill with appropriately formulated epoxy type coatings following sound surface preparation such as abrasive blasting or acid cleaning.

When epoxy type internal coatings were first used in the late 1940's and 1950's, problems were encountered with obtaining good adherance of the coating to the pipe wall, with the result that the coating would sometimes peel off in sheets and clog the filters, scrubbers, regulators, and other equipment in the line. Manufacturers and applicators claim that this problem has been overcome by better surface preparation of the pipe. However, interviews with a number of pipeline operators were conducted as a part of this study, and it was found that considerable skepticism still exists as to whether a pipe can be sufficiently well cleaned in place to ensure perfect retention of the coating. This, of course, is precisely the kind of situation which ERDA R,D,&D programs are intended to address.

Current practice is generally to spray the pipe sections at the mill prior to laying the pipe, as it is less expensive and more reliable than in-place coating. The main drawback, as mentioned above, is that several inches of coating are burned off at the joints when the pipe sections are welded. In the case of wet gas lines or liquid lines, field cleaning or repair at the weld joint is usually necessary to prevent accelerated corrosion. Thus, much of the benefit of the coating as a friction reducer is lost, since the insides of the joints are the principal flow inhibitors. Also, the benefits of corrosion protection and leak stopping, which are most needed at the welds, are also lost. Thus, what is needed is a scientifically planned and well conducted demonstration program, in an actual, operating pipeline, to show that in-place coatings can in fact equal the durability of the mill-spray coatings, while overcoming the limitations and shortcomings of the former. Accordingly, a program of R,D,&D is recommended.

3.8 Additives to Reduce Viscosity

It is recommended that a program of analysis, experiments, tests, and demonstrations of viscosity - reducing additives be conducted.

(1) System studies and simulations should be performed. The operating problems and penalties must be identified, and their costs estimated. Additional capital equipment requirements must be identified, and similations conducted to assess the economics of drag-reducing additives.

(2) A facility should be found or established for evaluating and comparing performance of additives, and the analytical results of step (1) above should be confirmed.

(3) A demonstration program should be conducted on an actual, operating pipeline.

The question of scaling from laboratory to field equipment is important because the tendency is for reductions in drag to be much smaller percentagewise in field-size equipment than in laboratory-scale systems (Savins 1976). However, the problem has been addressed (Savins 1976) and those investigators at least now feel that scaling in fact no longer constitutes a problem (Savins 1977). With this recent progress, existing facilities, e.g., University of Tulsa, are adequate for step 2 above. Thereafter the program could move directly into field demonstration in an operating pipeline.

This program offers several attractions. First, because the technology is near, it can be brought to fruition quickly. Second, the liquid pipeline companies are suffering the effects of rising power costs now, as opposed to gas pipelines whose fuel costs are artificially regulated to very low levels, so that the economic incentives exist. Third, by virtue of both of these considerations, it should be possible to arrange for substantial cost sharing by the industry. Thus, it appears that an opportunity may exist to realize a significant saving quickly and with minimal investment of ERDA funds. Accordingly, it is strongly recommended that this program be rapidly pursued.

4.0 POWER CONVERSION IMPROVEMENTS

The majority of the drivers, or prime movers, in gas pipeline service are reciprocating, spark-ignited, i.e., Otto-cycle gas engines, fueled by natural gas from the pipeline. Prior to the Arab oil embargo, turbines had been entering service in increasing numbers, primarily because their lower maintenance was able to offset the higher efficiency of the reciprocators. In recent years, with the continued rise in gas prices, the turbines were the first to be taken out of service on those lines whose throughput has been declining. In an effort to reverse this trend, turbine manufacturers have initiated R&D programs to improve turbine efficiency. As will be discussed later, two such improvements appear to offer a highly promising opportunity, yet appear to be beyond the development ment reach of industry.

Almost all prime movers on liquid pipelines are electric motors. As will be seen, there do not appear to be any attractive energy conservation opportunities in motor improvements.

4.1 Heat Engine Improvements

There are of course many small ways to improve the efficiency of heat engines. The engine manufacturers are well aware of these potential improvements but do not incorporate them because they are not cost-effective under the existing fuel price structure. As fuel prices rise, more and more of these small improvements will be introduced. However, small improvements already under development by industry do not appear to offer attractive opportunities for ERDA-sponsored R&D. For the larger improvements which would justify ERDA-sponsored R&D, i.e., step function improvements in importance, it is necessary to address the basic thermodynamic cycle, as opposed to small improvements in the hardware. Such basic cycle improvements are discussed below for the three principal engine types, i.e., Brayton, Otto, and diesel.

4.1.1 Brayton Engine Improvements

The use of Brayton engines as gas turbines in gas pipelines has increased significantly in recent years. In particular, the two-shaft gas turbine driving a centrifugal compressor has gained increasing acceptance in the gas pipeline industry, for several good reasons.

(1) The gas turbine's natural characteristics match those of the centrifugal compressor. With the free power turbine coupled directly to the compressor load, both the turbine and the compressor have basically the same power to speed characteristics (see Fig. 4.1.1-1). The power absorbed by the centrifugal compressor varies approximately as the cube of the speed, and the output of the power turbine at best efficiency also varies essentially as the cube of the power turbine speed. The gas turbine and the centrifugal compressor can therefore operate at the same speed, thereby avoiding any necessity for reduction gearing and other complications. The normal pipeline operating line goes through the high efficiency points of the gas turbine (Fig. 4.1.1-2).

(2) The gas turbine has achieved a high degree of reliability-availability, longevity, and low maintenance. A continuing study by one of the major producers of industrial gas turbines in high use factor service has shown an average reliability of 99.5% and availability of 97.4%. A significant percentage of the units in service have accumulated over 60,000 hours of operation and several units have surpassed 100,000 hours of operation. Reliability and availability in this instance are defined as follows:

$$\text{Reliability} = \frac{\text{installed hours less unsched. outage}}{\text{installed hours}} \times 100\%$$

$$\text{Availability} = \frac{\text{installed hrs. less (sched. \& unsched. outage)}}{\text{installed hours}} \times 100\%$$

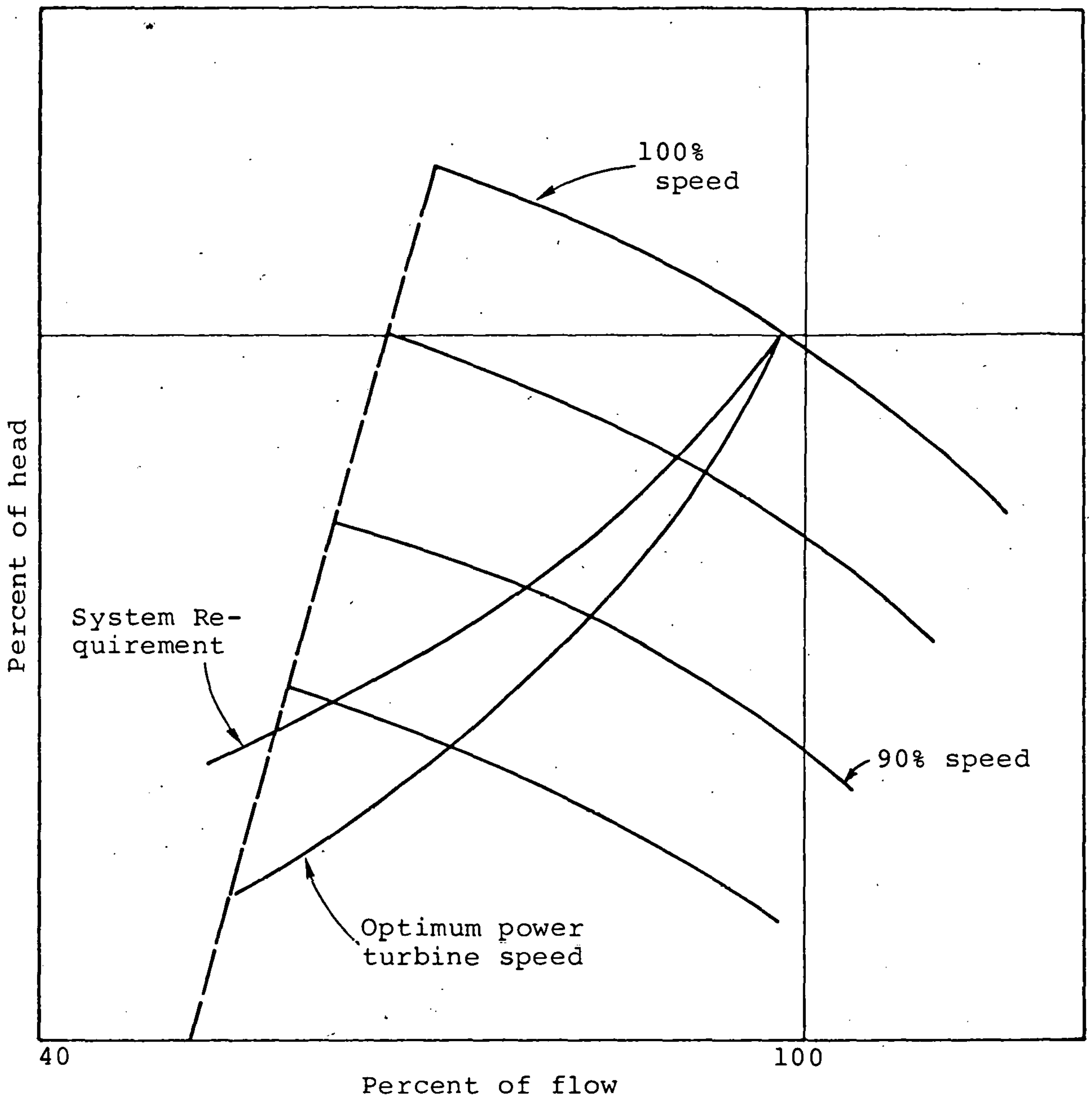

Fig. 4.1.1-1 Typical Centrifugal Compressor Characteristics

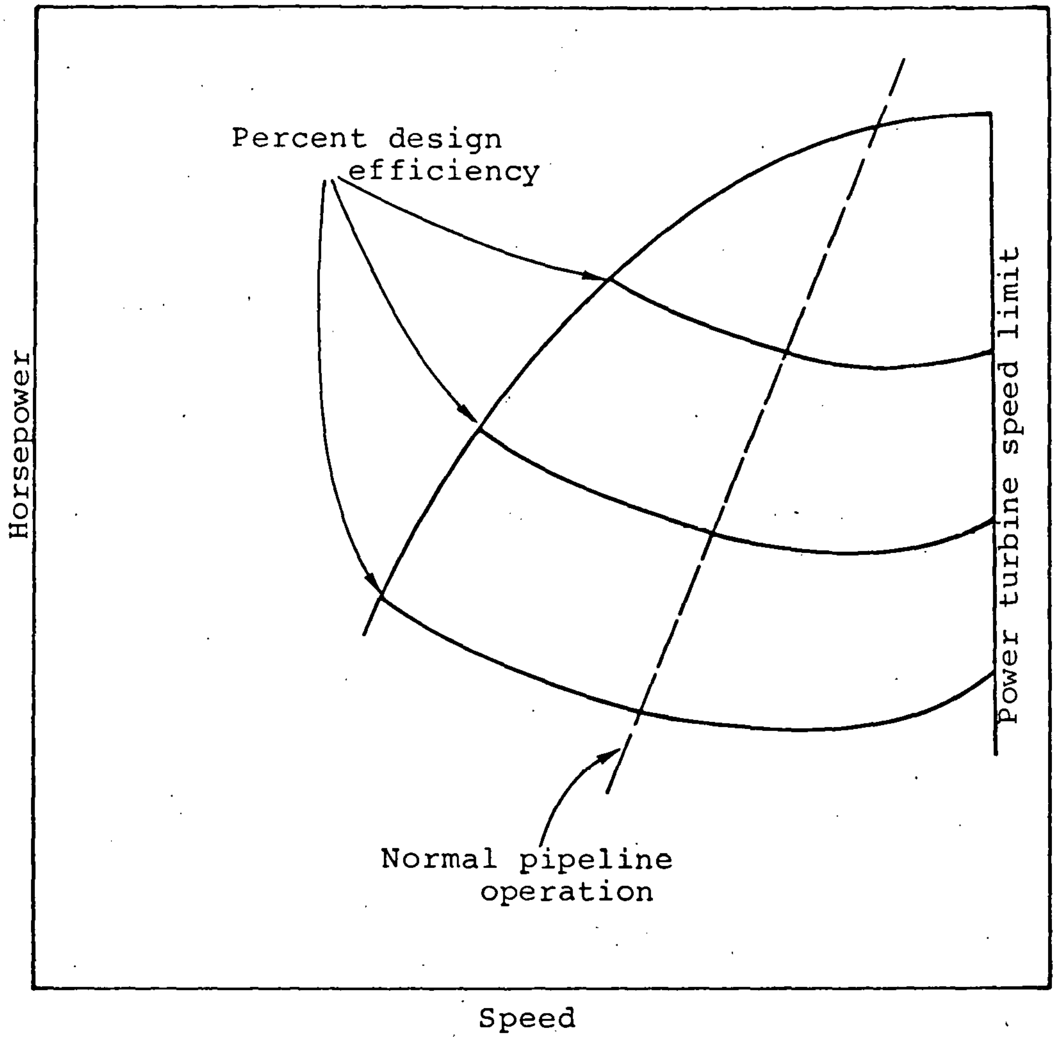

Fig. 4.1.1-2 Gas turbine performance match with pipeline requirements

(3) The average cost per installed horsepower of the gas turbine compressor has proved to be significantly lower than that of comparable reciprocating type units. For example, in 1973, cost statements filed with the Federal Power Commission by 35 pipeline companies covering installation of 503,163 compressor horsepower (54% of which were gas turbine driven) showed an average cost per installed horsepower of $219 for gas turbine units as compared with $295 per installed horsepower for reciprocating units. (Gas Turbine Intern'l, Jul-Aug'74)

The major disadvantage of the gas turbine as compared with the reciprocating engines used in gas pipelines is its higher fuel consumption. This, of course, is becoming a matter of increasing importance as the cost of fuel rapidly escalates. As a result, gas transmission companies as well as gas turbine manufacturers have been investigating ways of decreasing fuel consumption through modification of existing gas turbine installations and in applying gas turbines to new installations.

Advances in technology of simple cycle gas turbines have resulted in marked improvement in efficiency, with some of the newer industrial types approaching 30% overall thermal efficiency. However, with the cost of gas fuel increasing more rapidly than efficiency improvements in the simple cycle machines, it has become necessary to consider other approaches. The two foremost approaches for effecting major improvements in fuel economy are the use of regeneration, and the addition of a Rankine cycle bottoming plant to either simple cycle or regenerative cycle gas turbine power plants.

4.1.1.1 Regenerative Cycles

Figure 4.1.1.1-1 is a schematic representation of a regenerative cycle gas turbine power plant. The regenerator, which in the case of an industrial type gas turbine is a stationary heat exchanger (or recuperator), takes

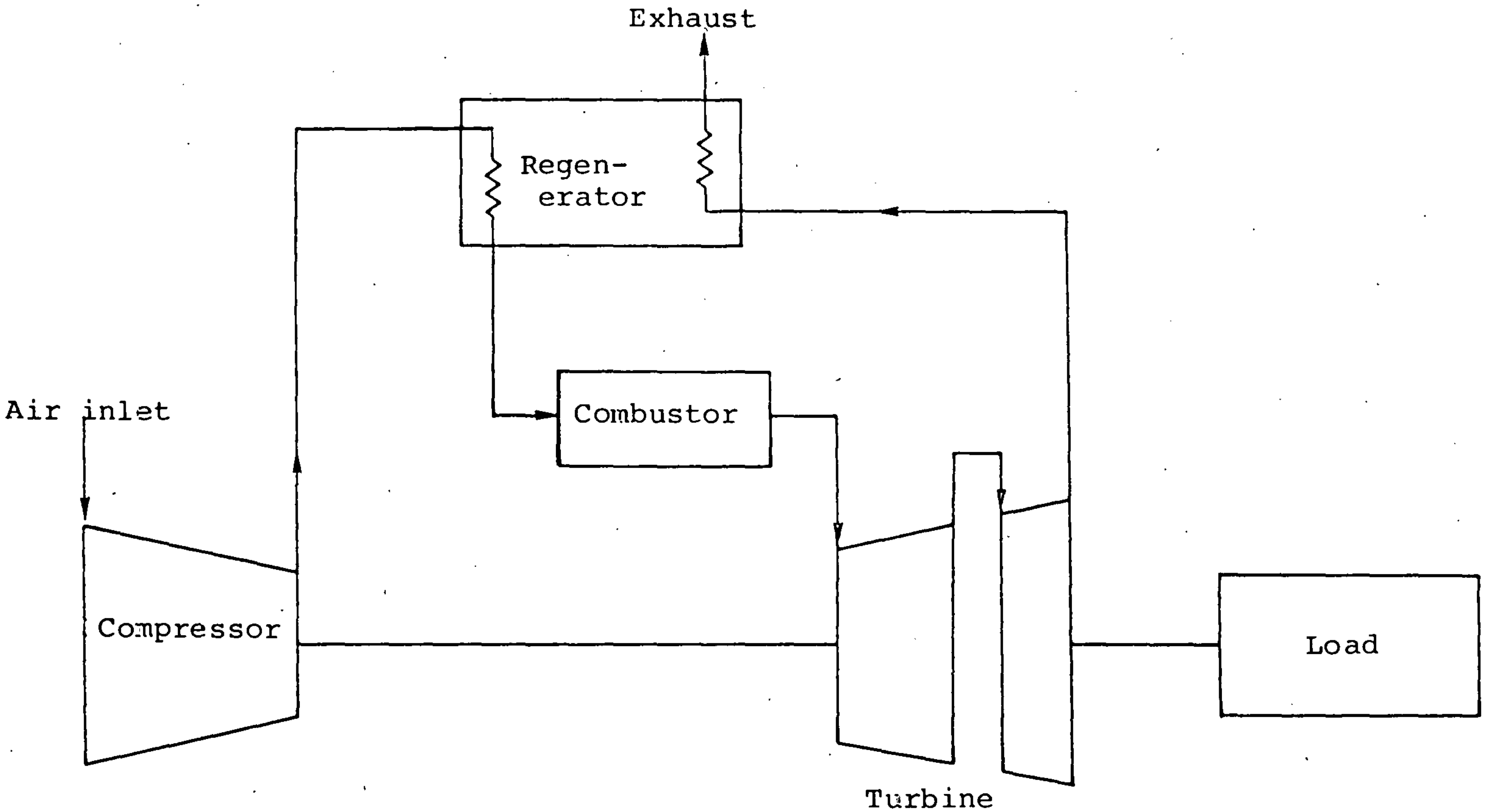

Fig. 4.1.1.1-1 Two-shaft Regenerative Cycle Gas Turbine

heat from the gas turbine exhaust and transfers this heat to the compressor discharge air before it enters the combustion chamber.

The use of regeneration is limited to gas turbine engines with moderately low pressure ratios. For a typical industrial gas turbine, a regenerator would be completely useless at pressure ratios of 14 or higher because the compressor discharge temperature is so high that it is equal to the exhaust temperature and therefore there is no chance for regeneration. The effect can be seen in Fig. 4.1.1.1-2. In most cases a pressure ratio of 6 to 8 represents a good compromise for both simple and regenerative cycle gas turbines, considering both fuel cost and installed first cost.

Addition of a regenerator of 80% effectiveness to a simple cycle two-shaft gas turbine increases thermal efficiency at full load by approximately 30%. Regenerators with an effectiveness of approximately 81% have been used with gas turbines for some years. Higher efficiencies can be achieved by adding more heat exchange surface to the regenerator, but only at the expense of considerable added bulk, weight, and cost. Figure 4.1.1.1-3 shows the effect of regenerator effectiveness on thermal efficiency and cost. When effectiveness is increased from 81 to 85%, the heat exchange surface must increase approximately 50%, and the additional regenerator surface costs about $150,000. Analysis shows that for fuel costs between $.50 and $1.00 per million BTU, the 81% is optimum. When fuel values reach $1.00 to $1.50 per million BTU, an effectiveness of 85% would be economical. (Heard, '76)

4.1.1.2 Brayton-Rankine Combined Cycles

Figure 4.1.1.2-1 illustrates a combined cycle power plant in which a Rankine cycle system is added as a bottoming plant to a simple cycle gas turbine.

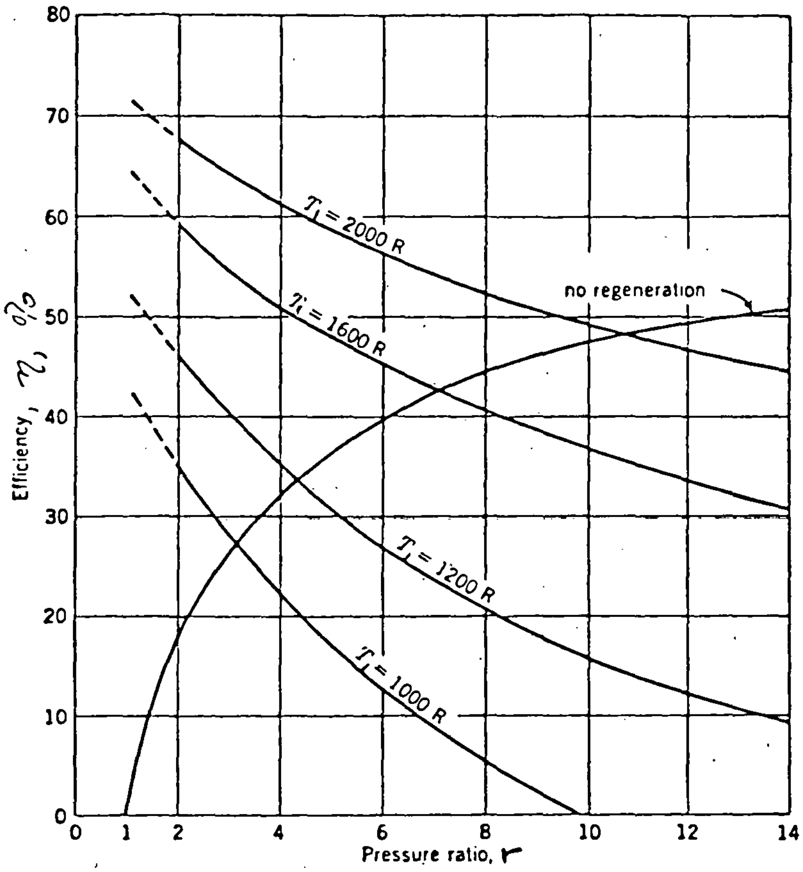

Fig. 4.1.1.1-2 Effect of pressure ratio in regenerative Brayton cycles

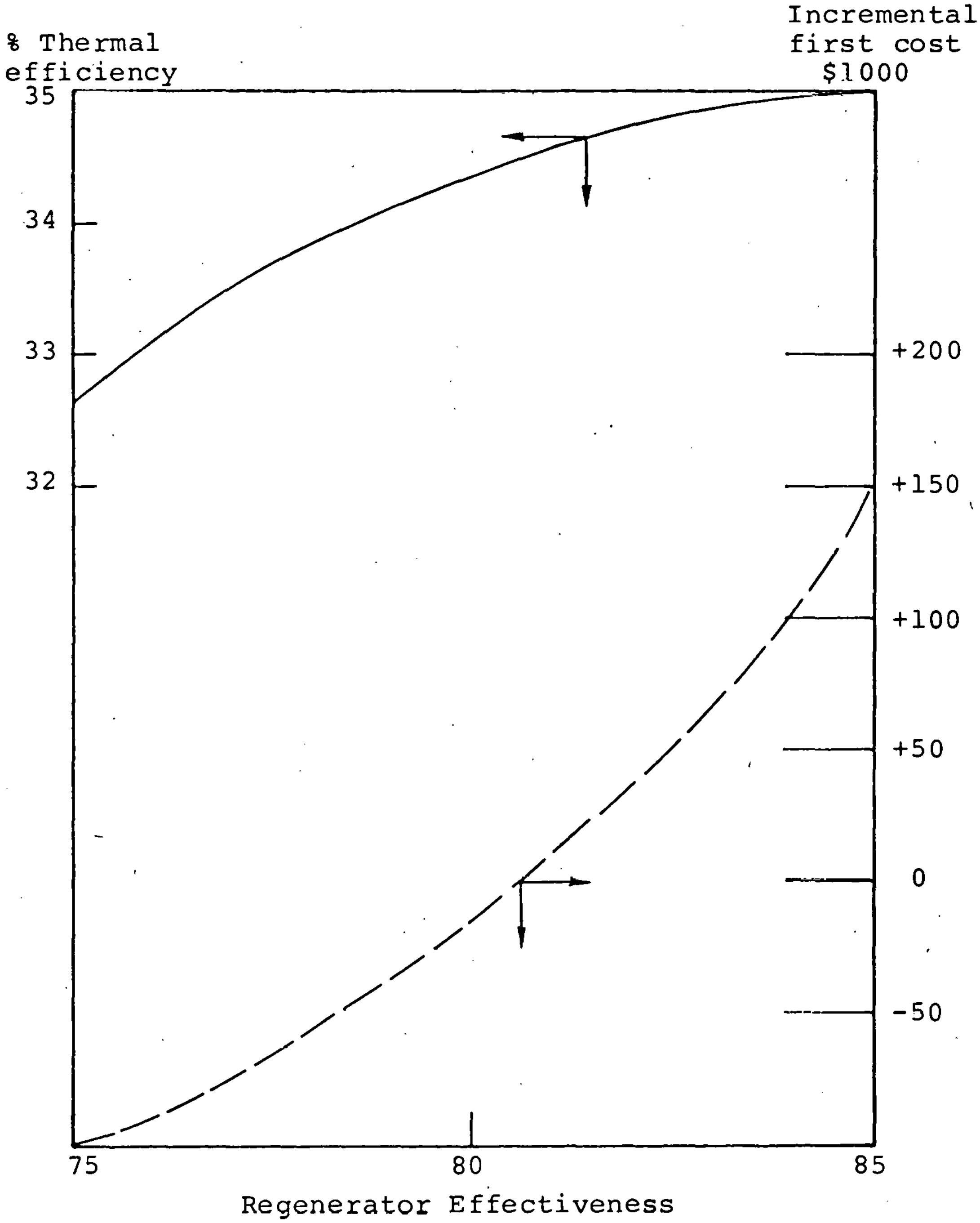

Fig. 4.1.1.1-3 Effects of regenerator effectiveness

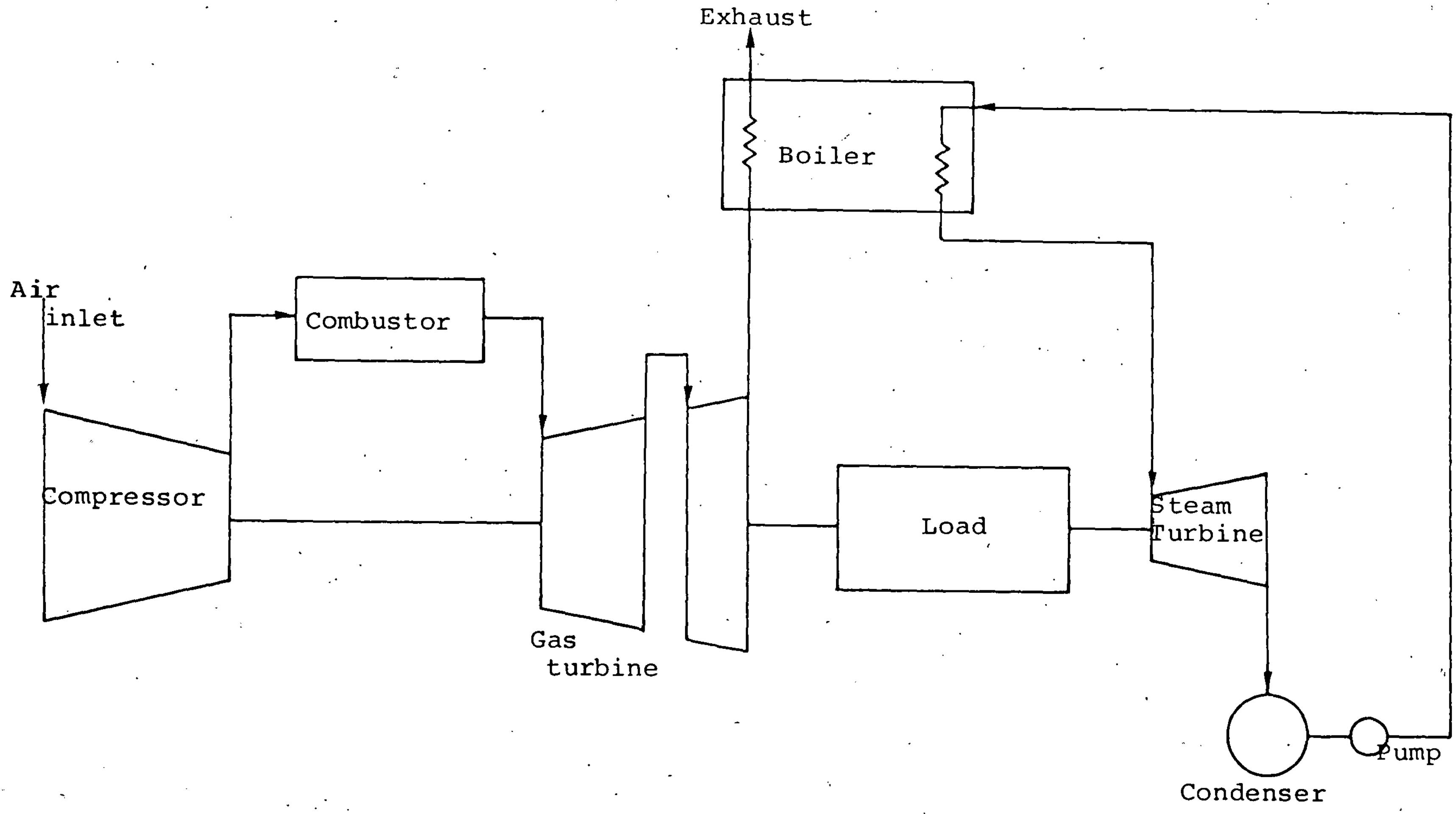

Fig. 4.1.1.2-1 Simple cycle gas turbine/Rankine cycle combined power plant

Exhaust heat from the gas turbine goes to a heat recovery boiler where vapor is generated and used to drive a turbine which is connected to the load. Exhaust from the vapor turbine is condensed and pumped back to the boiler to form a closed loop system. The single drive arrangement shown, with the vapor turbine directly connected to the outboard end of the load, is advantageous in a pipeline application, since the vapor turbine can be designed with an output speed to match the compressor and power output stage of the gas turbine. The gas turbine can also furnish the necessary speed control for both units under most conditions.

The number of combined cycle power plants installed to date is somewhat limited, and for the most part confined to large sizes in electric utility service. As far as this study could determine, only two plants of less than 15,000 KW capacity are in service in municipal utilities. The majority of the combined cycle plants in electric utilities are of the supplemental fired heat recovery type in which exhaust heat from the gas turbine is supplemented by separately fired burners in supplying heat to the boiler.

As for gas pipeline service, there are only four combined cycle systems known to be in operation. These systems were installed between 1968 and 1970, and are reported to be operating successfully; however, with recent advances in technology, it is apparent that system efficiencies can be increased considerably.

The combined cycle system of most interest for pipeline service utilizes a simple cycle, two-shaft gas turbine exhausting its heat to an unfired heat recovery boiler. Studies by the General Electric Co.(Heard,'75) have indicated that such a system, using a standard 14,600 hp gas turbine and a steam turbine connected directly to a centrifugal compressor, can produce a combined output of 22,000 hp at ISO conditions with an overall thermal efficiency of approximately 39.2%.

Figure 4.1.1.2-2 (Heard,'75) shows estimated thermal efficiencies at various ambient temperatures for the above combined cycle system using water as the working fluid. Steady state conditions typical of normal pipeline compressor operation were assumed. Thermal efficiency curves for a standard regenerative cycle gas turbine are shown for comparison. Two features of particular interest can be noted from these curves.

(1) The effect of varying ambient temperatures. At increaing ambients, the standard gas turbine efficiencies decrease significantly. For the combined cycle, however, the efficiencies increase at higher ambient temperatures. At the 100% flow point and 59°F, the combined cycle efficiency (39.2%) is approximately 15% higher than that of the standard regenerative gas turbine (34%).

(2) Part-load characteristics. The efficiency of the combined cycle plant decreases at a markedly higher rate at partial flows than that of the regenerative cycle plant. For example, at 85% rated flow and 59°F (a condition which requires 55% of the ISO rated horsepower), the combined cycle efficiency drops to 32% as compared to 30% for the regenerative cycle turbine. This would indicate that the combined cycle plant is utilized to its best advantage when operating at high load factors.

Solar Division of International Harvester Co. is actively investigating combined cycle power plants utilizing their industrial gas turbines in the 1200-10,000 hp range and expects to have initial field evaluation units in the 1978-79 era. Their studies have indicated that a combined cycle system for small gas turbines would be noncompetitive in initial cost, performance, and operating characteristics if it were designed around commercially available steam turbines, boilers, condensers, and related equipment. Accordingly, they are undertaking

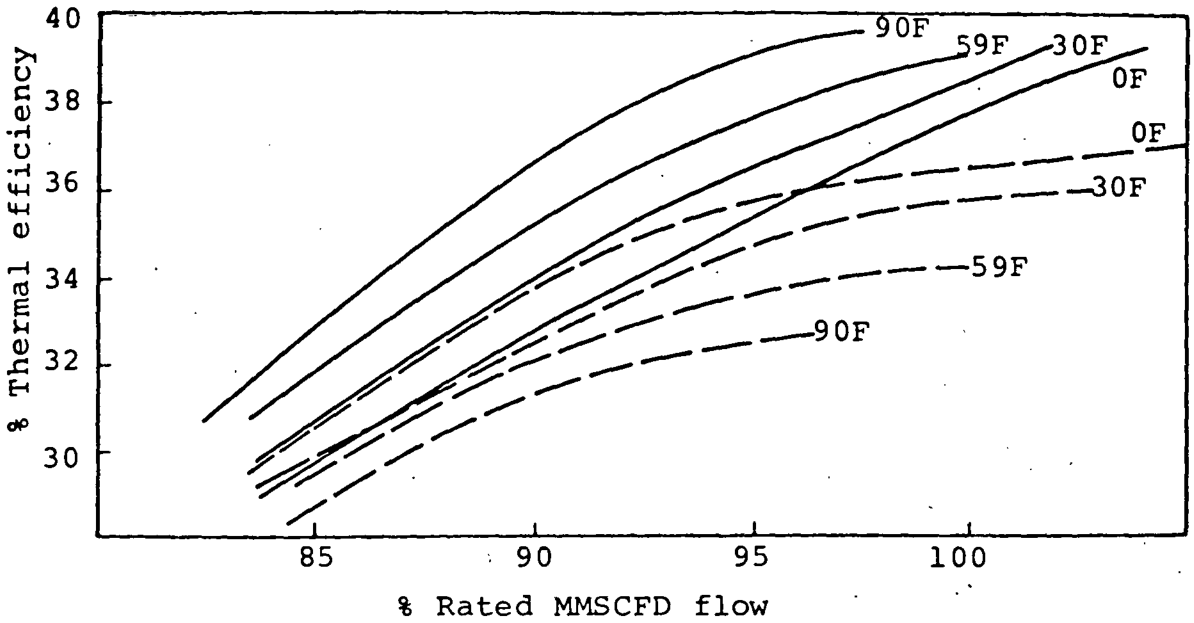

Fig. 4.1.1.2-2 Efficiencies of regenerative and combined cycles

—— Single compressor combined cycle

— — Standard regenerative gas turbine

the design and development of several major system components, described in a paper by Solar (Wardell et al.,'76). They include:

(1) A once-through boiler, in which heating, boiling, and superheating take place in one tube. Steam drums are not used and feed water flows are controlled as a function of output steam conditions.

(2) A new steam turbine having a projected efficiency of 80%. This compares with 66% in the best available commercial turbines. The new turbine concept is a 2-stage type with the high pressure unit running 66,000 rpm geared to a low pressure unit running at approximately 15,000 rpm, the same speed as the gas turbine drive.

(3) New condensers of both water-cooled and air-cooled type.

Utilizing these new components, Solar is projecting combined cycle power plants having thermal efficiencies ranging from 34.9% for the Saturn engine to 41.8% for the new Mars engine which is currently being developed. The engines cover a horsepower range from 1160 to 10,300 in the simple cycle versions, and in the combined cycle versions will cover a range of 1824 to 13,790 hp. The performance of the combined cycle power plants represents an increase in ISO hp from 34-57% and an improvement in fuel consumption of 25-38%.

The use of organic working fluids in the Rankine bottoming cycle offers potential for further improvement in efficiency of combined cycle power plants. Although all combined cycle plants constructed to date have used water as the working fluid, studies by Thermo Electron Corp. (Morgan et al.,'74) have indicated possible achievement of over 43% thermal efficiency in a simple cycle gas turbine/organic Rankine cycle bottoming plant, and over 47% in a recuperated version of this plant. These studies are based on the use of current technology gas turbine engines in large power sizes (over 60,000 kw). Other advantages of organic working fluids are low freezing temperature and possible reduction in size and weight, depending on the working fluid used. There are disadvantages in organic fluids as well, in that many of those that are readily available are flammable or toxic and all are subject to decomposition at moderately high temperatures such as might occur in exhaust heat recovery boilers. Nevertheless, there appears to be enough potential to warrant further investigation of organic fluids in combined cycle power plants.

4.1.1.3 Comparison of Gas Turbine Cycles

The decision to select a simple cycle, regenerative cycle, or combined cycle turbine power plant is sensitive to the duty cycle under which the plant will operate, the installed first cost of the equipment, the fuel cost, and the operating environment. With the rapid increase in fuel cost, the 25-28% thermal efficiency of the typical present simple cycle gas turbine is probably not high enough to make this power plant competitive with the other plants in the future.

As mentioned previously, the combined cycle power plant has a decided advantage in efficiency at the higher ambient temperatures. This would be an important factor to consider in selecting a plant to operate in a hot climate.

On the other hand, if the plant were required to operate a large part of the time at low load, the regenerative cycle turbine would appear to be more advantageous since the rate of increase in specific fuel consumption at part loads is less. This is illustrated in Fig. 4.1.1.3-1, which shows part load characteristics of typical gas turbine engines in simple cycle, regenerative cycle, and combined cycle versions of the type used in pipeline service.

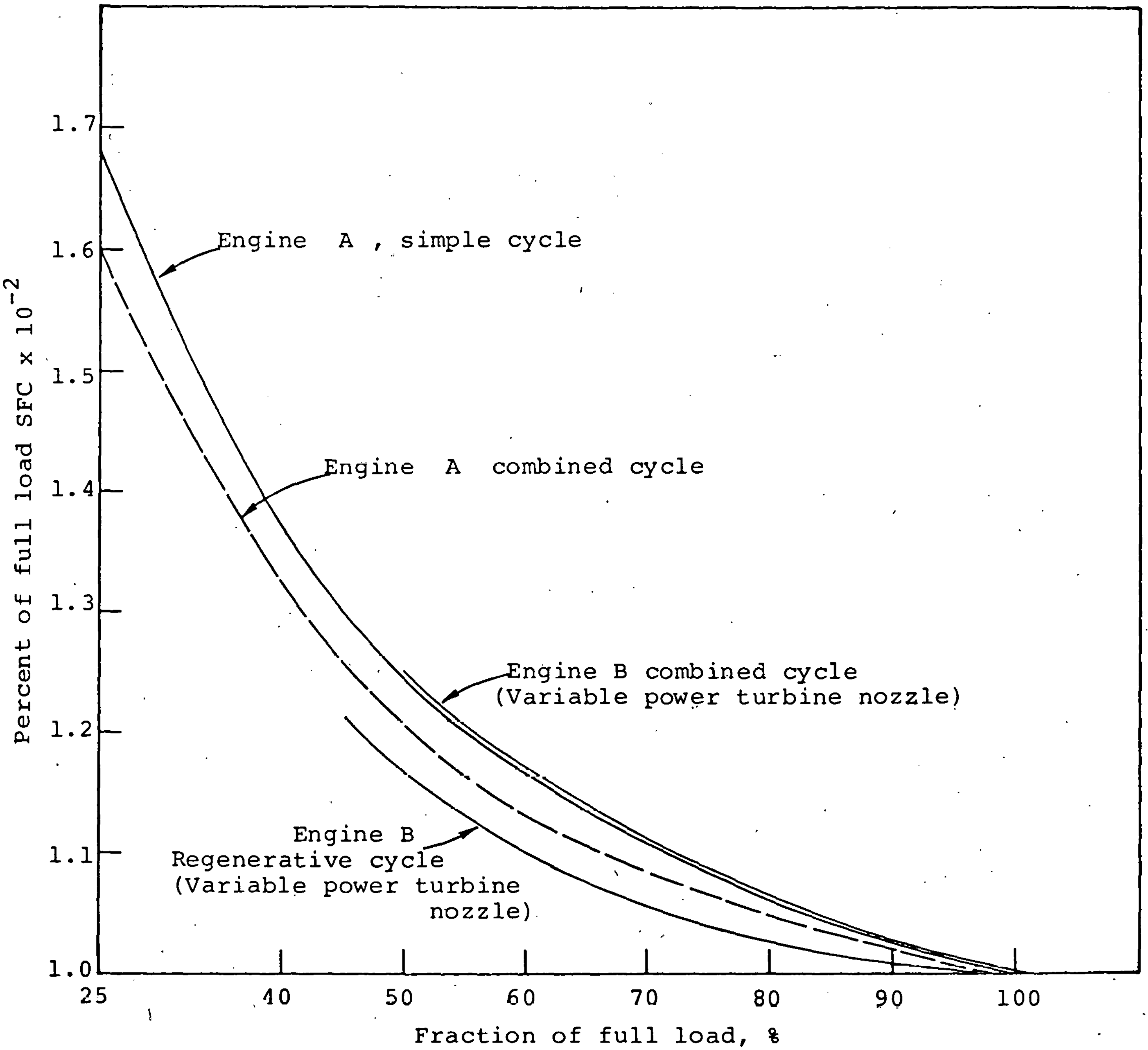

Fig. 4.1.1.3-1 Increase in specific fuel consumption at part load, typical two-shaft gas turbines

To compare these three cycles in a quantitative, precise way, detailed simulations were performed using the pipeline economic model (PEM) previously developed under Task 1 and described in Reports 3021 and 3069. The model was also introduced in Report 3024, Section 7.1. The model is a simulator which accepts as input the design characteristics of the pipeline, the operating and capital costs, and a market (throughput) projection. Its output is the detailed financial and energy consumption history over the life of the project.

The reference pipeline used in the comparison studies was designed by Pipetech and was based on earlier system designs from the Pipetech files. Some of these designs were actually built, and thus the reference designs for this study represent typical, realistic situations. The costs, derived by adjusting actual system costs, are therefore highly accurate in terms of this study. The PEM consists of two major submodels: a fluidics submodel and a financial projection submodel. The gas dynamics of the line are calculated using a (proprietary) model previously developed by Pipetech and used by them in the design of actual pipelines. It is therefore more than sufficiently accurate for this study. Figure 4.1.1.3-2 displays a typical output from the gas dynamics model. The financial projection model is an adaptation of a business projection model previously developed by S^3, modified to reasonably simulate pipeline operation, and bench marked against a highly detailed pipeline-peculiar (proprietary) financial model previously developed by Pipetech.

The reference gas pipeline was designed and costed on the basis of reciprocating gas engines, since that approach represents the best current practice. An installation schedule was prepared for that system, detailing the points in project life at which additional capacities must be installed. A survey of representative gas turbines was then made, and is

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*********************************
* PIPE LINE TECHNOLOGISTS, INC. *
*       STEADY STATE GAS        *
*      PIPELINE FLOW MODEL      *
*********************************

*** THIS IS THE ERLA EXAMPLE GAS PIPELINE ***
*** YEAR 3 ***
```

FLOWING PROFILE SUMMARY

	SEGMENT			INLET			OUTLET			AVERAGE			
NO.	FROM MILE POST	TO MILE POST	ELEV RISE FT	PRESS PSIA	TEMP DEG.F	GAS RATE MCF/D	PRESS PSIA	TEMP DEG.F	GAS RATE MCF/D	VELOCITY FT/SEC	PRESS DROP PSI/MI	STANDARD FLOW RATE MMSCF/D	LINE PACK MSCF
1	0.00	.00	0.	450.0	70.0	7139.61	450.0	70.0	7139.67	27.73	5.69	228.73	1.
2	.00	59.22	55.	920.0	124.6	4025.33	883.3	72.3	3413.93	14.54	1.80	226.80	56448.
3	59.22	88.00	29.	883.3	72.3	3413.93	831.0	69.0	3620.93	13.64	1.82	226.80	25240.
4	88.00	121.72	-9.	831.0	69.0	3620.93	765.9	68.3	3557.53	14.68	1.92	226.80	31840.
5	121.72	152.00	-6.	990.0	107.8	3306.67	940.4	75.3	3202.50	12.64	1.64	226.25	33112.
6	152.00	185.56	21.	940.4	75.3	3202.50	883.8	69.4	3374.65	12.75	1.69	226.25	36380.
7	185.56	250.23	40.	883.8	69.4	3374.65	762.7	68.2	3965.12	14.14	1.87	226.25	63247.
8	250.23	263.00	8.	990.0	108.3	3303.18	968.7	87.4	3198.63	12.63	1.66	225.69	13944.
9	263.00	315.89	-9.	968.7	87.4	3198.63	881.5	69.3	3375.76	12.74	1.65	225.69	57257.
10	315.89	382.48	-11.	881.5	69.3	3375.76	757.8	68.3	3584.10	14.17	1.86	225.69	64813.
11	382.48	399.00	-2.	985.1	108.6	3315.09	959.6	84.7	3200.44	12.66	1.64	225.12	16870.
12	399.00	450.00	-13.	959.6	84.7	3200.44	873.6	69.3	3400.59	12.79	1.65	225.12	55543.
13	450.00	518.39	-18.	873.6	69.3	3400.59	745.5	68.2	4046.24	14.33	1.87	225.12	65679.
14	518.39	567.71	-18.	969.2	108.5	3305.37	852.9	69.1	3483.40	13.26	1.68	224.56	71734.
15	567.71	658.00	-18.	852.9	69.1	3483.40	717.4	68.2	4210.80	14.78	1.93	224.56	65284.
16	658.00	686.00	-7.	932.6	108.3	3398.84	884.4	76.0	3403.07	13.40	1.72	224.00	26593.
17	686.00	759.00	-54.	894.4	76.0	3403.07	803.2	69.4	1859.20	9.90	1.13	112.00	49754.
18	759.00	784.00	10.	803.2	69.4	1859.20	771.0	69.4	1545.44	10.61	1.29	112.00	16773.
19	734.00	822.00	-39.	771.0	69.4	1545.44	682.2	68.8	2223.42	14.36	2.34	112.00	18106.
20	822.00	837.00	-15.	682.2	68.8	2223.42	643.3	68.7	2370.48	15.90	2.59	112.00	6458.
21	837.00	891.00	-15.	643.3	68.7	2370.48	474.4	68.3	3291.17	18.97	3.13	112.00	19470.
													800954.

DELIVERY SUMMARY

	MILE POST	PRESS PSIA	TEMP DEG F	GAS RATE MMSCF/D
1	0.00	450.00	70.0	-228.73
2	686.00	884.40	76.0	112.00
3	891.00	474.37	68.3	112.00
22	891.00	474.37	68.3	-.00

Fig. 4.1.1.3-2(a) Typical output, gas dynamics submodel

COMPRE SUMMARY

NO.	MILE POST	ELEVATION FEET ABOVE SEA LEVEL	SUCTION PRESSURE PSIA	SUCTION TEMP. DEG.F	DISCHARGE PRESSURE PSIA	DISCHARGE TEMP. DEG.F	COMPRESSOR OPERATING HORSEPOWER	FUEL REQUIRED MMSCF/D	COOLING REQUIRED TONS	STATION THROUGHPUT MMSCF/D
1	.00	90.6	450.0	70.0	990.0	194.8	10031.	1.926	0.	226.80
2	121.72	159.0	765.9	68.3	990.0	107.8	2884.	.554	0.	226.25
3	256.23	212.1	762.7	68.2	990.0	106.3	2928.	.562	0.	225.69
4	382.48	200.5	757.8	68.3	985.1	108.6	2939.	.564	0.	225.12
5	518.39	167.1	745.5	68.2	969.2	108.5	2930.	.563	0.	224.56
6	656.00	131.0	717.4	68.2	932.6	108.3	2921.	.561	0.	224.00
							24634.	4.730	0.	

Fig. 4.1.1.3-2(b) Typical output, gas gynamics submodel

presented in Table 4.1.1.3-1. The reference system was then converted from reciprocators to simple-cycle and combined-cycle turbines, as shown in Tables 4.1.1.3-2 and 4.1.1.3-3, respectively. The heat rates were then adjusted from the full-load values to those associated with the actual operating horsepowers required, using the part-load characteristics from Fig. 4.1.1.3-1.

For purposes of the comparison, the simple cycle is regarded as the baseline. The output from the economic model for this case is shown in Fig. 4.1.1.3-3. Figure 4.1.1.3-4 presents the output for the combined cycle. For the comparison, the total costs are taken to be equal for both cases. Thus, the question being asked of the model is, if these improved cycle engines could be installed and operated at the same cost as the simple cycle, what would be the benefit? With this information, it is possible to calculate the operator's incentive in energy and dollar savings to adopt the innovations. It is noted that although the total costs input for the two cases are the same, the capital infusion rates are different, reflecting the differences in timing for capacity additions. Some of the more important inputs and assumptions are listed in Table 4.1.1.3-4. Table 4.1.1.3-5 presents some significant figures, extracted from the two outputs.

Several points of interest are evident. First, from lines 1 through 4, it is seen that the energy saving actually realized for the assumed duty cycle (which is determined by the assumed market growth and the power increments necessary to meet it), though reduced somewhat from the full-load values, is still very large. Second, from the viewpoint of economy, the 20-year present value of energy saved would be $26.8 million for the combined cycle. In terms of prime mover capacity, this latter figure represents $514 per horsepower for the particular set of assumptions used, which is almost twice the

Table 4.1.1.3-1

REPRESENTATIVE GAS TURBINES
IN GAS PIPELINE COMPRESSOR SERVICE

		Simple Cycle		Regenerative Cycle		(Estimated)[2] Combined Cycle	
Manufacturer	Model No.	HP (ISO) Rating	SFC BTU/HP-hr	HP (ISO) Rating	SFC BTU/HP-hr	HP (ISO) Rating	SFC BTU/HP-hr
Solar	Saturn	1160	11600	1100	9510	1804	7284
Solar	Centaur	3830	9600	3580[1]	8100[1]	5600	6578
Solar	Centaur[3]	4850	9100	4540	7460	7090	6288
Solar	Mars[3]	10300	8080	9630	6625	13680	6080
Allison	501K5	2745	11070	2600	9080	4120	7370
Allison	501K13	3165	9980	3000	8180	4750	6650
Cooper-Bessemer (P&W)	RT25 (GG12)	2750	13800	2600	11315	4125	9190
Cooper-Bessemer (Rolls Royce)	COB125 (Avon)	12500	9800	11875	8035	18750	6525
DeLaval	Turbopac	3300	11600	3130	9510	4950	7725
Ingersoll Rand	GT40/22	4250	9430	4040	7730	6375	6280
General Electric	Frame 3 (Mod. 3000)	6850	11000	6850[1]	9000[1]	9800	7690
General Electric	M3872	8780	10850	8750	9000	13125	7225
General Electric	M3102	10800	10190	10800	9000	16200	6785
General Electric	M3132	13100	9760	13100	9000	19650	6500
Orenda	OT370	8090	11800	8200	9675	13325	7860
Orenda	OT-F-270	9830	11550	9070[1]	7770[1]	14745	7690

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[1]From mfr.'s published data; all other figures on regenerative cycle are estimates, based on assumptions that regenerative cycle hp is 5% lower and SFC is 18% lower than simple cycle.

[2]Combined cycle hp figures represent increase of 33% to 55% over simple cycle hp, depending on engine. Where published data are not available, 50% is assumed.

[3]Development models

Table 4.1.1.3-2

REFERENCE GAS SYSTEM CONVERSION
RECIPROCATORS TO SIMPLE-CYCLE TURBINES

Yr.	Engine	TIHP	Yr.	Engine	TIHP	Yr	Engine	TIHP	Rqmt.
1	Mars	10300	2	Mars	20600	→		20600	19415
8	Cent-D	4850	11	Cent	8680	14	Cent	12510	12075
1	Cent	3830	4	Cent-D	8680	14	Cent	12510	11822
8	Cent-D	4850	12	Cent	8680	14	Cent	12510	11766
1	Cent	3830	4	Cent-D	8680	14	Cent	12510	11755
8	Cent-D	4850	12	Cent	8680	14	Cent	12510	11731
1	Cent	3830	4	Cent-D	8680	14	Cent	12510	11708
8	Cent-D	4850	12	Cent	8680	14	Cent	12510	11687
1	Cent	3830	4	Cent-D	8680	14	Cent	12510	11645
8	Cent-D	4850	12	Cent	8680	14	Cent	12510	11587
1	Cent	3830	4	Cent-D	8680	14	Cent	12510	11518
10	Cent-D	4850	15	Saturn	6010	→		6010	5770
7	Cent	3830	16	Saturn	4990	16	Booster	5030	5028
								156,740	147,507

$$\frac{1567}{1475} = 1.06237$$

i.e., 6.24%
excess

Table 4.1.1.3-3

REFERENCE GAS SYSTEM CONVERSION
RECIPROCATORS TO COMBINED-CYCLE TURBINES

St.	Yr.	Engine	TIHP	Yr.	Engine	TIHP	RQMT.
1	1	Mars	13680	7	Cent-D	20770	19415
2	8	Cent	5600	12	Cent-D	12690	12075
3	1	Cent	5600	6	Cent-D	12690	11822
4	8	Cent	5600	12	Cent-D	12690	11766
5	1	Cent	5600	6	Cent-D	12690	11755
6	Same	as	St. 2	→	→	12690	11731
7			3	→	→	12690	11708
8			2	→	→	12690	11687
9			3	→	→	12690	11645
10			2	→	→	12690	11587
11			3			12690	11518
12	10	Cent-D	7090	No further installatn.		7090	
13	7	Cent	5600	"	"	5600	5028
						160,360	147,500

$$\frac{160,360}{147,507} = 1.0871,$$

i.e., 8.71% excess

SYSTEMS, SCIENCE AND SOFTWARE
GAS PROJECTION MODEL
PIPELINE TRANSPORTATION SYSTEMS ENERGY CONSERVATION STUDY

DATE OCTOBER 13, 1976 09:29:35 #5 100876
RUN ID THE NATURAL GAS REFERENCE SYSTEM
BASELINE CASE

PJB REPORT NO. 3H CAPITAL INVESTMENT PLANNING AND ENERGY CONSERVATION IMPACT PROJECTION (DOLLARS IN THOUSANDS)

TIME PERIOD	1976	1977	1978	1979	1980	1981	1982	1983
ACTIVITY								
ANNUAL THROUGHPUT (MMMMCF-MILES)	.000	57.562	60.936	64.512	67.654	71.477	75.155	78.9
NOMINAL TARIFF (UNIT TRANSP. CHARGE)	.000	660.000	693.000	727.650	764.032	802.234	842.346	884.4
ACTUAL TARIFF	.000	632.128	644.703	635.103	657.246	629.268	611.501	630.9
NOMINAL TRANSPORTATION REVENUES	.000	37991.023	42228.454	46942.505	51842.370	57390.993	63306.632	69786.9
ACTUAL TOTAL REVENUES	.000	36386.633	39285.425	40972.055	44596.515	44977.949	45757.490	49780.4
LEVERAGE								
LONG-TERM (FUNDED) DEBT TO CAPITAL &	61.454	61.454	61.454	57.304	55.615	54.210	53.354	56.1
LONG-TERM (FUNDED) DEBT TO ASSETS &	60.003	59.549	59.092	54.713	52.667	50.543	49.105	51.5
PROFITABILITY								
OPERATING INCOME (FPC RULES)	.000	20629.696	20342.559	20616.535	20845.830	20220.546	19779.415	20516.7
ANNUAL FPC RATE BASE	206345.000	206296.959	203339.029	205989.750	208405.182	202119.022	197666.761	205020.9
RATE OF RETURN ON RATE BASE (%)	.000	10.000	10.004	10.009	10.003	10.004	10.006	10.0
RATE OF RETURN ON PAID-IN CAPITAL(%)	.000	11.483	11.151	11.468	12.400	12.343	12.499	13.6
RATE OF RETURN ON TOTAL CAPITAL (%)	.000	4.705	4.568	4.639	5.191	5.354	5.524	5.6
ENERGY CONSUMPTION								
ANNUAL ENERGY USAGE OF GAS (MMCF)	.000	1559.083	2005.124	2263.467	2978.737	3307.456	3719.575	4400.
ANNUAL ENERGY COSTS	.000	2213.897	2989.640	3543.570	4896.529	5700.731	6741.059	8373.
PRESENT VALUE OF ENERGY USED	.000	2012.634	2470.777	2662.337	3344.396	3544.673	3805.152	4296.
UNIT COST OF ENERGY $/MMCF	.000	1.420	1.491	1.566	1.644	1.726	1.812	1.
OTHER MEASURES								
TOTAL ANNUAL UNIT COSTS	.000	431.286	428.012	422.044	427.943	414.534	406.700	418.
PRESENT VALUE OF AVERAGE UNIT COSTS	.000	392.078	353.729	317.088	292.291	257.393	229.572	214.
NET INCOME (BOOK PROFIT)	-5113.000	9924.909	9637.772	9911.749	10717.246	10668.165	10003.237	11835.
PRESENT VALUE OF BOOK PROFITS	-5113.000	9022.645	7965.101	7446.843	7320.024	6624.091	6098.146	6073.
NET CASH GENERATED DURING THE PERIOD	.000	11990.008	17475.867	-6393.690	12358.145	14239.059	12138.490	10059.
PRESENT VALUE OF NET CASH GENERATED	.000	10900.080	14442.065	-4803.674	8440.779	7599.493	6851.862	5572.
DISCOUNT FACTOR (@10.000 %) =	1.003	.909	.826	.751	.683	.621	.564	.

Fig. 4.1.1.3-3(a)

SYSTEMS, SCIENCE AND SOFTWARE
LAC PROJECTION MODEL
PIPELINE TRANSPORTATION SYSTEMS ENERGY CONSERVATION STUDY

DATE OCTOBER 13, 1976 09:29:35 #5 100876
RUN ID THE NATURAL GAS REFERENCE SYSTEM
BASELINE CASE

P38 REPORT NO. 34 CAPITAL INVESTMENT PLANNING AND ENERGY CONSERVATION IMPACT PROJECTION (DOLLARS IN THOUSANDS)

TIME PERIOD	1984	1985	1986	1987	1988	1989	1990	1991
ACTIVITY								
ANNUAL THROUGHPUT (MMMMCF-MILES)	81.836	85.513	89.088	92.843	96.288	100.124	103.597	106.9
NOMINAL TARIFF (UNIT TRANSP. CHARGE)	928.686	975.121	1023.877	1075.070	1128.824	1185.265	1244.528	1306.7
ACTUAL TARIFF	637.807	639.150	656.957	653.625	710.195	701.613	791.995	828.8
NOMINAL TRANSPORTATION REVENUES	75999.933	83385.776	91214.742	99812.914	108691.813	118674.064	128929.953	139709.2
ACTUAL TOTAL REVENUES	52195.564	54655.833	58526.721	60703.142	68383.046	70248.600	82048.696	88620.2
LEVERAGE								
LONG-TERM (FUNDED) DEBT TO CAPITAL %	54.626	59.229	57.197	52.810	51.496	59.234	68.329	56.2
LONG-TERM (FUNDED) DEBT TO ASSETS %	49.389	53.502	50.921	46.547	44.678	51.987	50.323	47.6
PROFITABILITY								
OPERATING INCOME (FPC RULES)	21023.334	20771.450	20692.507	20915.106	20933.534	22519.146	24225.127	23656.8
ANNUAL FPC RATE BASE	210150.963	207645.766	206765.299	208959.906	209138.326	225079.184	242022.133	236443.9
RATE OF RETURN ON RATE BASE (%)	10.004	10.003	10.008	10.009	10.009	10.005	10.009	10.0
RATE OF RETURN ON PAID-IN CAPITAL(%)	12.997	13.373	11.157	12.208	13.022	15.412	13.605	13.4
RATE OF RETURN ON TOTAL CAPITAL (%)	5.528	5.111	4.399	4.770	5.230	5.202	4.694	4.8
ENERGY CONSUMPTION								
ANNUAL ENERGY USAGE OF GAS (MMCF)	4466.495	4920.097	5294.936	6183.640	6942.746	7740.591	9563.023	10614.3
ANNUAL ENERGY COSTS	8924.424	10322.299	11664.140	14302.946	16861.721	19739.403	25606.156	29642.3
PRESENT VALUE OF ENERGY USED	4163.310	4377.662	4497.031	5013.096	5372.664	5717.802	6742.902	7144.0
UNIT COST OF ENERGY $/MMCF	1.998	2.098	2.203	2.313	2.429	2.550	2.678	2.8
OTHER MEASURES								
TOTAL ANNUAL UNIT COSTS	437.672	445.919	472.581	481.661	496.255	527.446	619.839	644.4
PRESENT VALUE OF AVERAGE UNIT COSTS	204.177	189.113	182.200	168.826	158.122	152.782	163.223	154.2
NET INCOME (BOOK PROFIT)	11233.691	11557.959	9643.312	10551.332	11255.180	13320.435	11758.539	11634.9
PRESENT VALUE OF BOOK PROFITS	5240.577	4901.703	3717.914	3698.178	3586.247	3858.456	3096.391	2785.3
NET CASH GENERATED DURING THE PERIOD	14240.634	35462.175	8017.922	-3125.719	16499.091	17640.694	16520.712	10690.5
PRESENT VALUE OF NET CASH GENERATED	6643.361	15039.424	3091.256	-1095.545	5257.119	5109.681	4350.420	2559.2
DISCOUNT FACTOR (@10.000 %) =	.467	.424	.386	.350	.319	.290	.263	.2

Fig. 4.1.1.3-3(b)

SYSTEMS, SCIENCE AND SOFTWARE
LAC PROJECTION MODEL
PIPELINE TRANSPORTATION SYSTEMS ENERGY CONSERVATION STUDY

DATE OCTOBER 13, 1976 09:29:35 #5 100876
RUN ID THE NATURAL GAS REFERENCE SYSTEM
BASELINE CASE

P3B REPORT NO. 3B CAPITAL INVESTMENT PLANNING AND ENERGY CONSERVATION IMPACT PROJECTION (DOLLARS IN THOUSANDS)

TIME PERIOD	1992	1993	1994	1995	1996	TOTAL	AVERAGE
ACTIVITY							
ANNUAL THROUGHPUT (MMMMCF-MILES)	106.910	106.910	106.910	106.910	106.910	1767.152	88.358
NOMINAL TARIFF (UNIT TRANSP. CHARGE)	1372.092	1440.697	1512.732	1588.368	1667.787	21823.527	1091.176
ACTUAL TARIFF	854.543	870.033	886.897	905.715	924.882	14132.206	706.610
NOMINAL TRANSPORTATION REVENUES	146690.373	154024.891	161726.135	169812.439	178303.061	2026414.203	101320.710
ACTUAL TOTAL REVENUES	91359.193	93015.170	94818.127	96829.943	98879.102	1312239.691	65611.994
LEVERAGE							
LONG-TERM (FUNDED) DEBT TO CAPITAL %	53.726	50.661	47.130	43.053	38.359	54.744	54.744
LONG-TERM (FUNDED) DEBT TO ASSETS %	44.464	40.966	37.203	33.130	28.727	48.160	48.160
PROFITABILITY							
OPERATING INCOME (FPC RULES)	22923.784	21989.547	21055.362	20121.222	19177.466	422955.727	21147.786
ANNUAL FPC RATE BASE	229102.689	219746.439	210390.189	201033.943	191677.693	4433334.500	221666.725
RATE OF RETURN ON RATE BASE (%)	10.006	10.007	10.008	10.009	10.005	9.540	9.540
RATE OF RETURN ON PAID-IN CAPITAL(%)	13.584	13.737	13.914	14.205	14.371	12.103	12.103
RATE OF RETURN ON TOTAL CAPITAL (%)	5.158	5.534	5.959	6.484	7.014	4.080	4.080
ENERGY CONSUMPTION							
ANNUAL ENERGY USAGE OF GAS (MMCF)	10660.011	10660.011	10660.011	10660.011	10660.011	129259.623	6462.981
ANNUAL ENERGY COSTS	31969.182	33092.641	34694.773	36429.511	38250.986	345617.285	17280.864
PRESENT VALUE OF ENERGY USED	6848.611	6537.311	6240.160	5956.517	5685.766	96433.670	.000
DISCOUNTED VALUE OF ENERGY USED (@ 10.00 %) =	96433.670						
UNIT COST OF ENERGY $/MMCF	2.952	3.100	3.255	3.417	3.588	46.954	2.348
OTHER MEASURES							
TOTAL ANNUAL UNIT COSTS	660.809	673.850	687.852	701.970	718.454	10518.242	525.912
PRESENT VALUE OF AVERAGE UNIT COSTS	143.811	133.318	123.716	114.778	106.794	4052.262	202.613
DISCOUNTED AVERAGE (ANNUAL) UNIT COSTS (LONG-RUN AVERAGE COSTS) (@ 10.00 %) =	202.613						
NET INCOME (BOOK PROFIT)	11741.006	11873.175	12026.184	12277.404	12420.843	219679.820	10960.999
PRESENT VALUE OF BOOK PROFITS	2555.203	2349.095	2163.015	2007.454	1846.279	87243.206	4362.160
DISCOUNTED VALUE OF BOOK PROFITS (@ 10.00 %) =	87243.206						
NET CASH GENERATED DURING THE PERIOD	13779.900	11333.811	11198.299	11143.376	11087.507	243156.699	12157.835
PRESENT VALUE OF NET CASH GENERATED	2563.669	2242.334	2014.103	1822.031	1648.007	100249.553	5012.478
DISCOUNTED NET CASH FLOW (@ 10.00 %) =	100249.553						
DISCOUNT FACTOR (@10.000 %) =	.210	.198	.180	.164	.149	.000	.000

****** INTERNAL RATE OF RETURN ******

DCF - ROI OF $ 86430.430 (FROM YEAR 1 OVER 10 YEARS) = 6.89 %
DCF - ROI OF $ 86430.400 (FROM YEAR 1 OVER 15 YEARS) = 10.65 %
DCF - ROI OF $ 86430.430 (FROM YEAR 1 OVER 20 YEARS) = 12.14 %

Fig. 4.1.1.3-3(c)

TIME PERIOD	1976	1977	1978	1979	1980	1981	1982	1983
OTHER LINE ITEMS								
OPERATION AND MAINTENANCE EXPENSES	.000	6242.000	6567.000	7159.000	7722.000	8370.000	8558.000	9606.0
INTEREST EXPENSES	.000	10371.648	10371.648	10371.648	9775.445	9219.243	8643.040	8347.8
TOTAL EXPENSES	.000	24825.767	26081.204	27227.135	29037.474	25629.473	30565.598	33053.0
UNUSED TAX LOSS	.000	.000	.000	.000	.000	.000	.000	.0
UNUSED INVESTMENT TAX CREDITS	3839.113	.000	.000	.000	.000	.000	.000	.0
LONG-TERM BORROWING	129645.600	.000	.000	.000	.000	.000	3513.000	21061.0
NET ADDITIONS TO EQUITY	86430.400	.000	.000	.000	.000	.000	.000	.0
ADDITIONS TO PLANT & EQUIPMENT	203943.000	5569.000	.000	16941.000	.000	.000	3665.000	23705.0
LONG-TERM DEBT RETIREMENT	.000	.000	.000	7202.533	7202.533	7202.533	7202.533	7202.5
PLANT & EQUIPMENT (@ ORIGINAL COST)	203943.000	209512.000	209512.000	226453.000	226453.000	226453.000	230138.000	253843.0
NET PROPERTY & EQUIPMENT	203943.000	203846.918	198027.141	209148.361	202858.002	196567.641	193962.279	211274.59
TOTAL DEBT BALANCE	129645.600	129645.600	129645.600	122443.066	115290.533	108030.000	104348.467	118206.9
TOTAL EQUITY CAPITAL	81317.400	81317.400	81317.400	91229.148	91229.148	91229.148	91229.148	92205.0

TIME PERIOD	1984	1985	1986	1987	1988	1989	1990	1991
OTHER LINE ITEMS								
OPERATION AND MAINTENANCE EXPENSES	10052.000	11545.000	12084.000	12653.000	13409.000	16038.000	16986.000	17793.0
INTEREST EXPENSES	9456.555	8880.352	10716.056	10030.635	9345.215	8865.572	12133.450	11688.7
TOTAL EXPENSES	35817.312	38131.983	42101.083	44720.748	47783.213	52810.252	64213.716	68904.5
UNUSED TAX LOSS	.000	.000	.000	.000	.000	.000	.000	.0
UNUSED INVESTMENT TAX CREDITS	.000	.000	.000	.000	.000	.000	.000	.0
LONG-TERM BORROWING	.000	30344.000	.000	.000	4258.000	51102.000	4695.000	.0
NET ADDITIONS TO EQUITY	.000	.000	.000	.000	.000	.000	.000	.0
ADDITIONS TO PLANT & EQUIPMENT	.000	9092.000	3502.000	15592.000	.000	47550.000	3325.000	3921.0
LONG-TERM DEBT RETIREMENT	7202.533	7397.700	8567.755	8567.755	10253.533	10253.533	10253.533	10490.0
PLANT & EQUIPMENT (@ ORIGINAL COST)	253843.000	262935.000	266437.000	282029.000	232029.000	329579.000	332904.000	336825.0
NET PROPERTY & EQUIPMENT	234223.363	206264.172	202462.422	210653.395	202819.254	242535.117	236705.145	231378.6
TOTAL DEBT BALANCE	111004.400	133950.699	125382.944	116815.189	110819.655	151668.123	146109.590	135619.6
TOTAL EQUITY CAPITAL	92205.069	92205.069	93830.460	104381.792	104381.792	104381.792	104381.792	105326.2

TIME PERIOD	1992	1993	1994	1995	1996	TOTAL	AVERAGE
OTHER LINE ITEMS							
OPERATION AND MAINTENANCE EXPENSES	18639.000	19526.000	20458.000	21418.000	22446.000	266979.000	13348.950
INTEREST EXPENSES	10849.560	9783.233	8696.039	7608.846	6521.652	191696.469	9584.823
TOTAL EXPENSES	70647.130	72041.262	73538.200	75047.577	76809.858	962986.609	48149.330
UNUSED TAX LOSS	.000	.000	.000	.000	.000	.000	.000
UNUSED INVESTMENT TAX CREDITS	.000	.000	.000	.000	.000	3839.113	191.956
LONG-TERM BORROWING	.000	.000	.000	.000	.000	244618.600	11648.505
NET ADDITIONS TO EQUITY	.000	.000	.000	.000	.000	86430.400	4115.733
ADDITIONS TO PLANT & EQUIPMENT	.000	.000	.000	.000	.000	336825.000	16039.286
LONG-TERM DEBT RETIREMENT	13329.089	13589.922	13589.922	13589.922	13589.922	176687.869	8413.708
PLANT & EQUIPMENT (@ ORIGINAL COST)	336825.000	336825.000	336825.000	336825.000	336825.000	5817013.000	277000.617
NET PROPERTY & EQUIPMENT	222022.562	212666.312	203310.066	193953.816	184597.570	4373219.750	208248.559
TOTAL DEBT BALANCE	122290.413	108700.491	95110.569	81520.648	67930.726	2464136.594	117339.837
TOTAL EQUITY CAPITAL	135326.222	105865.586	106693.526	107827.554	109180.890	2037341.516	97001.977

****** AND THAT'S THE WAY IT WILL BE .... THIS IS THE S-CUBED FINANCIAL PROJECTION MODEL *JFM* VERSION #5 100876

Fig. 4.1.1.3-3(d)

SYSTEMS, SCIENCE AND SOFTWARE
LAC PROJECTION MODEL
PIPELINE TRANSPORTATION SYSTEMS   ENERGY CONSERVATION STUDY

DATE   NOVEMBER 1, 1976   11136145   #6   101476
RUN ID   GAS REF SYSTEM CONVERSION TO COMBINED CYCLE
CASE P38.1   LINKED WITH PEPGAS •GAS2

P38 REPORT NO. 38   CAPITAL INVESTMENT PLANNING AND ENERGY CONSERVATION IMPACT PROJECTION (DOLLARS IN THOUSANDS)

TIME PERIOD	1976	1977	1978	1979	1980	1981	1982	1983
ACTIVITY								
ANNUAL THROUGHPUT (MMMMCF-MILES)	.000	57.768	61.326	64.945	68.556	72.228	75.742	79.47
NOMINAL TARIFF (UNIT TRANSP. CHARGE)	.000	660.000	693.000	727.650	764.032	802.234	842.346	884.46
ACTUAL TARIFF	.000	632.265	621.210	592.348	566.912	566.747	599.076	622.1
NOMINAL TRANSPORTATION REVENUES	.000	38120.313	42499.196	47256.963	52379.136	57943.542	63801.068	70293.40
ACTUAL TOTAL REVENUES	.000	36518.374	38096.598	38469.797	38865.287	40934.846	44996.602	49445.64
LEVERAGE								
LONG-TERM (FUNDED) DEBT TO CAPITAL %	61.414	61.414	61.414	60.051	58.588	62.154	62.839	65.71
LONG-TERM (FUNDED) DEBT TO ASSETS %	60.000	59.647	59.101	57.269	55.339	58.565	58.799	61.25
PROFITABILITY								
OPERATING INCOME (FPC RULES)	.000	20945.772	20375.466	19786.703	19205.913	19948.070	21226.235	22444.59
ANNUAL FPC RATE BASE	212374.000	209457.723	203625.168	197792.613	191960.057	199356.000	212124.488	224293.07
RATE OF RETURN ON RATE BASE (%)	.000	10.000	10.006	10.003	10.005	10.006	10.006	10.00
RATE OF RETURN ON PAID-IN CAPITAL(%)	.000	11.201	10.560	9.896	9.910	11.412	11.135	12.13
RATE OF RETURN ON TOTAL CAPITAL (%)	.000	4.686	4.323	4.195	4.354	4.583	4.390	4.41
ENERGY CONSUMPTION								
ANNUAL ENERGY USAGE OF GAS (MMCF)	.000	1273.805	1471.607	1647.000	1843.321	2066.802	2808.139	3415.38
ANNUAL ENERGY COSTS	.000	1808.803	2194.165	2578.461	3013.664	3567.340	5089.247	6499.25
PRESENT VALUE OF ENERGY USED	.000	1644.366	1813.360	1937.236	2058.373	2215.037	2872.747	3335.14
UNIT COST OF ENERGY $/MMCF	.000	1.420	1.491	1.566	1.644	1.726	1.812	1.90
OTHER MEASURES								
TOTAL ANNUAL UNIT COSTS	.000	430.720	417.241	409.028	393.402	377.798	416.527	436.0
PRESENT VALUE OF AVERAGE UNIT COSTS	.000	391.563	344.827	307.309	268.699	234.583	235.118	223.7
NET INCOME (BOOK PROFIT)	-5113.000	9951.593	9381.287	8791.525	8804.014	10138.451	9892.336	10784.57
PRESENT VALUE OF BOOK PROFITS	-5113.000	9046.903	7753.130	6605.203	6013.260	6295.180	5583.966	5534.1
NET CASH GENERATED DURING THE PERIOD	.000	17806.531	17236.225	9242.962	9255.452	10589.889	11312.528	12886.0
PRESENT VALUE OF NET CASH GENERATED	.000	16187.755	14244.814	6944.374	6321.598	6575.488	6365.627	6612.6
DISCOUNT FACTOR (@10.000 %) =	1.000	.909	.826	.751	.683	.621	.564	.5

Fig. 4.1.1.3-4(a)

SYSTEMS, SCIENCE AND SOFTWARE
LAC PROJECTION MODEL
PIPELINE TRANSPORTATION SYSTEMS ENERGY CONSERVATION STUDY

DATE NOVEMBER 1, 1976 11136145 #6 101476
RUN ID GAS REF SYSTEM CONVERSION TO COMBINED CYCLE
CASE P38.1 LINKED WITH PEPGAS •GASZ

P38 REPORT NO. 38 CAPITAL INVESTMENT PLANNING AND ENERGY CONSERVATION IMPACT PROJECTION (DOLLARS IN THOUSANDS)

TIME PERIOD	1984	1985	1986	1987	1988	1989	1990	1991
ACTIVITY								
ANNUAL THROUGHPUT (MMMMCF-MILES)	82.646	86.152	89.768	93.706	97.137	101.105	105.183	108.[illegible]
NOMINAL TARIFF (UNIT TRANSP. CHARGE)	928.686	975.121	1023.877	1075.070	1128.824	1185.265	1244.528	1306.[illegible]
ACTUAL TARIFF	616.923	627.790	621.390	640.794	691.340	734.920	730.448	734.[illegible]
NOMINAL TRANSPORTATION REVENUES	76751.893	84008.382	91911.129	100740.598	109650.489	119836.758	130903.530	141999.[illegible]
ACTUAL TOTAL REVENUES	50903.378	54085.259	55780.803	60046.295	67154.612	74304.950	76830.920	79790.[illegible]
LEVERAGE								
LONG-TERM (FUNDED) DEBT TO CAPITAL %	64.423	64.669	62.963	67.568	66.043	64.368	61.839	58.[illegible]
LONG-TERM (FUNDED) DEBT TO ASSETS %	59.184	58.777	56.261	60.566	58.038	55.438	52.117	48.[illegible]
PROFITABILITY								
OPERATING INCOME (FPC RULES)	23026.096	22824.781	22610.531	23964.975	25283.470	24372.477	23450.953	22540.[illegible]
ANNUAL FPC RATE BASE	230196.781	228151.561	225951.561	239588.783	252627.006	243502.229	234377.451	225252.[illegible]
RATE OF RETURN ON RATE BASE (%)	10.003	10.004	10.007	10.003	10.008	10.009	10.006	10.[illegible]
RATE OF RETURN ON PAID-IN CAPITAL(%)	11.091	11.663	11.275	13.788	12.376	12.395	12.401	12.[illegible]
RATE OF RETURN ON TOTAL CAPITAL (%)	4.187	4.372	4.432	4.745	4.459	4.686	4.964	5.[illegible]
ENERGY CONSUMPTION								
ANNUAL ENERGY USAGE OF GAS (MMCF)	3069.445	3843.627	4166.690	4719.399	5459.426	6024.902	6707.810	7403.[illegible]
ANNUAL ENERGY COSTS	6113.024	8063.669	9178.743	10916.113	13259.208	15364.196	17960.975	20815.[illegible]
PRESENT VALUE OF ENERGY USED	2851.771	3419.783	3538.803	3826.031	4224.793	4450.461	4729.686	4963.[illegible]
UNIT COST OF ENERGY $/MMCF	1.998	2.098	2.203	2.313	2.429	2.550	2.678	2.[illegible]
OTHER MEASURES								
TOTAL ANNUAL UNIT COSTS	447.148	460.683	465.456	461.138	515.571	532.984	537.205	543.[illegible]
PRESENT VALUE OF AVERAGE UNIT COSTS	208.598	195.374	179.453	161.626	164.277	154.386	141.463	130.[illegible]
NET INCOME (BOOK PROFIT)	9853.478	10362.029	10017.148	12249.419	10995.501	11011.863	11017.694	11226.[illegible]
PRESENT VALUE OF BOOK PROFITS	4596.720	4394.512	3862.044	4293.347	3503.506	3189.745	2901.303	2687.[illegible]
NET CASH GENERATED DURING THE PERIOD	11999.045	11717.970	10434.646	12495.372	13025.758	12722.133	10041.067	9985.[illegible]
PRESENT VALUE OF NET CASH GENERATED	5597.643	4969.564	4023.008	4379.552	4150.408	3685.149	2644.127	2390.[illegible]
DISCOUNT FACTOR (@10.000 %) =	.467	.424	.386	.350	.319	.290	.263	[illegible]

Fig. 4.1.1.3-4(b)

SYSTEMS, SCIENCE AND SOFTWARE
LAC PROJECTION MODEL
PIPELINE TRANSPORTATION SYSTEMS ENERGY CONSERVATION STUDY

DATE NOVEMBER 1, 1976 11136145 #6 101476
RUN ID GAS REF SYSTEM CONVERSION TO COMBINED CYCLE
CASE P38.1 LINKED WITH PEPGAS *GAS2

P38 REPORT NO. 38 CAPITAL INVESTMENT PLANNING AND ENERGY CONSERVATION IMPACT PROJECTION (DOLLARS IN THOUSANDS)

TIME PERIOD	1992	1993	1994	1995	1996	TOTAL	AVERAGE
ACTIVITY							
ANNUAL THROUGHPUT (MMMMCF-MILES)	108.666	108.666	108.666	108.666	108.666	1787.723	89.386
NOMINAL TARIFF (UNIT TRANSP. CHARGE)	1372.092	1440.697	1512.732	1588.368	1667.787	21823.527	1091.176
ACTUAL TARIFF	745.165	756.916	769.585	783.747	798.368	13028.370	651.418
NOMINAL TRANSPORTATION REVENUES	149099.611	156554.592	164382.318	172601.434	181231.504	2051965.406	102598.270
ACTUAL TOTAL REVENUES	80973.998	82250.896	83627.570	85166.531	86755.383	1224997.453	61249.873
LEVERAGE							
LONG-TERM (FUNDED) DEBT TO CAPITAL %	55.556	51.673	47.181	41.956	35.869	59.894	59.894
LONG-TERM (FUNDED) DEBT TO ASSETS %	44.706	40.553	36.052	31.142	25.788	52.683	52.683
PROFITABILITY							
OPERATING INCOME (FPC RULES)	21629.431	20718.462	19807.560	18886.033	17974.414	431021.375	21551.069
ANNUAL FPC RATE BASE	216127.896	207003.119	197878.342	188753.564	179628.787	4520022.625	226001.131
RATE OF RETURN ON RATE BASE (%)	10.008	10.009	10.010	10.006	10.006	9.536	9.536
RATE OF RETURN ON PAID-IN CAPITAL(%)	12.870	13.105	13.339	13.672	13.905	11.191	11.191
RATE OF RETURN ON TOTAL CAPITAL (%)	5.813	6.313	6.876	7.569	8.308	4.655	4.655
ENERGY CONSUMPTION							
ANNUAL ENERGY USAGE OF GAS (MMCF)	7403.608	7403.608	7403.608	7403.608	7403.608	92918.903	4645.945
ANNUAL ENERGY COSTS	21856.027	22948.828	24096.270	25301.083	26566.137	247190.469	12359.523
PRESENT VALUE OF ENERGY USED	4756.509	4540.304	4333.926	4136.930	3948.887	69617.154	.000
DISCOUNTED VALUE OF ENERGY USED (@ 10.00 %) =		69617.154					
UNIT COST OF ENERGY $/MMCF	2.952	3.100	3.255	3.417	3.588	46.954	2.348
OTHER MEASURES							
TOTAL ANNUAL UNIT COSTS	550.449	558.370	567.208	575.929	586.732	9682.974	484.149
PRESENT VALUE OF AVERAGE UNIT COSTS	119.794	110.471	102.017	94.169	87.214	3854.762	192.738
DISCOUNTED AVERAGE (ANNUAL) UNIT COSTS (LONG-RUN AVERAGE COSTS) (@ 10.00 %) =		192.738					
NET INCOME (BOOK PROFIT)	11434.243	11642.310	11850.443	12146.118	12353.535	208790.789	9942.418
PRESENT VALUE OF BOOK PROFITS	2488.425	2303.369	2131.407	1985.988	1836.274	81892.937	4094.647
DISCOUNTED VALUE OF BOOK PROFITS (@ 10.00 %) =		81892.937					
NET CASH GENERATED DURING THE PERIOD	9952.753	9942.264	9951.711	10034.305	10130.740	230762.527	10988.692
PRESENT VALUE OF NET CASH GENERATED	2166.009	1967.024	1789.903	1640.689	1505.870	104181.533	5209.077
DISCOUNTED NET CASH FLOW (@ 10.00 %) =		104181.533					
DISCOUNT FACTOR (@10.000 %) =	.218	.198	.180	.164	.149	.000	.000

****** INTERNAL RATE OF RETURN ******

DCF - ROI OF $ 88841.999 (FROM YEAR 1 OVER 10 YEARS) = 6.80 %
DCF - ROI OF $ 88841.999 (FROM YEAR 1 OVER 15 YEARS) = 11.31 %
DCF - ROI OF $ 88841.999 (FROM YEAR 1 OVER 20 YEARS) = 12.70 %

Fig. 4.1.1.3-4(c)

TIME PERIOD	1976	1977	1978	1979	1980	1981	1982	1983
OTHER LINE ITEMS								
OPERATION AND MAINTENANCE EXPENSES	.000	6242.000	6567.000	7159.000	7722.000	8078.000	8558.000	9606.0
INTEREST EXPENSES	.000	10661.040	10661.040	10661.040	10068.760	9476.480	11000.760	11326.8
TOTAL EXPENSES	.000	24877.537	25587.899	26564.195	26970.118	27287.519	31548.617	34651.6
UNUSED TAX LOSS	.000	.000	.000	.000	.000	.000	.000	.0
UNUSED INVESTMENT TAX CREDITS	6042.858	.000	.000	.000	.000	.000	.000	.0
LONG-TERM BORROWING	133263.002	.000	.000	.000	.000	26457.000	11480.000	26311.0
NET ADDITIONS TO EQUITY	68841.999	.000	.000	.000	.000	.000	.000	.0
ADDITIONS TO PLANT & EQUIPMENT	209972.000	.000	.000	.000	.000	26457.000	11480.000	26311.0
LONG-TERM DEBT RETIREMENT	.000	.000	.000	7403.500	7403.500	7403.500	7403.500	7403.5
PLANT & EQUIPMENT (@ ORIGINAL COST)	209972.000	209972.000	209972.000	209972.000	209972.000	236429.000	247909.000	274220.0
NET PROPERTY & EQUIPMENT	209972.000	204139.445	198306.891	192474.334	186641.779	207266.225	212178.752	231603.3
TOTAL DEBT BALANCE	133263.002	133263.002	133263.002	125859.502	118456.002	137509.502	141566.002	160493.5
TOTAL EQUITY CAPITAL	83728.999	83728.999	83728.999	83728.999	83728.999	83728.999	83728.999	83728.9

TIME PERIOD	1984	1985	1986	1987	1988	1989	1990	1991
OTHER LINE ITEMS								
OPERATION AND MAINTENANCE EXPENSES	10052.000	11545.000	12084.000	12653.000	13909.000	16038.000	16986.000	17793.0
INTEREST EXPENSES	12839.480	12129.613	12260.244	11382.418	13954.831	13027.476	12100.120	10981.0
TOTAL EXPENSES	36954.864	39688.643	41782.903	43211.446	50080.956	53887.588	56505.011	59047.2
UNUSED TAX LOSS	.000	.000	.000	.000	.000	.000	.000	.0
UNUSED INVESTMENT TAX CREDITS	.000	.000	.000	.000	.000	.000	.000	.0
LONG-TERM BORROWING	.000	11144.000	.000	43128.000	.000	.000	.000	.0
NET ADDITIONS TO EQUITY	.000	.000	.000	.000	.000	.000	.000	.0
ADDITIONS TO PLANT & EQUIPMENT	.000	11144.000	.000	43128.000	.000	.000	.000	.0
LONG-TERM DEBT RETIREMENT	8873.333	9511.111	10972.833	10972.833	11591.944	11591.944	13987.944	13987.9
PLANT & EQUIPMENT (@ ORIGINAL COST)	274220.000	285364.000	285364.000	328492.000	328492.000	328492.000	328492.000	328492.0
NET PROPERTY & EQUIPMENT	223986.172	227512.949	219586.172	254787.395	245662.617	236537.840	227413.062	218288.2
TOTAL DEBT BALANCE	151620.168	153253.057	142280.223	174435.391	162843.445	151251.502	137263.557	123275.6
TOTAL EQUITY CAPITAL	83728.999	83728.999	83728.999	83728.999	83728.999	83728.999	84705.627	85946.7

TIME PERIOD	1992	1993	1994	1995	1996	TOTAL	AVERAGE
OTHER LINE ITEMS							
OPERATION AND MAINTENANCE EXPENSES	18639.000	19526.000	20458.000	21418.000	22446.000	266979.000	13348.950
INTEREST EXPENSES	9862.049	8743.013	7623.978	6504.943	5385.907	210651.146	10532.557
TOTAL EXPENSES	59814.992	60675.757	61636.163	62583.774	63757.793	887114.625	44355.731
UNUSED TAX LOSS	.000	.000	.000	.000	.000	.000	.000
UNUSED INVESTMENT TAX CREDITS	.000	.000	.000	.000	.000	6042.858	302.143
LONG-TERM BORROWING	.000	.000	.000	.000	.000	251783.002	11989.667
NET ADDITIONS TO EQUITY	.000	.000	.000	.000	.000	88841.999	4230.571
ADDITIONS TO PLANT & EQUIPMENT	.000	.000	.000	.000	.000	328492.000	15642.476
LONG-TERM DEBT RETIREMENT	13987.944	13987.944	13987.944	13987.944	13987.944	198447.104	9449.862
PLANT & EQUIPMENT (@ ORIGINAL COST)	328492.000	328492.000	328492.000	328492.000	328492.000	5936286.000	282775.523
NET PROPERTY & EQUIPMENT	209163.508	200038.730	190913.953	181789.176	172664.398	4450926.875	211948.698
TOTAL DEBT BALANCE	109287.669	96299.726	81311.781	67323.837	53335.893	2686475.219	127927.391
TOTAL EQUITY CAPITAL	87428.196	89128.241	91026.974	93138.787	95361.582	1798941.984	85663.903

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Fig. 4.1.1.3-4(d)

Table 4.1.1.3-4

INPUTS AND ASSUMPTIONS

1. Full-load efficiency advantage of the combined cycle over the simple cycle is one-third.
2. Full-load efficiency advantage of the regenerative cycle over the simple cycle is 18%.
3. Present values calculated at 10%.
4. All profits paid in dividends.
5. All excess working capital reinvested at 6%.
6. Straight-line depreciation on first three capital outlays.
7. Market growth as shown in line 1 of the figures.
8. Base year 1975, inflated as shown in Fig. 4.1.1.3-3

Table 4.1.1.3-5

COMPARISON OF ENGINE CYCLES

	Simple	Comb'd.
Energy use, avg., BSCF/yr	6.46	4.65
Efficiency improvement, avg., %	0	28.02
Energy cost, avg., M $/yr	17.28	12.36
Present value of energy used, M$(20 years)	96.43	69.62
Actual tariff, avg., K$/GSCF-Mi	707	651
Tariff reduction, %	0	7.92
Present value of book profit, M$	87.24	81.89
10-yr RoI-DCF, %	6.89	6.80
20-yr " " "	12.14	12.70
Unused investment tax credits, M$	3.84	6.04

average 1975 cost of gas turbines in pipeline service. The above dollars per horsepower figure is based on a total horsepower requirement of 147,507 (see Table 4.1.1.3-2) and is derived as follows:

$$\frac{\$(96.43-69.62)10^6}{\$(96.43)10^6} \times 147{,}507 \text{ hp} = 41{,}011 \text{ hp}$$

attributable to bottoming cycle

$$\frac{\$(96.43 - 69.62)10^6}{41.011 \text{ hp}} = \$654/\text{hp}.$$

Bottoming engines, because of their requirements for boilers and condensers, are almost certain to be more expensive than open-cycle gas turbines, but not by a factor of two. It is therefore concluded that the bottoming engine offers to the ERDA a highly attractive opportunity for pipeline energy conservation.

Third, from the point of view of the consumer, the tariff reduction is approximately 8% with the improved engines. And fourth, from the point of view of the pipeline operator, lines 7-10 offer little inducement to invest in energy-conservative devices. Book profits actually fall slightly. For practical purposes, RoI is the same for all three cases, the small variations being primarily due to differences in timing of the capital infusions. Unused investment credits increase strongly for the energy-conservative cases, as explained in another report of this series, R-3024, Section 7.0 (see pp. 1 and 2 of this report). Basically, they derive from the limit upon profit imposed by FPC regulation. Clearly, some change in that regulation is needed to induce pipeline operators to conserve energy.

One regulatory change that seems reasonable would in effect divide the advantage between the consumer and the pipeline operator. This could be accomplished by allowing the operator to retain, in addition to the standard regulated return, a portion, e.g., one half, of the additional profit generated by the energy-conservative innovation. For purposes of policy promulgation, the allowance for additional profit should be related to the quantity of energy saved. However, other regulatory changes appear more effective (see R.3024 of this series).

4.1.1.4 Combined System Performance

Both the organic fluid Rankine cycle and the steam Rankine cycle were seen to offer high potential for waste heat recovery. Projected efficiencies for the gas turbine combined cycle are shown in Fig. 4.1.1.4-1.

For a typical first generation gas turbine with a thermodynamic efficiency of approximately 20% (heat rate of 12725), the addition of an organic Rankine cycle can increase the over-all system efficiency to 31%, while a steam bottoming system with 150°F condensing temperature will increase the over-all combined efficiency to 27%. (These figures are calculated for different site conditions and condensing temperatures than in the examples in Section 4.1.1.3. Thus, lower efficiencies are calculated in this case.) However, this turbine can also be improved by adding a recuperator. In fact, efficiency improvement with a recuperator is equal to the improvement with an organic Rankine cycle and better than a steam Rankine cycle. Because of this it is unlikely that the bottoming engine would be cost-effective. This conclusion bears repeating for emphasis. The bottoming engine, inherently, does not appear to be a viable candidate for retrofit on the first-generation turbine.

For a second generation gas turbine, representative of those installed on pipelines, the typical thermodynamic efficiency is 27% respectively. The organic Rankine bottoming system appears to be cost effective, and can be applied as retrofit to the second generation gas turbine installations.

For a third generation gas turbine with a typical efficiency of 35%, recuperation is not possible because there is not enough positive temperature differential between the

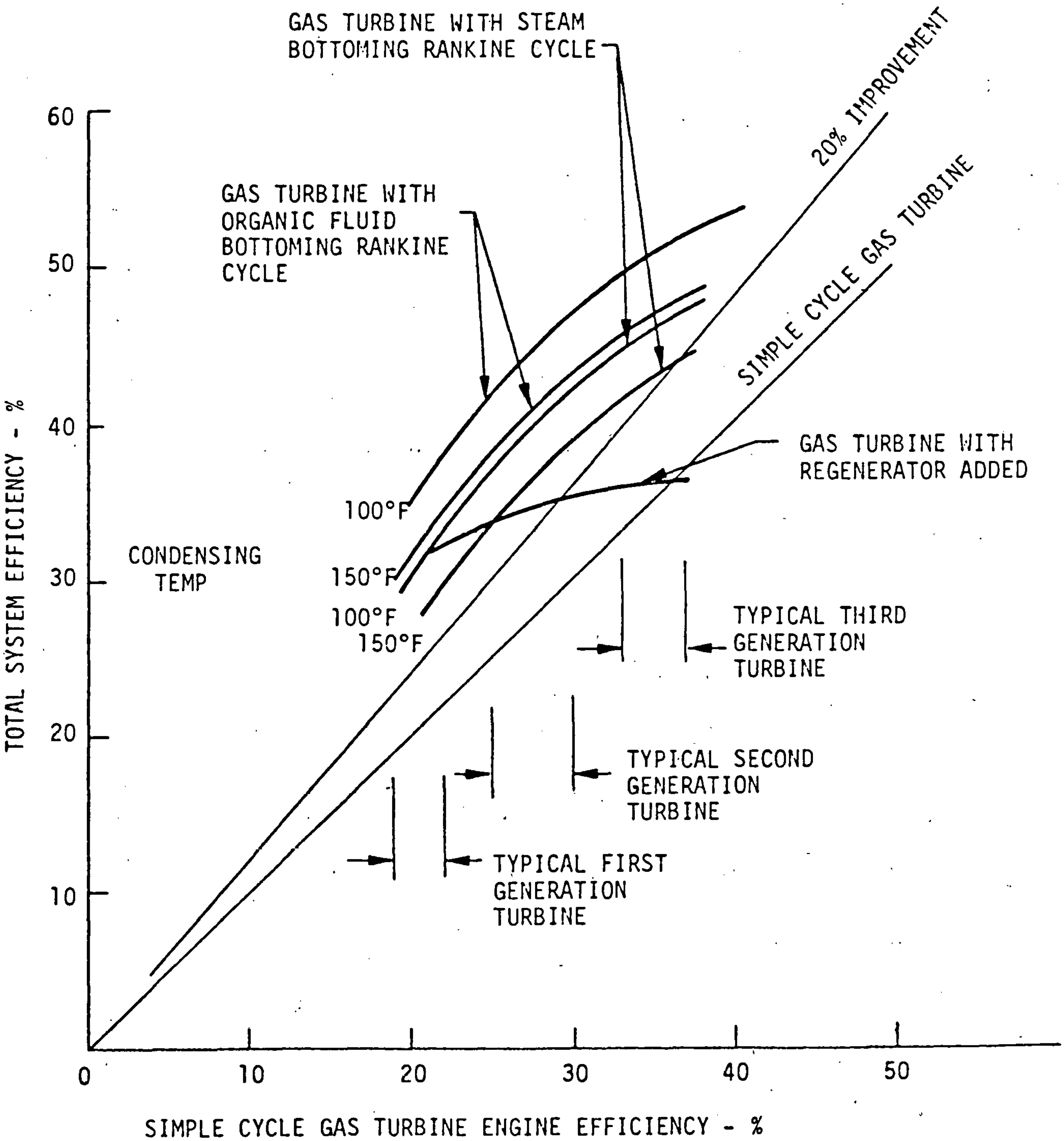

Fig. 4.1.1.4-1 Combined System Efficiency with Gas Turbine

power turbine exhaust gas temperature and the compressor discharge gas temperature. The steam Rankine bottoming system barely offers a 20% improvement but the organic Rankine bottoming system offers a combined efficiency of 47%, a 34% improvement over the open engine.

In summary, for gas turbine engines of the future, the organic Rankine bottoming system will offer far better than 20% improvement over the open engine, while the steam system will offer less than 20% improvement. In terms of product obsolescence, the organic system is much to be preferred.

The reason for this very significant conclusion lies in the upward trend of turbine inlet temperatures through the years. As inlet temperatures have risen, so have exhaust temperatures, so that the heat available to the bottoming engine increases significantly with each new generation of turbine. Along with these higher temperatures, pressure ratios increase, so that regeneration becomes less attractive, and finally impractical.

The situation with the gas reciprocators is discussed in Section 4.1.3 below, where it is shown that approximately 20% improvement can be realized by adding the organic Rankine even to the most efficient engines. This conclusion is opposite to that which was reached relative to the turbines. To repeat for emphasis, the older (first generation) turbines are not viable candidates for bottoming engine retrofit, whereas with the reciprocators the opposite is true, i.e., the older machines are indeed attractive retrofit candidates. Some further implications will be examined in Section 4.1.3.

4.1.2 Diesel Engine Improvements

The diesel engine has achieved an advanced state of development and has gained increasing acceptance in heavy duty vehicular as well as industrial applications in recent years. Today's modern, high output diesel engine approaches 40% brake thermal efficiency over a broad range of speeds and loads. While there is some potential for further improvement in efficiency through the use of high pressure ratio turbocharging and design refinements such as bore-stroke ratio, reduced friction, and improved combustion, such gains will probably be marginal. For achievement of significant gains in fuel economy, more basic changes are necessary. Foremost among these are combined cycle systems employing either the Brayton cycle or the Rankine cycle to recover part of the exhaust energy of the diesel cycle.

4.1.2.1 Diesel-Brayton Combined Cycles

This type of power plant in its usual form is known as a turbo-compound engine consisting of a diesel engine and a gas turbine whose outputs are connected together through gearing into a common output shaft.

Considerable work on turbo-compounding was carried out in the 1945-55 period, mainly in the aircraft field where the main benefits were realized in altitude performance. The Curtiss-Wright engine used in the DC-7 aircraft is a well known example of a 4-cycle spark ignition turbo-compound engine. The Napier Nomad turbo-compound engine (Sammons et al.,'55), developed during the same period, represented a very significant achievement as a high output, lightweight diesel engine for aircraft use. The Napier engine, shown in Fig. 4.1.2.1-1, consisted of a 2-stroke diesel engine and an axial flow compressor coupled together to form a common system: the 12-stage axial compressor provided a pressure ratio of 8:25:1 at maximum speed at an efficiency of 75-77.5%. The engine produced 3135 net hp with

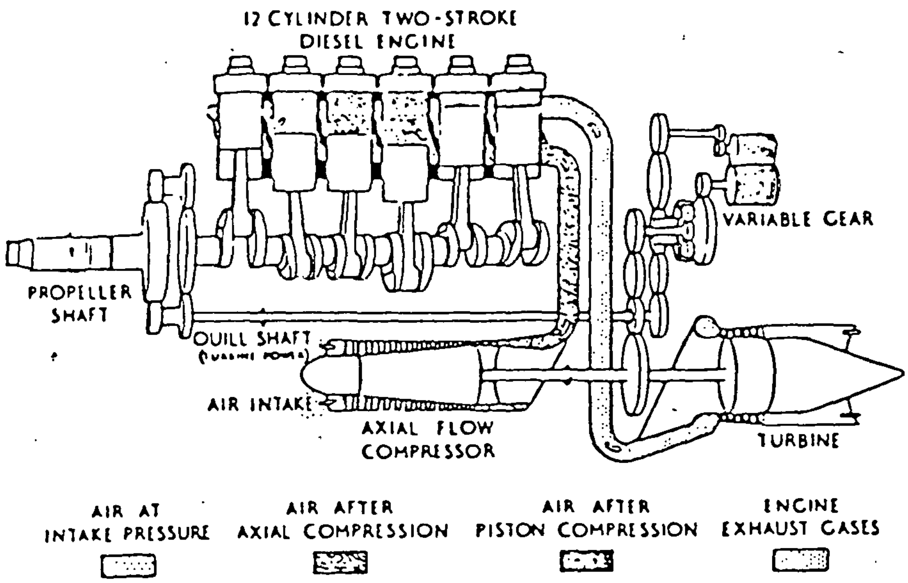

Fig. 4.1.2.1-1 Napier nomad turbocompound diesel engine - schematic arrangement

a specific fuel consumption of 0.345 lb/hp/hr and had only small variation in sfc with load. The engine was never put into production, primarily because its development came at a time when jet aircraft were superseding the propeller craft.

A more recent concept in turbo-compound engines is the Johnston engine (Hope et al.,'71) now undergoing development at Engine Systems, Inc., under contract to the Army Tank-Automotive Command. The engine, illustrated schematically in Fig. 4.1.2.1-2, consists of four basic subsystems: (1) a high-pressure ratio turbocharger, (2) an uncooled piston, (3) internal cooling of the cylinder liner and valve passages (details not shown in the figure), and (4) a separate exhaust turbine connected by reduction gearing to the engine crankshaft. The most unusual feature of the engine is the fact that the piston and cylinder are not cooled by conventional means. The engine utilizes a uniflow, two-stroke arrangement with excess airflow during the valve open position of the stroke, and the piston and cylinder are internally cooled by the excess scavenging air. The cooling medium is the working fluid, so that all the unavailable energy is contained in the exhaust and can be recovered in an exhaust turbine. The design by Engine Systems is based on a 4:1 pressure ratio and 83% efficiency in both the turbocharger turbine and exhaust power turbine. Matching the turbo unit to the engine for off-design conditions is planned to be accomplished by use of variable area nozzles in the power turbine.

The potential performance improvement of the Johnston engine over typical commercial diesel engines is shown in Fig. 4.1.2.1-3. The curve of specific fuel consumption vs. horsepower was computed as part of a study for the Army Tank-Automotive Command. The data indicate a potential reduction of 25 to 30% in fuel consumption along with an increase in horsepower of 100% for the Johnston engine over commercial engines of the same displacement, speed, and fuel/air ratio (Anderson et al., '75).

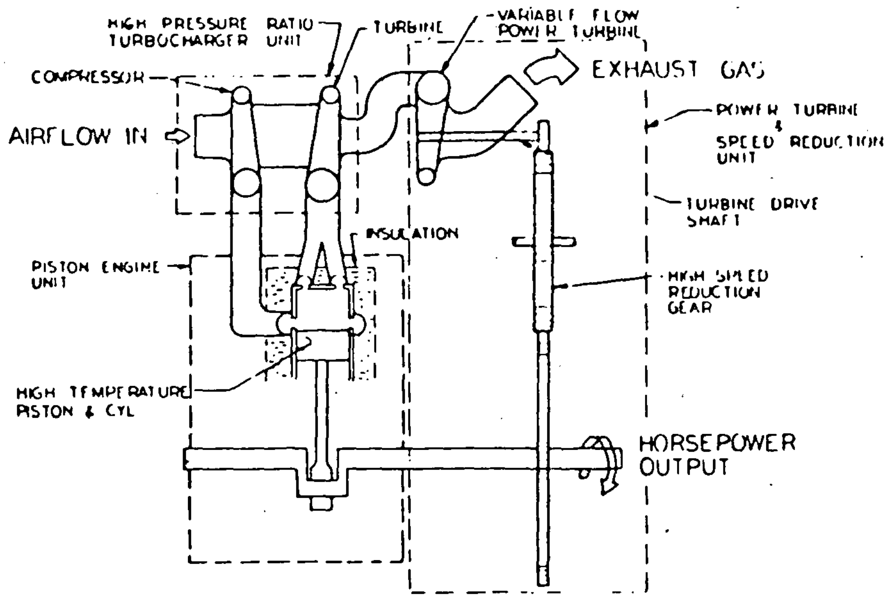

Fig. 4.1.2.1-2 Johnston engine schematic

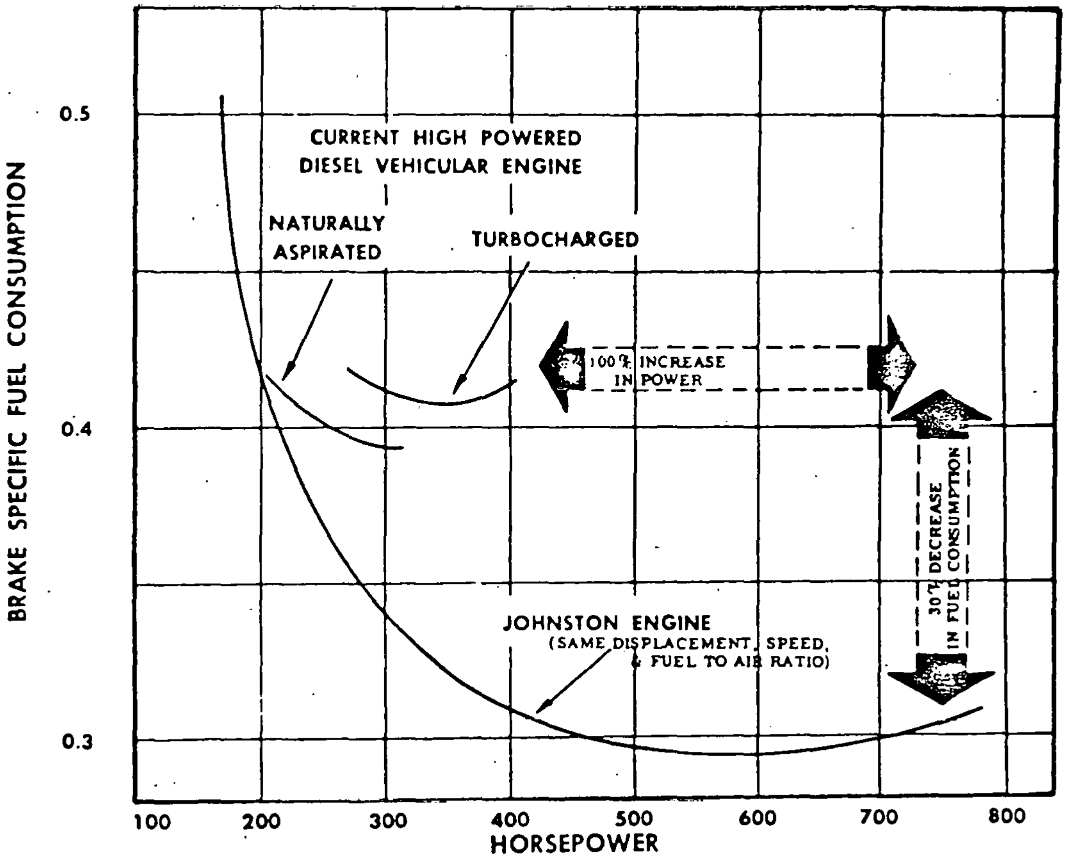

Fig. 4.1.2.1-3 Johnston engine performance

The key to performance in the Johnston engine lies in the method of cooling the piston and cylinder. Cooling is not accomplished in the conventional manner of rejecting heat to a separate cooling medium, either liquid or air. Conventional engines lose approximately 30% of their heat energy by this route. Instead, the piston-cylinder configuration is designed as shown in Fig. 4.1.2.1-4, with an elongated piston in a uni-flow scavenged, two-stroke arrangement, which keeps the piston rings on the bottom of the piston, away from the area of high metal temperatures. The cylinder is internally cooled with excess scavenge air and the cylinder head and valves by air flow which enters the exhaust stream, so that most of the cooling energy is recovered in the exhaust turbine. The primary engine parts must be manufactured from high temperature alloys to enable the engine to run at much higher temperatures (800-1300°F) than heretofore used in conventional engines.

Engine Systems, Inc., under contract with the U.S.Army Tank-Automotive Command, has designed and built a single-cylinder research engine around this concept and conducted limited tests which demonstrated the feasibility of the design, including the fact that the engine components could operate satisfactorily at elevated (1500°F) metal temperatures (Anderson et al.'75). In a subsequent contract, heat transfer and stress analyses were performed for each of the major components, i.e., the cylinder head, exhaust valve, piston, and cylinder liner (Anderson et al.'76). A study was also made of a General Motors 6V71 diesel engine, modified to the Johnston engine concept, as a potential candidate for the Army XM-723 vehicle. The study indicated a specific fuel consumption of 0.297, representing a reduction of 27-45% as compared with present commercial, high-output diesel engines used in this vehicle. The study also showed that the engine would offer an appreciable weight reduction and would fit into a slightly smaller envelope than the existing commercial engines in the XM-723 vehicle. Further development awaits availability of funds.

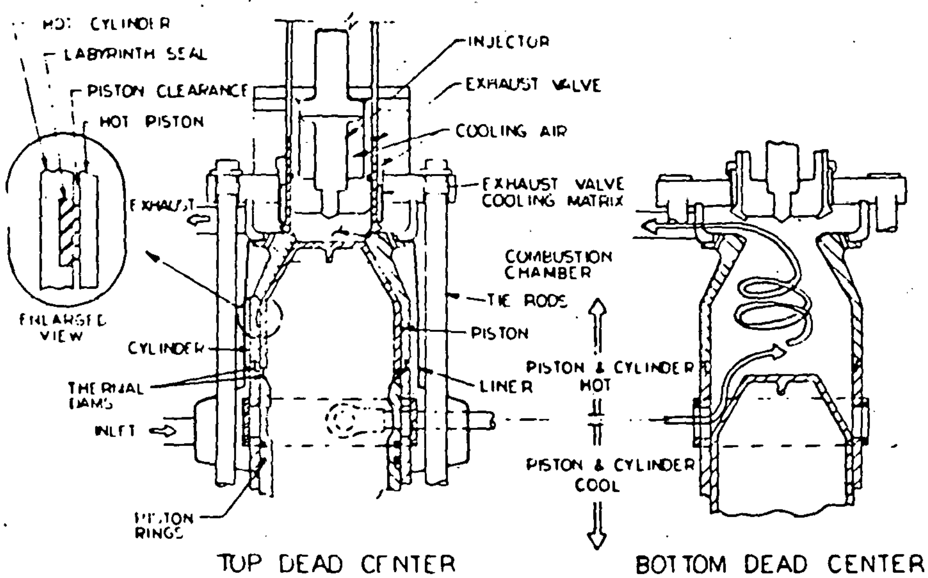

Fig. 4.1.2.1-4 Johnston engine-piston cylinder configuration

The methods by which the Johnston engine accomplishes internal cooling are particularly adaptable to two-stroke engines. Another development program, supported jointly by the Cummins Engine Co. and the Army Tank-Automotive Command, seeks to accomplish the same basic objective of internal cooling of a four-stroke engine. The basic approach, which is to insulate the cylinder and use ceramic hot parts, is described in a paper by Kamo ('76). Cummins is hopeful that basic feasibility of a 0.18 BSFC will be demonstrated in 1978.

It is important to emphasize the fundamental distinction between the type of turbocompounding that is represented by the Curtiss-Wright and Napier engines on the one hand, and by the Johnston and Cummins engines on the other. That distinction lies in the fact that these latter engines are cooled internally by air, which then enters the exhaust stream. Thus, the approximately 30% of input energy that is ordinarily lost in the cooling jacket is available to the bottoming engine. While both types of engine are turbocompounded, it is the internal cooling of the Johnston and Cummins engines that gives them their high potential for improved performance. The importance of internal cooling, and the opportunity that it offers, are further discussed in Section 4.1.3, where its exploitation in the pipeline industry is recommended.

4.1.2.2 Diesel-Rankine Combined Cycles

Recent development effort on power plants utilizing diesel engines combined with Rankine bottoming cycles has been prompted by rising fuel costs and the increasing emphasis on energy conservation. Earlier work on automotive Rankine cycle systems which was directed toward reducing air pollution has provided much of the technology for the diesel-Rankine cycle power plant. A number of industrial firms, including Aerojet-General Corp., Thermo Electron Corp., Sundstrand Aviation, Steam Engine Systems, Lear Motors Corp., Steam Power Systems, and Brobeck & Associates, have built and demonstrated Rankine cycle power plants in passenger cars or buses under sponsorship of federal or state government agencies. Others, notably General Motors Research Laboratories and Ford Motor Co., have developed experimental Rankine cycle systems for vehicles, either as in-house projects or utilizing subcontractors. While the majority of these systems have been steam powered, three of the companies (Aerojet, Thermo Electron, and Sundstrand) have used organic working fluids. Extensive investigation, supported by NASA, the AEC, and DOD agencies, have also been conducted in recent years on small organic Rankine cycle units for space applications and ground electric power. Studies to determine critical parameters for organic working fluids have been made by these and other organizations. These programs have likewise provided useful technology for current efforts on organic Rankine bottoming cycle systems.

Selection of the working fluid for a Rankine cycle system often proves to be an extremely complex process, since it involves several major considerations, such as: thermodynamic characteristics, thermal decomposition, compatibility with materials used in structural components, safety characteristics (flammability, toxicity, etc.), freezing temperature, and cost. Water has the advantage of being inexpensive, plentiful, chemically stable to high temperatures, and having well defined thermodynamic properties. However, it is not a good working fluid for low-temperature applications because its high latent heat of vaporization makes it necessary to employ low boiling pressures; therefore, cycle efficiency is low.

The general nature of this inferiority of water to other fluids may be seen by referring to Figs. 4.1.2.2-1 - 4.1.2.2-3. The first of these merely displays the Carnot-equivalent cycle on the temperature-entropy plot for later comparison. The second

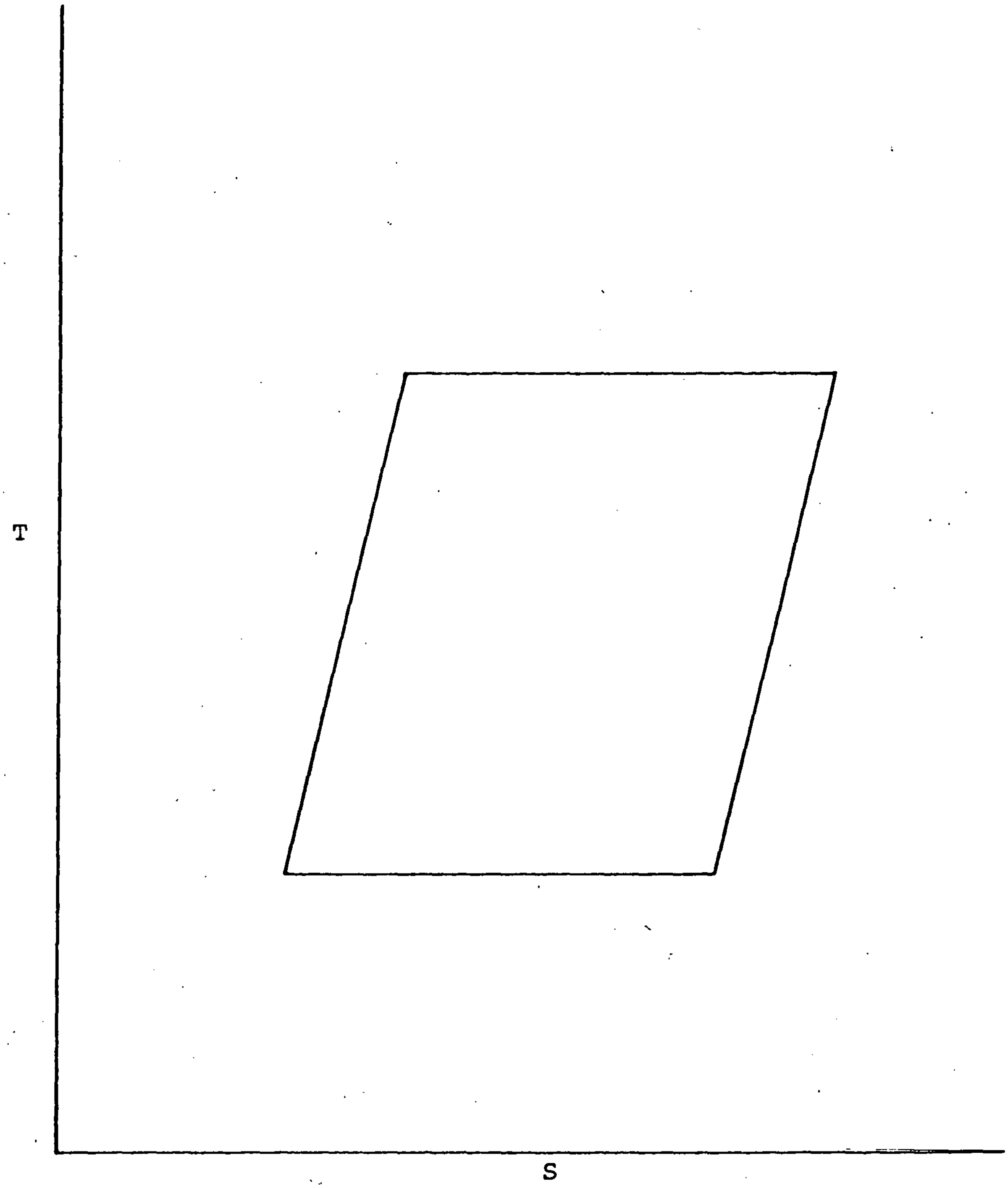

Fig. 4.1.2.2-1 Carnot-equivalent cycle

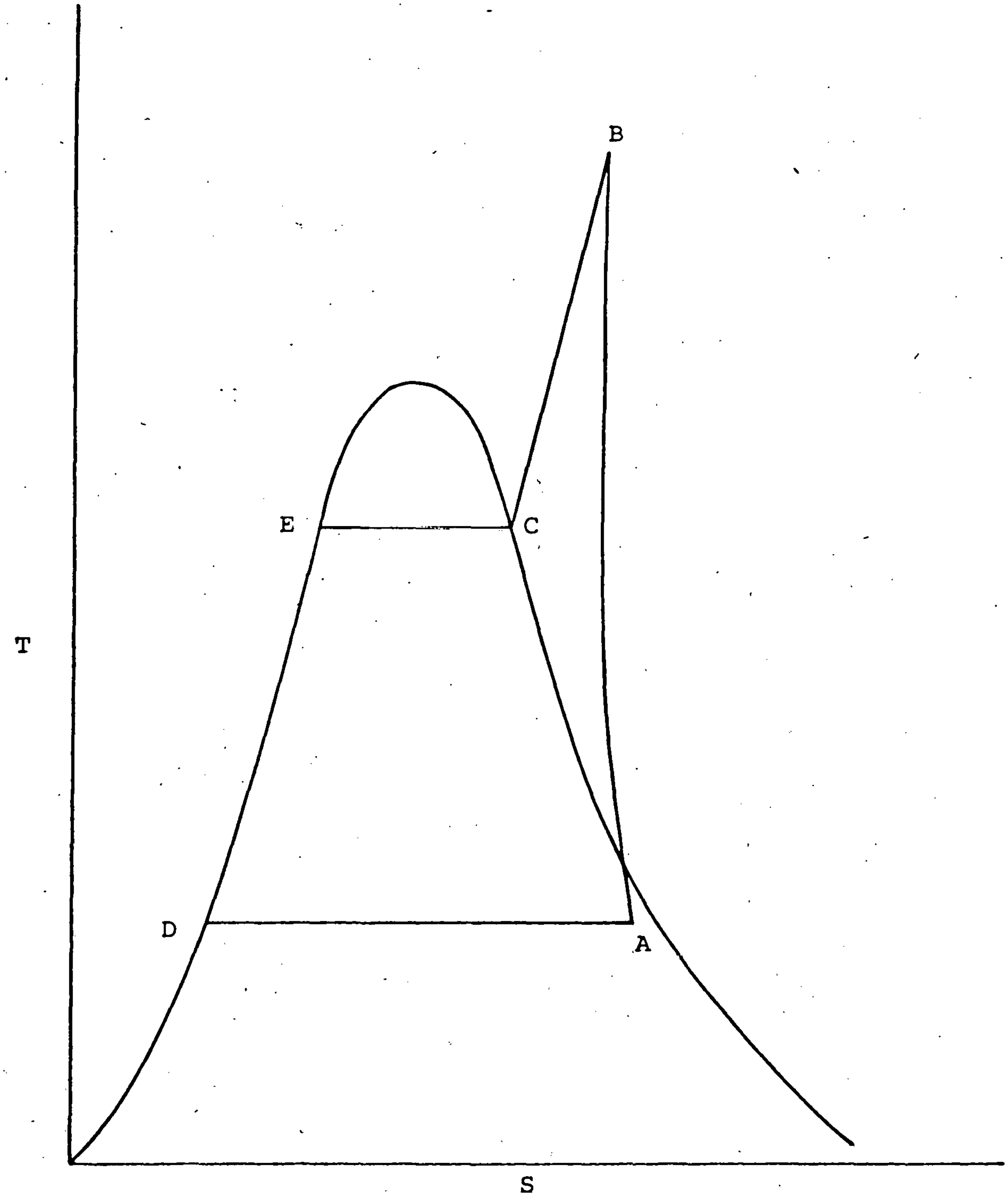

Fig. 4.1.2.2-2 Saturation line for water

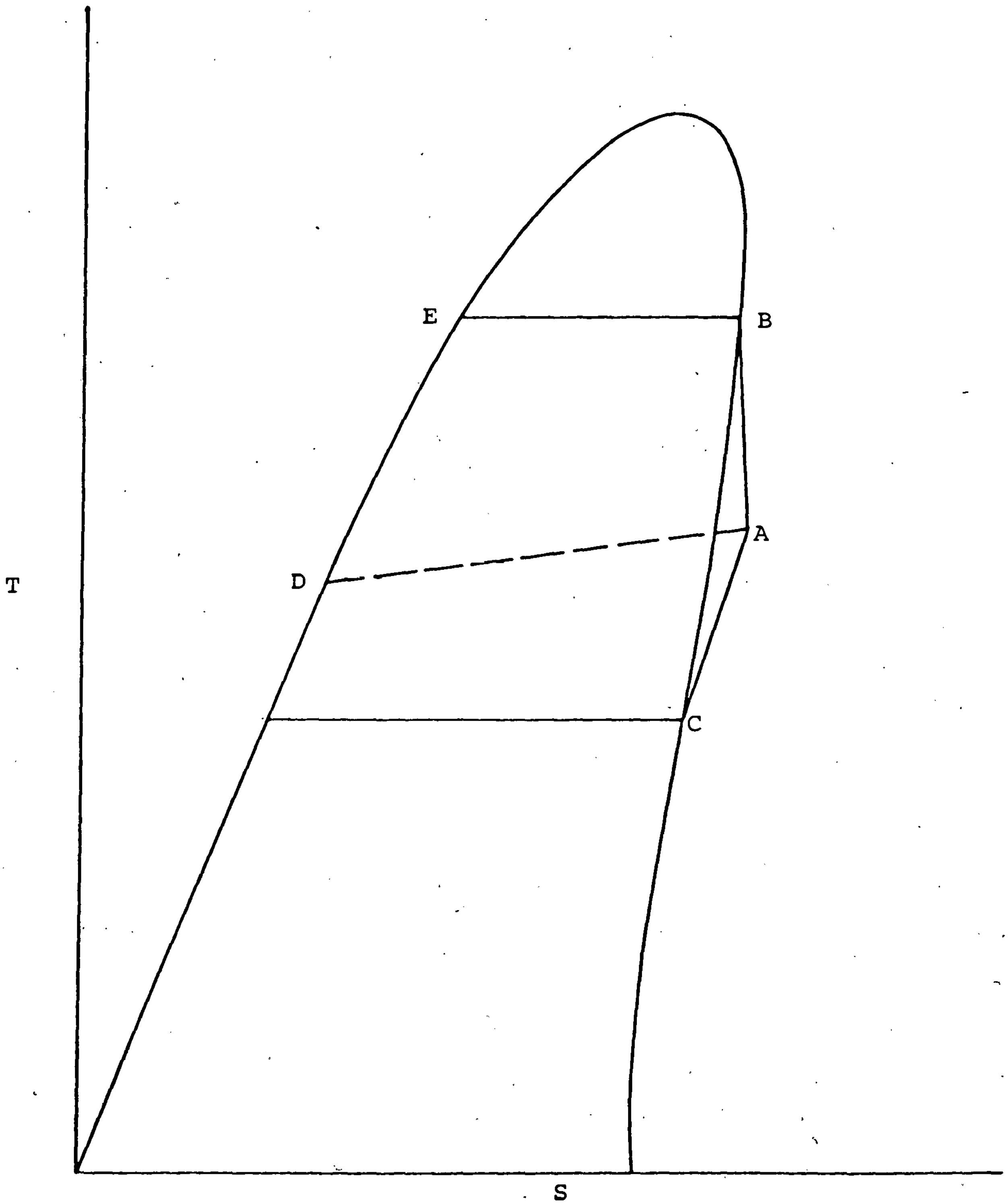

Fig. 4.1.2.2-3 Typical saturation line for organic fluid

figure shows the saturation line for water and a typical steam cycle. Because of the negative slope of the vapor side of the dome, to expand to a point A with acceptable moisture content in the steam, it is necessary to superheat to point B, much higher in temperature than point C, the boiling temperature. Instead of the parallelogram of the ideal cycle, one is left with the pointed cycle shape, in which only an infinitesimal quantity of heat is added at the maximum cycle temperature. The cycle is therefore very inefficient compared to the ideal.

On the other hand, as shown in Fig. 4.1.2.2-3, the organics generally possess a positive slope vapor line, which permits expansion to the superheated (dry) point A from the saturated point B. By regeneration to point C, the area to be compared with Fig. 4.1.2.2-1 is ADEBA and with the area ADECBA on Fig. 4.1.2.2-2. If the peak temperatures B are the same for both organic and water, the saturation pressure at which the water boils (from E to C in Fig. 4.1.2.2-2) is low and so is cycle efficiency. A number of other fluids besides water have been used or are being used in Rankine cycle systems. These include refrigerants (Freon 12 and 113), trifluoroethanol, fluorochemical (FC-75), isopropyl biphenyl, monochlorobenzene, toluene and pyridine. Each of these fluids has advantages and disadvantages, and at the present time no ideal fluid exists to fit all applications. In general, however, for low level heat recovery systems, most of the organic fluids are superior to water from a thermal efficiency standpoint.

One of the most significant recent efforts in diesel-Rankine combined cycles is the power plant developed by Thermo Electron Corp., using a truck diesel engine compounded with an organic Rankine system (Patel <u>et</u> <u>al</u>.,'76). A schematic of this system is shown in Fig. 4.1.2.2-4. The working fluid is Fluorinol-50, which is a mixture of 50 mole percent trifluoroethanol and 50 mole percent water. The design point characteristics

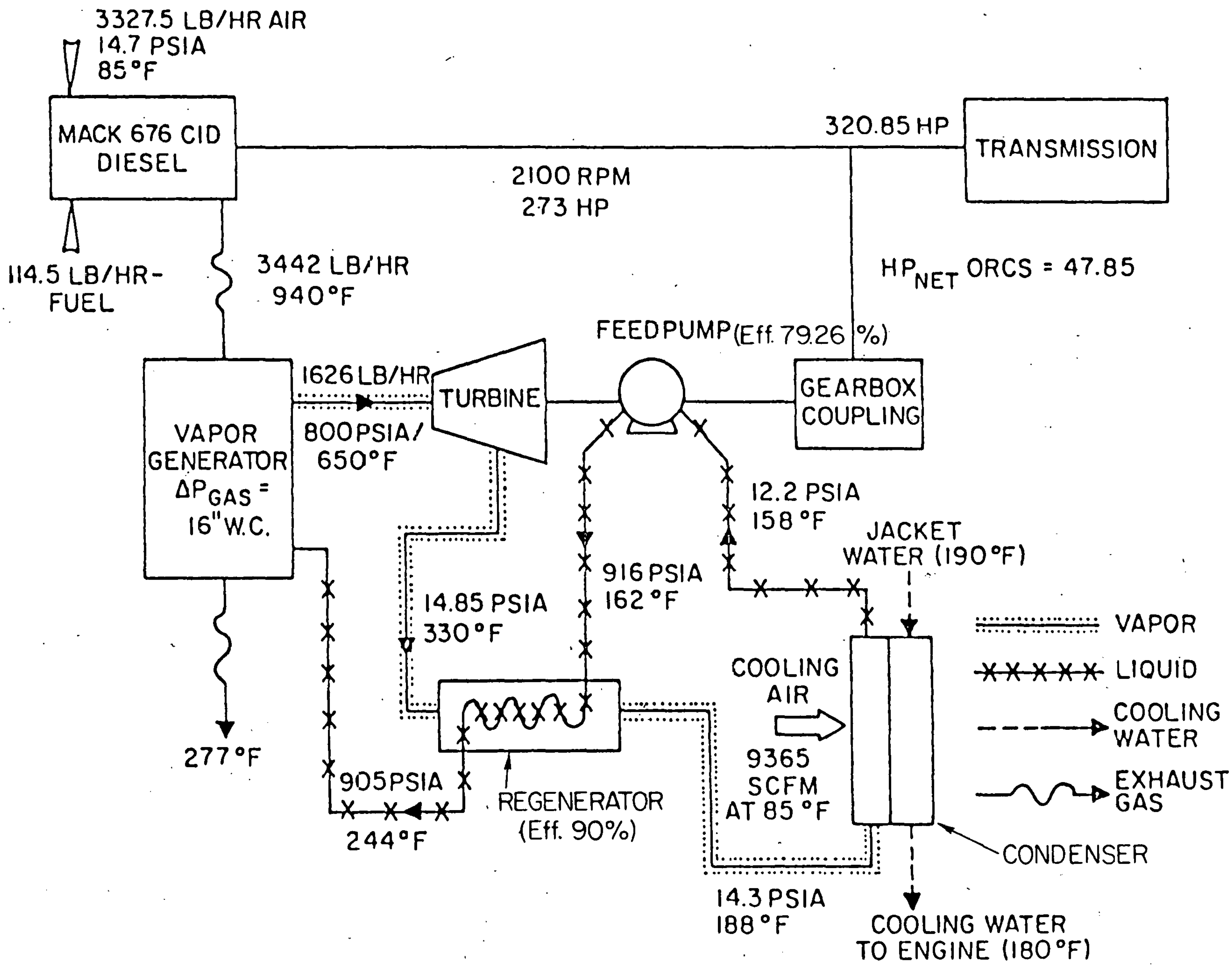

Fig. 4.1.2.2-4 Diesel-organic Rankine-cycle compound engine flow schematic

are given in the shematic. Since the turbine efficiency is the most critical parameter affecting overall system performance, a three-stage axial turbine running at 60,000 rpm with a projected efficiency of 75.5% was selected.

At the conclusion of a conceptual design study, a test system was assembled using a new Mack engine and existing but non-optimum organic Rankine cycle hardware. Performance mapping consisted of running 26 tests over the normal operating range of the diesel engine. The maximum power obtained from the ORCS was 35.6 hp, representing a gain of 13% in power without additional fuel. On the basis of these results it was concluded that, with optimum hardware, a 15% improvement in fuel economy over a typical duty cycle could be achieved. This would represent a potential reduction of 1.8 billion gals/yr. (120,000 barrels/day) in the near term transportation requirements.

A comprehensive study of high efficiency electrical power plants, consisting of diesel engines complete with organic working fluid Rankine cycle engines was prepared for the National Science Foundation, Division of Advanced Energy Research and Technology, by Thermo Election Corp. (Morgan et al., '74). The results showed that, using a commercially available 37.3 efficient diesel engine in a 5.5 MWe combined cycle system, an overall efficiency of 46.3% could be achieved (24% power increase over the basic diesel engine at zero additional fuel consumption. It was also shown that, using a large experimental 4-cycle spark-ignited gas engine with a combustion air refrigeration system, there is a potential for greater than 50% overall efficiency.

Another major effort planned for future application to a diesel engine is being initiated by Sundstrand Aviation under a contract recently awarded by ERDA. This program involves the design, development, test and demonstration of a 600 KW organic Rankine cycle, waste heat power conversion system. Prototype systems now being manufactured at Sundstrand are expected to be running by the summer of 1977. Following initial tests using a laboratory heat source, it is planned to install several systems in bottoming cycle plants with diesel engines at electric utilities. A 22.5% conversion efficiency is predicted for the organic Rankine cycle system, using toluene (CP-25) as the working fluid and based on cycle conditions of 550°F/300 psi at the turbine inlet.

4.1.3 Otto Engine Improvements

Otto cycle engines accounted for approximately 40% of the total horsepower installed in gas pipeline compressor stations in the United States between 1963 and 1973 (Gas Turbine Intern'l.,'74). They probably account for half of all presently existing installations. These are spark-ignition reciprocating units operating on natural gas fuel. Many of the units are integral engine-compressor types in which some of the cylinders are used for power and the remainder for compression. Others are matched engine-compressor sets in which the engine drives a separate reciprocating compressor.

Natural gas has a very high antiknock rating, making possible the achievement of high efficiency in engines through the use of high compression ratios. There are presently a number of manufacturers marketing engines with compression ratios of 10.5:1 or higher. The application of turbo-supercharging has achieved further gains in power output and fuel economy, with some of the modern, high-compression, turbocharged engines showing specific fuel consumption figures at full load as low as 6500 BTU/hp-hr (39% thermal efficiency).

A number of gas engines have been installed in recent years in on-site power (also referred to as "total energy") installations. Most of these installations have been in commercial buildings where the primary uses of waste heat energy from the engines have been for space heating, water heating, and air conditioning. The same concept has been explored for industrial applications with additional uses of heat energy such as process heating, cooling, evaporators, and possible conversion of any residual heat to mechanical energy (Baker '62).

Only limited effort to date has been directed toward adapting bottoming cycles to gas reciprocating engines to produce additional shaft power, however the same principles used in the diesel applications,which were discussed in Section 4.1.2 can be adapted to gas engines. Studies have shown that adding an organic Rankine bottoming system to a gas reciprocator can result in more than a 20% improvement in fuel economy, even in the most efficient engines, Figure 4.1.3-1.

Figure 4.1.3-2 shows a typical heat balance for a four-stroke, naturally aspirated gas engine. The unconverted heat is removed through four mechanisms :

(1) Cooling water
(2) Exhaust gas
(3) Lubricating oil
(4) Radiation

The low level of lube oil temperatures (190°F or less) and of radiative surface temperatures makes the conversion of heat from these sources into useful work impractical. The primary sources for waste heat recovery therefore are jacket-water and exhaust gas. The full load heat rejection from the jacket water is seen in Fig. 4.1.3-2 to be about equal to the useful shaft work and to the energy in the exhaust. There are two possibilities for recovery of this energy.

The first and obvious recovery possibility is by raising steam with the heat from the jacket. In principle, the jacket itself could be designed as a steam generator, but practical limitations would more likely favor simply using the cooling water heat for building heat or to power absorption refrigeration equipment. Current practice is to operate with cooling water temperatures around 250°F, which is quite compatible with standard heating and cooling equipment. At this low temperature, conversion to mechanical power is economically beyond consideration, and no innovative proposals for such conversion have been identified in the course of this study. It is therefore concluded that this possibility presents no R&D opportunity and no further work on the concept is recommended.

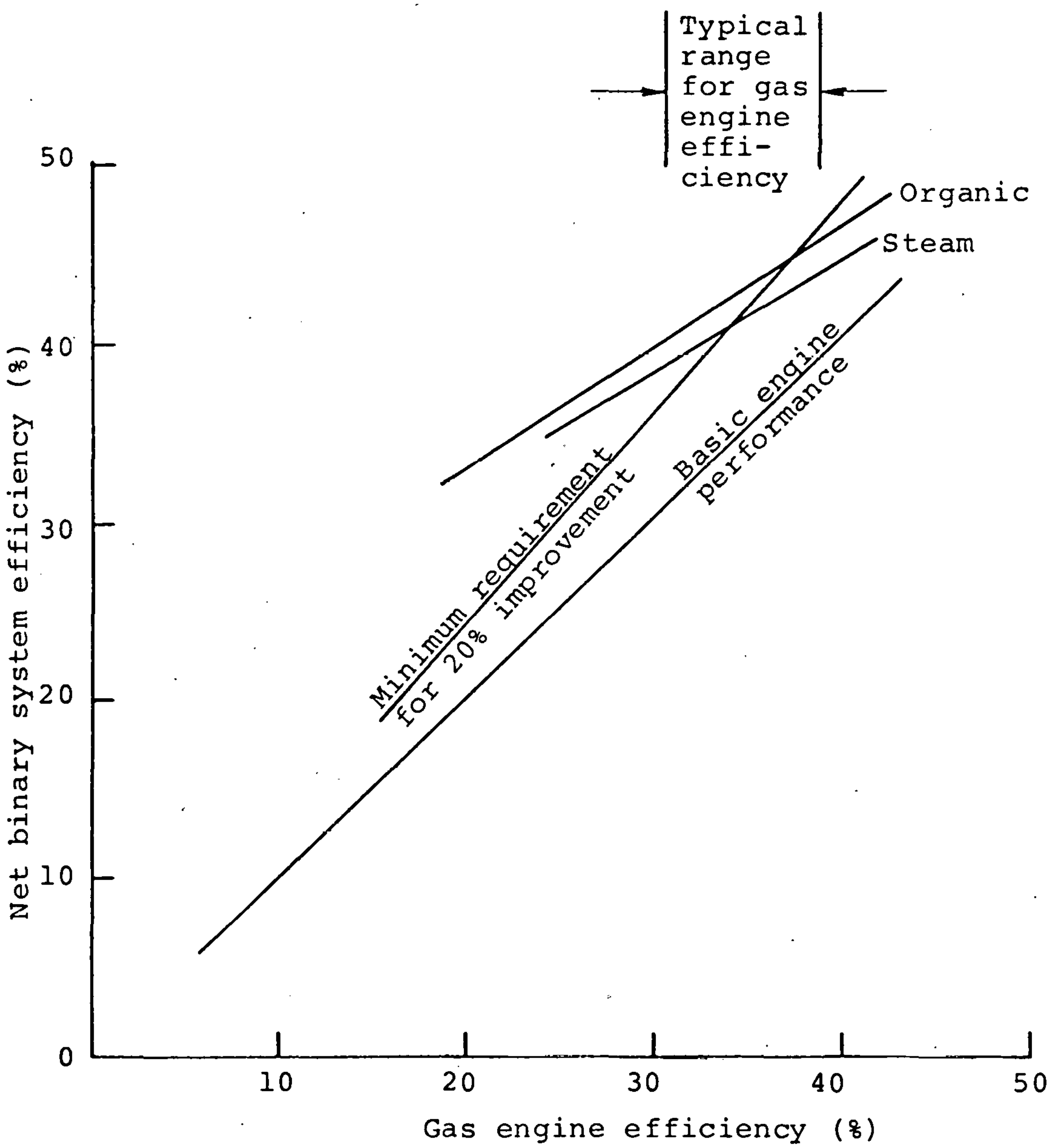

Fig. 4.1.3-1 Combined-cycle efficiency with gas reciprocators

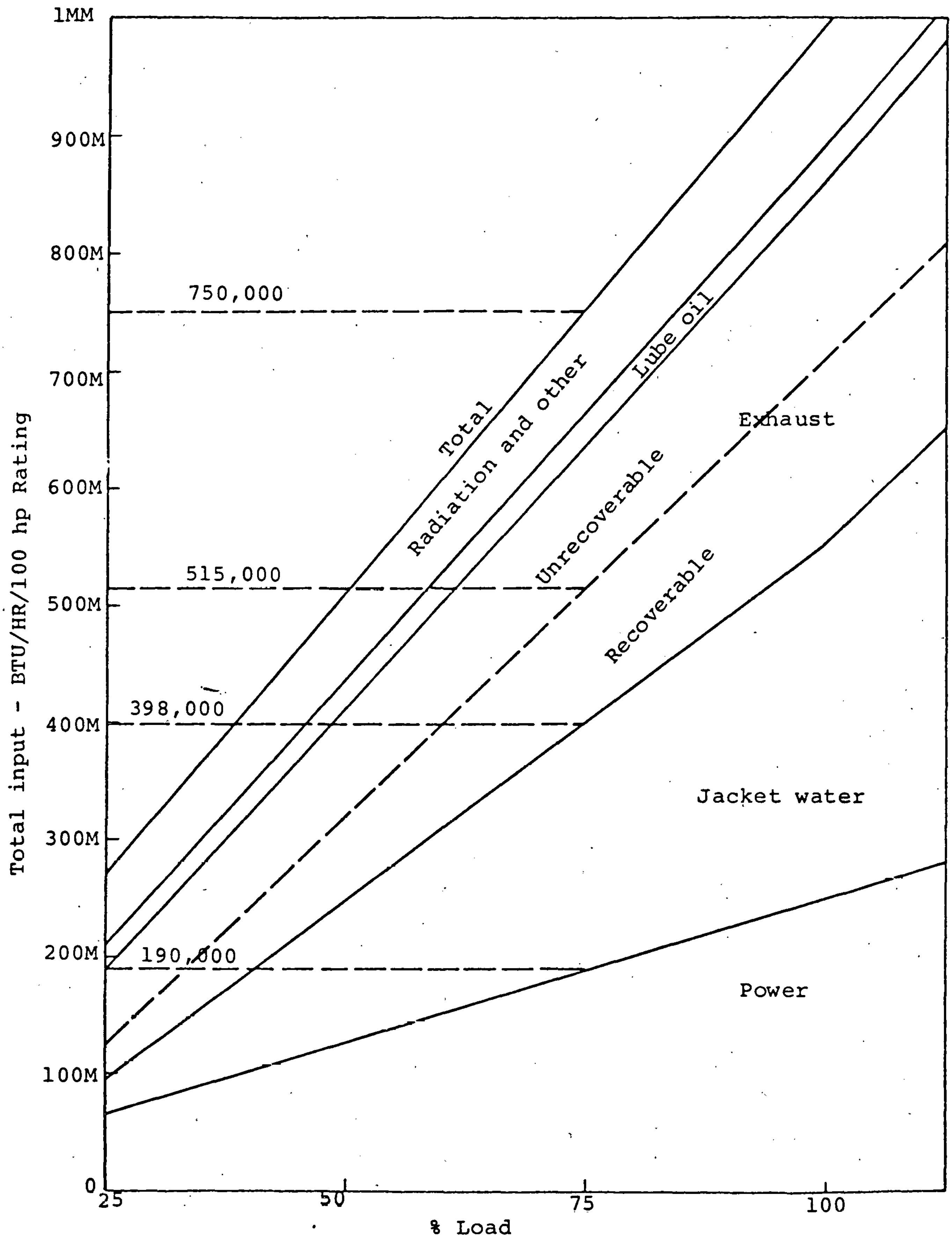

Fig. 4.1.3-2 Typical heat balance, 4-stroke, naturally aspirated gas engine

The second recovery possibility is to cool the engine internally with intake air in a manner similar to that discussed in Section 4.1.2, so that the heat appears in the engine exhaust at a much higher temperature than when removed in cooling water. This heat can then be converted to shaft work in a bottoming engine to supplement the shaft output of the primary engine. This of course is equivalent to turbo-compounding the gas engine in a way similar to that discussed for the diesel engine in Section 4.1.2 above. The compounding (bottoming) engine can be either an open cycle turbine or a closed cycle engine and in principle could be either a Brayton or Rankine engine and either a turbine or a reciprocator. Internal cooling of the gas engine therefore endows the engine with the same attractiveness for bottoming as the turbine, in which all of the unconverted heat appears in the exhaust. Figure 4.1.3-2 suggests that 25-30% of the input energy is available to the bottoming engine. In view of this attractiveness, two additional points are worth discussion.

First, if the internal cooling development can be conducted in a way that is applicable to both gas and diesel engines, the benefits would extend far beyond the pipeline industry. They would accrue in vehicular and marine transportation and in many stationary applications as well.

Second, it is worth noting that both the diesel and gas engines burn precious forms of fuel whose conservation holds attractions beyond the simple economics of fuel consumption. A rough estimate of the potential energetic and economic benefits of turbocompounding may be derived as follows. It has been found elsewhere in this study (Report R-3022) that the energy consumed in gas pipelines is approximately 0.7 Quad/yr. If only 40% of this energy is consumed in reciprocating engines, and if it is practical to retrofit half of these, and if internal cooling of these could recover the 25-30% of input which is presently lost in the cooling water, and if the bottoming engine can convert 25-30% of that, then the energy saving would be

$$0.7 \times 10^{15} \times 0.4 \times 0.5 \times 0.25 \times 0.3 = 1.05 \times 10^{13} \text{ BTU/yr}$$

If, as all evidence indicates, the true economic value of natural gas will soon exceed \$2/Mcf, the value of this saving will exceed \$20 million per year. And on a five-year payout basis, the justified R&D expenditures would exceed \$100 million.

Even if this rough estimate is several times too high, when the extreme breadth of potential application outside the pipeline industry is recognized, it is clear that an extremely attractive energy conservation opportunity lies in the internal cooling of pipeline engines. This attraction is further enhanced by recalling the discussion in Section 4.1.1.4, where it was seen that in contrast to the situation with the turbines, the older reciprocators are attractive candidates for bottoming retrofit. And by development of internal cooling, the advanced reciprocators of the future also become attractive candidates.

4.2 Electric Motor Improvements

Liquid pipelines are predominantly powered by electric motors of the integral horsepower polyphase a-c type. In the reference pipeline systems, the sizes range from approximately 1500 to 3500 hp. Motors of this size are generally built by the electrical manufacturers to customer specifications, including efficiencies and power factors. Efficiencies are generally in the range of 90 to 95%.

4.2.1 Improvements in Motor Efficiency

In conventional electric motors, power losses can be reduced by reducing core losses. There are several methods for doing this, e.g., adding more material to the magnetic core structure or using steel with improved core loss properties. Another method is to increase the cross sectional area of the conductors, which means adding winding material to the stator and rotor. Another technique is to shorten the air gap, thereby reducing the magnetizing current required. These are straightforward design approaches of the type which would be used by any electric motor manufacturer in meeting specifications for high efficiency, and do not appear to warrant any significant research and development effort to improve motor efficiencies over the present industry average.

The application of superconductors to electrical power equipment is another area which has received wide attention in recent years and is regarded as having excellent future potential for certain applications. Efforts are being concentrated on electric motors and generators as well as superconductive transmission lines.

Superconductivity is the total loss of electrical resistance shown by some materials when they are cooled to temperatures near absolute zero. Mercury, tin, lead, and many metal alloys become perfect conductors of electricity near absolute zero. Recent research has resulted in improved alloys such as niobiumtin and niobium-titanium, although the cost of these materials remains high (approximately $2 per gram for the high purity metal and $30 per pound for the regular grade).

The first efforts, in the 1960's, to incorporate superconducting windings in rotating electrical equipment were directed toward military applications in which size and weight were the dominant concerns. These programs established the feasibility of using superconductors in field windings and the need to shield the superconductor from any a-c magnetic field. Advances in superconducting magnet technology, namely, the commercial availability of stabilized NbTi conductors, were the principal causes of the recent efforts to develop large rotating electrical machinery with superconducting field windings.

In 1970, the International Research and Development Co. (IRD) of the United Kingdom demonstrated a 3250-hp homopolar electric motor with a superconducting NbTi field winding that could operate at full load in an industrial environment. Homopolar machines are variants of the Faraday-disk machine in which the armature (a thin circular disk) rotates inside an axial magnetic field. Such machines operate at low voltages and high currents (Hein,'74).

Although there has been an increasing interest recently in industrial application of high-rated homopolar machines, the main thrust of development to date has been for marine propulsion systems. The United Kingdom Ministry of Defense (MOD) has funded the development by IRD of a d-c superconducting generator and motor suitable for use as a propulsion system for high speed naval vessels. The U.S. Navy has launched a major effort to develop a superconducting motor generator for ship propulsion systems. Design contracts for a 30,000 hp m-g set have been awarded to the General Electric Co. and Garrett Corp. Two 3,000-hp prototype superconducting motors are being constructed by GE and are scheduled for sea trials beginning in late 1977. These units will be forerunners of 20,000 to 40,000 hp motors being planned for the future. Because of the low voltage, high current characteristics, it is necessary to use liquid metal brushes to carry the heavy current across the narrow gap separating the rotor and stator. A major advance in the design of these current collectors is claimed by GE engineers, using liquid sodium-potassium collectors which can handle 100 times the 60-A/in^2 capacity of solid collectors made of carbon-based materials and which do not wear.

For a-c machinery, the principal market area of interest has been central power stations. MIT, under a project funded by Edison Electric Institute, developed and demonstrated a 45-kva machine and initiated work on a machine with a capacity of 2 to 3 Mw. Westinghouse has built and tested a 5-Mva superconducting machine, demonstrating technical feasibility of their design. Design considerations prohibit the use of superconductors in the a-c armature as the losses would be too high; therefore, the superconductors are used in the d-c field winding only. Current collection problems for large blocs of power rule out rotating the armature winding, so the field winding is rotated. Problems associated with a rotating cryogenic system have presented a formidable challenge to the cryogenic and structural engineers. Although both MIT and Westinghouse have demonstrated workable designs, the economics of such generators has yet to be demonstrated (Hein,'74).

In assessing the adaptability of superconducting motors for pipeline use, there appears little likelihood that such machines will reach the stage of practical application within the next 10 years. The only types envisioned for near term use are the d-c homopolar type machines planned for marine propulsion systems, and these do not appear economically attractive except in large power sizes (20,000 to 40,000 hp). Although these machines offer a major advantage in power density over conventional motors, the gains in efficiency are only marginal and are largely offset by the losses involved in the refrigeration equipment. The motor sizes involved in liquid pipelines are much smaller and are a-c polyphase induction type. If superconducting d-c machines were used, it would be necessary to use a 3-phase rectifier bank in order to obtain a-c output, and this output would be synchronous a-c only. For these reasons, the expenditure of any significant R&D effort toward superconducting electric motors for pipeline application does not appear warranted at this time.

4.2.2 Improvements in Speed Variability

Pipeline motors operate upon alternating current, and are therefore constant-speed machines. Control is then effected by throttling excess pressure above that which the pipe is designed to accept. This throttled energy is of course wasted, and sometimes is a significant quantity. If a cheap way could be found to vary motor speed, either within the motor itself or in a variable-speed drive, a considerable saving of energy could be realized in petroleum-products pipelines.

An interesting possibility for accomplishing speed variability in pipeline pump motors is the use of DC motors powered by fuel cells. This concept is discussed in Section 4.3.6.2.

4.3 Fuel Cells

The fuel cell concept and characteristics are briefly described below and shown in Figs. 4.3.1-1 through 4.3.2-4 (ERDA76-54).

4.3.1 Fuel Cell Description

The fuel cell was first invented in 1839, but remained little more than a scientific curiosity until the first practical fuel cell was demonstrated 120 years later by Francis T. Bacon and J. C. Frost of Cambridge University. Since that time, fuel cells have been widely used in the space program where they have proved to be reliable sources of electrical power. However, their high cost and the difficulties involved in adapting their use to conventional hydrocarbon fuels have effectively retarded their adoption as ground power sources.

The fuel cell is an electrochemical device which directly combines fuel and air to produce electricity. As illustrated in Fig. 4.3.1-1, a hydrogen-rich fuel is electrochemically combined directly with oxygen from the air to produce electricity and water. Waste heat produced by the reaction process is removed with the exhausted air. Single fuel cells can be assembled in stacks of varying sizes to produce a wide range of output levels.

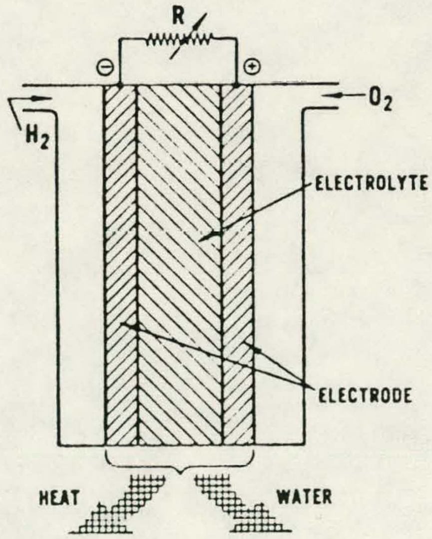

Fig. 4.3.1-1 - Fuel Cell Concept

Small fuel cell powerplants have been built and operated for a variety of space and military applications, and experimental demonstrators have been built for commercial applications. The specific arrangement of the powerplant is dependent on the fuel and oxidant used and the application requirements. For space applications, such as the Apollo manned voyage to the moon, the fuel cells operated on pure hydrogen and oxygen, supplying DC power for the spacecraft electrical needs. This very simple powerplant consisted of a cell stack and a few controls.

Commercial fuel cell powerplants operating on fossil fuel and air comprise three main elements as shown schematically in Fig. 4.3.1-2. The reformer section converts natural or synthetic hydrocarbon fuels into a more reactive form, usually a gaseous mixture of hydrogen with some carbon dioxide. The power section consists of a number of individual cells which convert the processed fuel with oxygen from the air to produce DC power. In

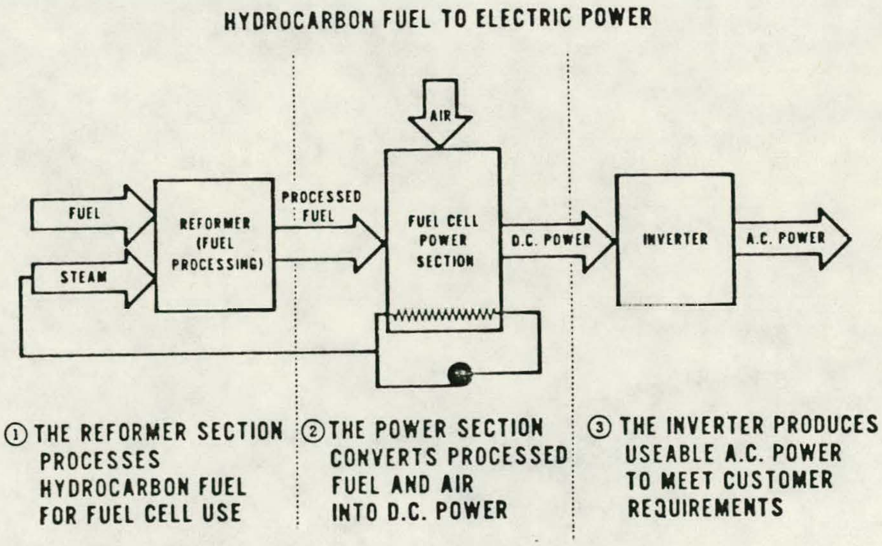

Fig. 4.3.1-2 - The fuel cell powerplant

the fuel cell stack the individual cells are connected electrically in series to permit generation of any voltage up to hundreds or thousands of volts DC. Connecting a number of cell stack assemblies in parallel permits generation of any power level from kilowatts to multimegawatts. The third major section is the inverter, which converts the DC output from the fuel cell section to alternating current (AC) electricity suitable for commercial applications.

4.3.2 Attractive Characteristics of Fuel Cells

Fuel cells have several unique characteristics which make them attractive for use in several power generation applications. These include the following:

(1) High theoretical conversion efficiency

Fuel cell powerplants ranging in power output from less than 100 kw to thousands of kilowatts are potentially capable of efficiencies comparable to the best diesel electric and large steam powerplants, as shown in Fig. 4.3.2-1

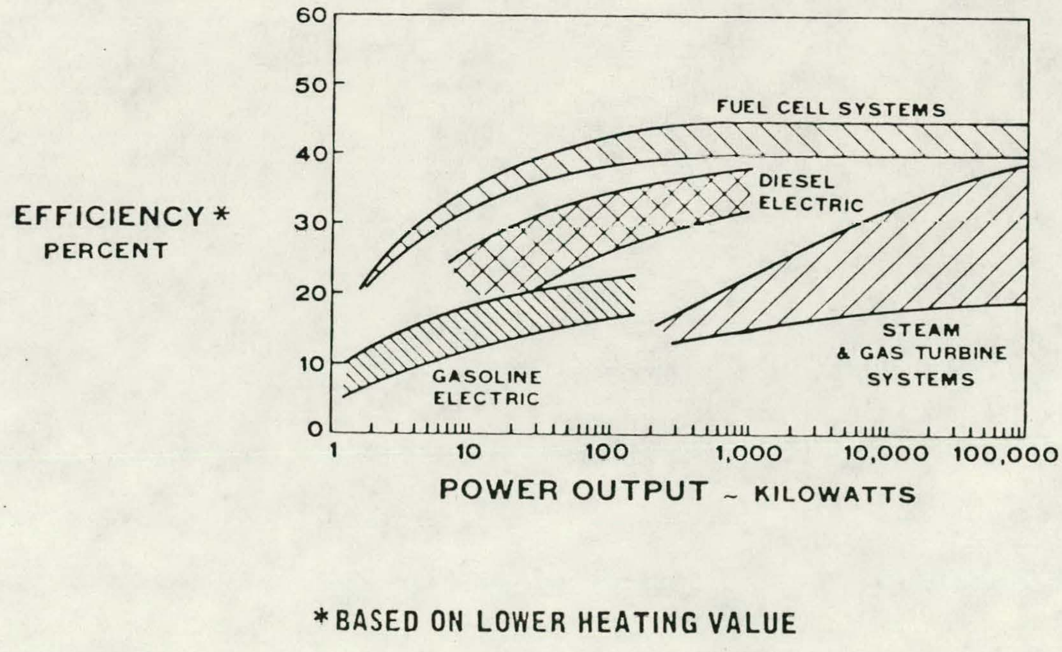

Fig. 4.3.2-1 - Fuel economy/efficiency for all sizes

First generation fuel cell powerplants, using phosphoric acid electrolyte, are being developed with efficiencies approaching 40%, while future advanced fuel cell concepts are expected to have efficiencies as high as 57%.

(2) High efficiency in small plant sizes

Efficient fuel cell systems based on present concepts can be built in sizes starting at about 25 kw. Small increases in efficiency can be obtained with increasing size up to about 1 megawatt, with little efficiency gain beyond this power level. Again referring to Fig. 4. 3.2-1, it is seen that fuel cell efficiencies are potentially quite good in sizes down to 10 kwe and below.

(3) Good part-load efficiency

Unlike conventional power generation equipment, fuel cell efficiency increases as load is reduced from rated power, down to about 40% load is illustrated in Fig. 4.3.2-2. This characteristic is important, as most generating equipment, except for base load plants, is required to operate over a wide range of outputs, in many cases averaging only about 50% over the equipment lifetime.

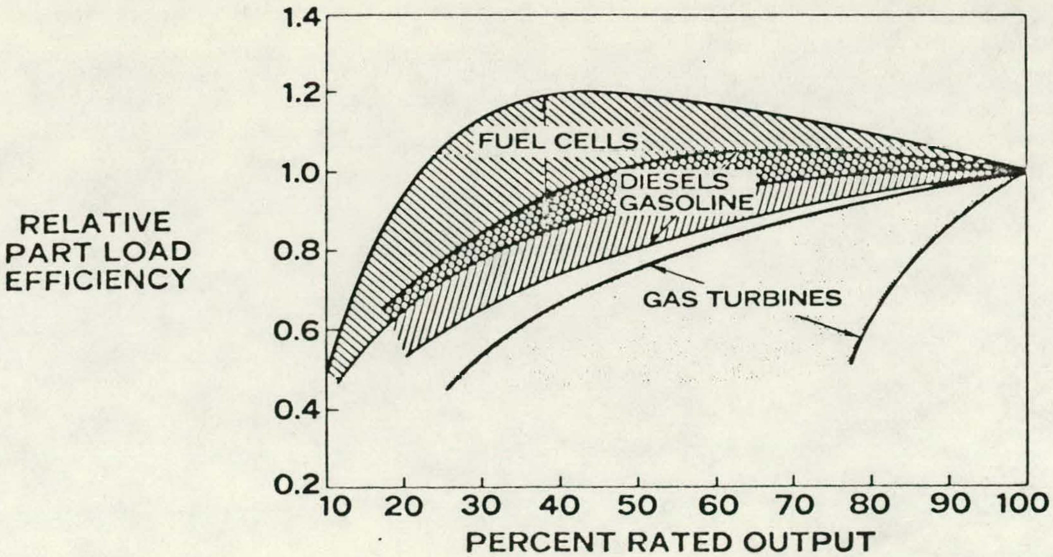

Fig. 4.3.2-2 - Fuel economy/efficiency at part load

(4) Low noise, thermal, and chemical pollution

Because of the static nature of the conversion process, fuel cell powerplants are inherently less noisy than conventional steam and internal combustion engine powerplants. The noise sources are confined to the ancillary equipment such as liquid feed pumps,and air blowers which are required for fuel cells. Cooling water is not required since waste heat can be transferred directly to the atmosphere. Measured emissions from experimental powerplants have shown that the fuel cell exhaust contains significantly lower emissions of particulates, oxides of nitrogen, and sulfur oxide, as indicated in Fig. 4.3.2-3.

POUNDS OF POLLUTANTS PER MILLION BTU HEAT INPUT

	FEDERAL STANDARDS*			
	GAS-FIRED CENTRAL STATION	OIL-FIRED CENTRAL STATION	COAL-FIRED CENTRAL STATION	EXPERIMENTAL FUEL CELLS**
PARTICULATES	0.1	0.1	0.1	0.0000029
NO_x	0.2	0.3	0.7	0.013-0.018
SO_2	NO REQUIREMENT	0.8	1.2	0.000023
SMOKE	20% OPACITY	20% OPACITY	20% OPACITY	NEGLIGIBLE

*FEDERAL STANDARDS EFFECTIVE 8-17-71
**YORK RESEARCH CORP., Y-7309 APRIL 1970

Fig. 4.3.2-3 - Environmental impact

(5) Siting flexibility

The characteristics of high efficiency in small sizes and low pollution allow considerable freedom in site selection of fuel cell powerplants. Those systems that require a minimum of fuel processing can be placed within buildings to make direct use of some of the waste heat for space heating and cooling. Fig. 4.3.2-4 shows the relative amounts and types of waste heat available.

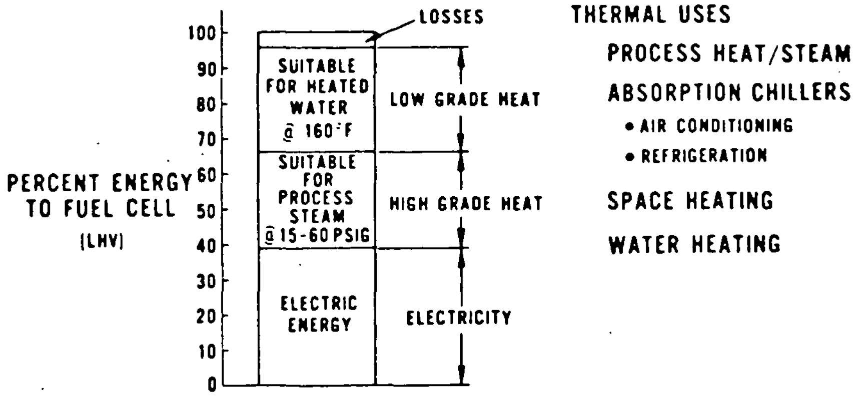

Fig. 4.3.2-4 - Waste heat recovery potential

4.3.3 Problem Areas

The primary problems associated with fuel cell development are cost and durability. For the fuel cell to produce electricity at room temperature, the electrodes must contain a very active electrocatalyst. The best catalysts discovered to date are very expensive noble metals - platinum and palladium. Nickel electrodes, which operate at higher temperatures, have electrode degradation and electrolyte decay problems. Fuel processing equipment is complex and expensive. Steam reforming, partial oxidation hydrocracking, and hydro-desulfurization are being tried with varying degrees of success. Steam reforming is a fuel processing technique now commonly being used with phosphoric acid electrolyte fuel cells. It requires a nickel catalyst, which is susceptible to poisoning by elements such as lead, chrome, and sulfur. Light distillate fuels low in contaminants give the best results with steam reforming, but the life of such a system is presently limited to about 15,000 hours (approximately two years) [Aronson, 1977]. Initial work has been done chiefly with phosphoric acid electrolyte, which has an operating temperature of 160 to 200°C. The next generation of fuel cells, expected to have higher efficiency, will probably use molten carbonate. Operating at a temperature of 500 to 700°C, this material is more tolerant of fuel impurities, but there are other problems yet to be solved. Life for existing molten carbonate cells is presently reported to be about 10,000 hours, as compared with a projected life of 50,000 hours by 1985 [Aronson, 1977]. The last element in the fuel cell system, the inverter which converts DC to AC, does not yet exist in the sizes needed for large-scale power conversion. Recent development of large SCR's which can be used in thyristor inverter circuits may be the answer, but there are a number of problems that must be overcome to meet the goal of 96% efficiency.

4.3.4 Fuel Cell Development Status

Fuel cells can be categorized according to their state of development into three generations [NASA, 1975]. Goals for the first and second generation programs of the electric utility industry are shown below [Pickett, 1977].

Characteristic	First Generation	Second Generation
Commercial introduction	1980	1985
Capital cost (1975 $)	$250/kw	$200/kw
Life	20 years	20 years
Stack refurbishment	40,000 hr	40,000 hr
Heat rate (Btu/kwh)	9300-9000	7500-7300
Thermal efficiency, %	36.7-37.8	45.5-46.8
Fuel	Naptha Natural gas Clean coal fuels	Distillate Naptha Natural gas Clean coal fuels

The two major programs in support of the electric utility industry fuel cell effort are the FCG-1 and RP114 programs, both initiated in 1972. The FCG-1 is sponsored by the United Technologies Corp. and nine major utilities, with the objective of bringing a first generation, 26-MW fuel cell power plant into commercial service by 1980. The nine utilities are supporting this effort through down payments on provisional orders for a total of 56 FCG-1's. The RP114 project was initiated by United Technologies Corp. and Edison Electric Institute (EEI). It is intended to broaden the application of fuel cells beyond the near-term needs of environmentally constrained utilities to result in a second generation powerplant for the utility industry to be used in a wide variety of applications.

The Electric Power Research Institute (EPRI), upon its formation in 1973, inherited the RP114 program [Pickett, 1976] In 1974, EPRI expanded its fuel cell activities and implemented a comprehensive, 5-year plan, addressing four major issues critical to achieving the objectives of the second-generation fuel cell powerplant. Critical issues include the following:

(1) Participation with ERDA in an early 4.8-MW demonstration of the FCG-1 technology, as a prerequisite to the second-generation program.

(2) Cost/availability of fuel cells in the near to intermediate future. In October 1974, a contract was awarded to Arthur D. Little, Inc. (RP318) to develop fuel cell scenarios and favorable means of integrating these within utility systems.

(3) Delineation of the techno-economics of fuel cells in a utility network. Of primary importance is a quantification of the benefits as well as a definition of the potential fuel cell market as a function of capital cost. A contract with Public Service Electric and Gas Co., New Jersey (RP729) is providing these assessments, using the scenarios from the RP318 project.

(4) A matrix of technology programs necessary to maximize the probability of achieving second-generation goals. These include projects to improve catalysts for phosphoric acid fuel cells (in the hope of stimulating a breakthrough necessary to achieve second-generation goals); to investigate molten carbonate electrode sintering mechanisms, since the molten carbonate fuel cell is the main thrust of the RP114 program; and to assess techniques for removal of CO_2 from the fuel stream in the alkaline fuel cell.

The FCG-1 program is being accomplished by scaling up from kilowatts to megawatts in steps: (a) a 1-MW pilot plant, (b) a 4.8-MW demonstration plant; and (c) 26 demonstration plants, all to precede the manufacture of 26 -MW FCG-1 powerplants on a production basis. The 1-MW pilot plant has been built and tested successfully. The 4.8-MW demonstration plant is under contract, jointly funded by ERDA ($25 million), EPRI ($5 million), and UTC ($12 million). Delivery is scheduled for mid-1978 and testing by early 1979. Results of testing the 1-MW unit indicate that the 4.8-MW plant will meet the performance goal of 9300 Btu/kw-hr at rated output.

Another major program called TARGET (Team to Advance Research for Gas Energy Transformation),sponsored by United Technologies Corp. (UTC) and a number of companies in the gas industry, is aimed at providing a family of fuel cell powerplants in the 25-250 KW range for on-site power generation in buildings and at industrial locations. This program began in 1967 and was sponsored by 28 natural gas transmission and distribution utilities. It has since expanded to include gas utilities and combination gas and electric utility companies. The program has been jointly funded at $56 million for effort from 1967 to the present. On the basis of market studies, 40 KW was selected as an appropriate size for initial entry into commercial and multiresidential markets. As a result, work has been directed toward development of a 40 KW powerplant, designated PC18. A pilot PC18 powerplant has been tested and is reported to have met or exceeded all of its operational and performance goals [Handley, 1976], including 40% overall efficiency over a wide operating range. This efficiency figure is based on the lower heating value of natural gas fuel, and includes the inverter as well as all other elements in the system. Limitations in utility company funding have precluded early completion of the development and demonstration required to introduce the 40 KW unit into the commercial market.

Third generation fuel cells are characterized by efficiency goals much higher than second generation systems. A goal of 50 to 60% efficiency has been established [NASA, 1975]. There is little existing technology for third generation systems. The only known candidate is the solid oxide electrolyte system. Demonstration systems are probably 15 to 20 years away.

NASA Lewis Research Center, at the request of ERDA and NSF, has studied a number of advanced energy conversion systems for central station, base load electric power generation using coal and coal-derived fuels [Warshay, 1976]. The program is identified as Energy Conversion Alternatives Study (ECAS). The General Electric Co. and the Westinghouse Electric Corp. were selected by competitive bidding to study these systems, one of which is fuel cell powerplants. In ECAS Phase I, three types of low-temperature fuel cells and two types of high-temperature fuel cells were subjected to a parametric analysis. An important part of the high-temperature fuel cell system study was the utilization of waste heat either by a steam bottoming cycle, the coal gasifier, or both. These are referred to as integrated cases. In the Westinghouse study of high-temperature fuel cells, efficiencies of 48 to 53% were projected for the zirconia solid electrolyte integrated case, and 46% for the molten carbonate system. In both GE and Westinghouse studies, the highest overall efficiencies of the low-temperature fuel cell powerplants (30 to 36%) was appreciably lower than efficiencies of the high-temperature systems.

4.3.5 Technology Assessment

While the feasibility of attaining high efficiencies in medium to large size fuel cell powerplants using fossil fuels is proven, extensive engineering development and demonstration remain to be done before overall economics, operating reliability, and durability are established. Development effort is needed on all major elements, including the fuel processing equipment, fuel cells, and inverters [NASA, 1975].

4.3.5.1 Fuel Processing Technology

Although fuel processing technology is well established, a direct application of this technology will not meet the expected requirements of commercial fuel cell powerplants for the near term or the future. The particular requirements of the fuel cell application dictate new requirements for the processor. For example, for a phosphoric acid fuel cell the specification limits for sulfur and CO impose stringent requirements on the fuel and fuel processor, and complex integration of the processor with the fuel cell stack is often necessary. The future U.S. energy scenarios anticipate the introduction of coal-derived fuels, which will impose new requirements on the fuel processor. In the first generation FCG-1 phosphoric acid fuel cell powerplant, the fuel processor development, though quite advanced, is by no means complete. Steam reforming is limited to lower molecular weight fuels (up to naptha). A number of advanced concepts are being developed under the EPRI program to extend the range of fuels to be processed by steam reforming. Other processes receiving attention are partial oxidation and cracking (thermal or catalytic). As far as second generation fuel cell systems are concerned, little other than systems work by UTC is being done on fuel processing for the molten carbonate system. Work needs to be done to determine the cells' tolerance to possible fuel impurities, their sensitivity to diluents, etc., in order to develop an optimum fuel processor for the molten carbonate system.

4.3.5.2 Fuel Cell Technology

For terrestrial applications, fuel cells can be categorized by operating temperature: low = ambient to 200°C, intermediate = 200°C to 700°C, and high = above 700°C. The first generation fuel cells will operate in the low temperature range. The problems most common to fuel cells in this category are in the area of electrocatalysis, invariant cell life, and system cost. In the intermediate temperature category the major problems are to develop the materials to assure long cell life in the severe conditions of high temperature and corrosive environment. In the high temperature category, using solid-oxide electrolyte, the materials problems will be more difficult to overcome than for the intermediate range.

4.3.5.3 Power Conditioning Technology

The basic element of the fuel cell power processor is the inverter, which accomplishes DC to AC power conversion. Inverters designed for low power applications are commonplace. However, large-scale inverters capable of handling megawatts of power have not been built, largely because of lack of semiconductors which can handle the large currents and voltages. In addition, the common designs are handicapped by poor efficiencies. UTC, working with semiconductor manufacturers, has developed and tested inverters for the TARGET program capable of handling 20 and 40KW power levels. They are also testing a 1.8MW unit inverter for use as a building block in the FCG-1 power processor, using components proved in the TARGET program. The scale-up is accomplished by operating the SCR's in a series/parallel combination to accommodate the higher fuel cell power voltages. Potential problems in the scale-up include:

(1) Simultaneous triggering of the SCR's, which are operated in series. Simultaneity is necessary to prevent excessive voltage buildup on the late-firing SCR's.

(2) Possible unbalanced load current sharing among SCR's operated in parallel; and

(3) Undetected failure of individual SCR's or drive circuits, resulting in overloading of the remaining SCR's.

The UTC goals show a target efficiency of 96% for the FCG-1 vs a demonstrated 90% on the 25KW TARGET inverters. Although some improvements can be expected from the scale-up to the megawatt size, efficiency, compatibility, and life tests will be required to verify design goals.

4.3.6 Pipeline Applications of Fuel Cells

In Section 4.2.2 above, it was noted that, if an economical and efficient means could be found to vary the speed of the pump motors, an appreciable saving in energy could be realized. To understand the losses that are thus avoided, it is necessary to examine products pipeline duty cycles in some detail.

4.3.6.1 Duty Cycles in Products Pipelines

There are two throttling conditions that arise in pipeline operation, depending upon whether the system is asked to pass maximum throughout or to accommodate to a specified pumping schedule below maximum capacity. These variations in duty arise from seasonal variations and from the necessity to switch products in order to accommodate all shippers' requirements. As an example of the former, the demand for fuel oil and LPG increases in the fall and winter, whereas in the spring and summer the traffic is predominantly gasoline. The switches to accommodate shippers arise because of the necessity, as a common carrier, to serve all shippers without discrimination. Thus common carrier products pipelines operate in products cycles of seven to ten days during which they guarantee to accept for shipment at least a (proportionate) part of every tender.

It may be noted in passing that similar situations can arise in crude lines if different types of crude are found in proximity to each other. However, this seldom occurs, so that the discussion here refers only to products lines. These conditions are discussed in reverse order, in the sections that follow.

4.3.6.1.1 Throttling losses at less than maximum throughout

Because of lower capital and maintenance costs and ease of operation, the large majority of liquid pipeline system pumps are driven by constant-speed electric motors. Except for older, small-diameter systems, generally crude oil gathering, the pipeline pumps are centrifugal units which are directly coupled to an electric motor driver and therefore rotate at the same speed as the motor. The available driver-pump options are, with few exceptions:

(1) Constant speed electric motor driving a centrifugal pump through a variable-speed coupling
(2) Variable-speed electric motor directly connected to a centrifugal pump
(3) Constant-speed belt drive connecting electric motor and positive displacement pump
(4) Variable-speed engine driving either centrifugal or positive displacement pumps.

The disadvantage of the constant-speed motor directly connected to the centrifugal pump is the inflexibility of pump speed and therefore the inability to vary the pumping rate and the velocity of the liquid in the pipeline.

With constant-speed motor-centrifugal pump units, flow variations are obtained by either or a combination of the following:

(1) Pumping through control valves that waste controlled amounts of the pump full discharge pressure in order to decrease line pressure and flow velocity to the specific flow required by the shipping schedule.
(2) Installing multiple pump and motor units at each pump station to operate either in series or parallel, in combinations that yield the specific flow required by the shipping schedule.

When a pipeline company is responsible for transporting a fixed quantity of liquid over a fixed time period with electric-motor driven centrifugal pumps, the management has the option of:

(1) Selecting the most nearly optimum combination of the available pumping units and pumping every minute of the time allotted at a constant average rate, throttling excess energy by a control valve, or
(2) Pumping with a selected number of units, without wasting pressure in a control valve, but pumping at a faster rate than necessary, thereby wasting energy both in start/stop and in increased fluid friction loss.
(3) Pumping under a combination of part time without throttling and part time with throttling, in such proportion that the average flow is the required one.

The lack of ability to adjust pipeline flow with economical and efficient variable speed motors or couplings results in one or more of the following economic losses :

(1) More pumps and motors are installed in a system than otherwise would be necessary.
(2) Energy is wasted when pressure is reduced by throttling in a control valve.
(3) Energy is wasted in fluid friction when flow in a pipeline is at a higher velocity than necessary.
(4) Energy is wasted by start/stop transients; however, this loss is not usually significant.

Although variable speed drivers and couplings are available, they are characterized by high initial capital costs and low efficiences, so that economic analysis for most pipeline systems results in the direct coupling of constant speed motors to centrifugal pumps.

It is instructive to consider an example representing the following conditions :

(1) Pumping schedule - 30,000 bbl/day of medium-viscosity petroleum products.
(2) Pump-motor combined efficiency - 85%.
(3) Pump and system characteristics as shown on Figure 4.3.6.1.1-1.

The operating options, together with the associated increases in necessary installed capacity and in energy consumption, are listed in Table 4.3.6.1.1-1. It is seen that the increase in available horsepower required is about 30%, and the energy wastage is from 5 to 15%. It is to be noted that the least wasteful option (1) does not include the energy wasted in the transient.

If the flow requirement were such that it became necessary to bring a third pump on stream, point e shows the flow below which throttling is required to maintain allowable line pressure. This limiting pressure constitutes the limit upon system capacity: The examples in the table relate to limits imposed by pump capacity. Both limits can be important in the operation of the pipeline.

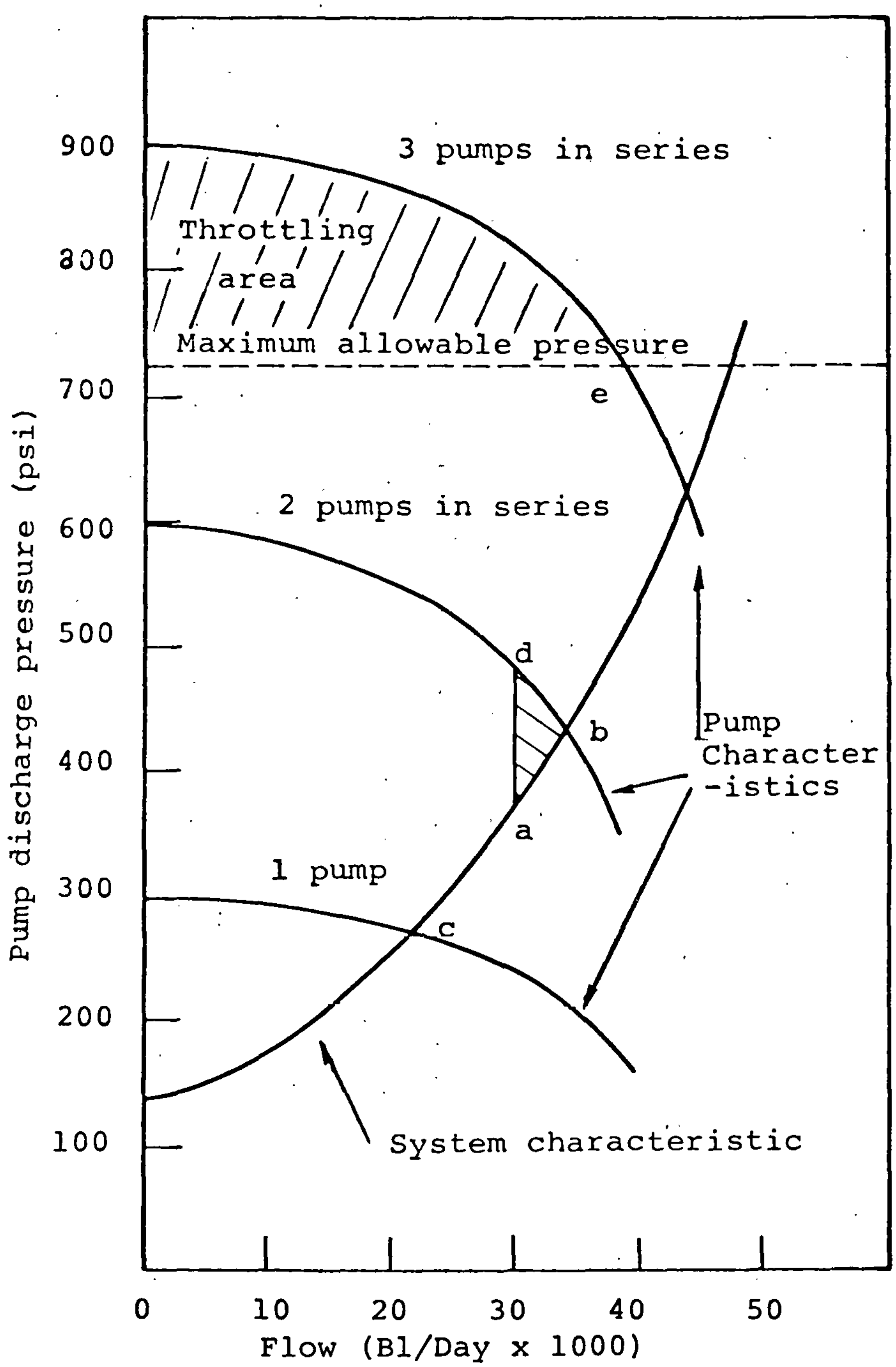

Fig. 4.3.6.1.1-1 - Pump and system characteristics

Table 4.3.6.1.1-1

Energy Wasted By Operating Options Available with Constant-Speed Pumps

(1)- Ideal - Operating Point a (unattainable)

30,000 bbl/day @380 psi for 24 hr = 24 x 228 hp = 5472 hp-hr

(2)- Operating on split rate schedule (attainable)

Point b

34,000 bbl/day @440 psi for 16.3 hr = 16.3 x 299.2 hp = 4877 hp-hr

Point c

21,500 bbl/day @276 psi for 7.7 hr = 7.7 x 118.7 hp = 914

5791 hp-hr

Increase over ideal	71.2 hp capacity	319 hp-hr average
	31.2%	5.8%

(3) - Operating two pumps and throttling (attainable)

Point d

30,000 bbl/day @495 psi for 24 hr = 24 x 297 hp = 7128 hp-hr

Increase over ideal	69 hp capacity	1656 hp-hr average
	30.2%	30.3%

(4) - Operating two pumps wide open (attainable)

Point b

34,000 bbl/day @440 psi for 21.17 hr = 21.17 x 299.2 hp = 6334 hp-hr

Increase over ideal	71.2 hp capacity	862 hp-hr average
	31.2%	15.0%

4.3.6.1.2 Throttling losses at maximum throughput

It has been seen in the preceding section that when the schedule calls for delivery at full capacity of two pumps, the optimum operating point is at b in Fig. 4.3.6.1.1-1. However, as will be seen from the discussion to follow, even when the demand exceeds capacity, it is seldom possible to achieve that optimum operating condition. The reason lies in the nature of the pipeline duty cycle, and can perhaps best be explained by an example, which accordingly is presented in Figs. 4.3.6.1.2-1 through -18. The case presented is a section of an actual, existing pipeline, and the duty cycles in the example are representative of those imposed upon that line in its day to day operations.

In -1 is seen the hydraulic gradient when the linefill is all gasoline, flowing 1000 bbl/hr. The associated pressures are identified, i.e., 682 and 64 psi at the upstream and downstream ends of the section respectively. The power requirement of 1000 hp, shown at the initial station at which it is applied, is the maximum single-unit capacity at that station.

In -2, a second 1000-hp unit has been brought on line at the midpoint station. Throughput is 1500 bbl/hr, only a 50% increase for a 100% power increase. In -3, a third 1000-hp pump has been brought on line at the quarter point. The effect of a fourth unit at the three-quarter point is shown in -4. Total power has now risen to 4000 hp and flow is 2200 bbl/hr.

Now, second 1000-hp units will be activated at each station in turn. When the second units are activated at the upstream station (-5) and at the midpoint station (-6), the flow is still only 2650 bbl/hr (shown as the gradient after the midpoint station).

When further increase in flow is attempted by activation of a second unit at the quarter point (-7), throttling becomes necessary to avoid excessive pipe pressures at the quarter and midpoint station discharges. A total of 337 psi is throttled away at these two stations. Addition of a second unit at the three-quarter point (-8, solid lines) increases flow to 3060, without throttling, but when an additional 600 hp, the last available unit, is brought on the first station, throttling again becomes necessary at the first and second stations (-8, dashed lines), while flow is increased by only 40 barrels. This is the maximum possible flow with an all-gasoline linefill. In practice, this last increment of 600 hp would almost never be justified for the small flow increment that it provides.

From the foregoing, it has been seen that throttling may be necessary at both maximum flow and a number of intermediate flows below the maximum. These throttling losses could be avoided if an economical, efficient, variable-speed motor or coupling were available.

It may be noted that the final discharge pressure at the downstream end of the section varies in the foregoing cases between a low of 11 psi and a high of 99. The pressure requirement at that point may vary, depending upon whether the fluid is being diverted into storage or into tankers for transport. Whatever the requirement, any excess pressure represents an energy wastage which could be eliminated if appropriate means of pump speed control were available.

Now suppose that the schedule calls for fuel oil to begin movement through the line, following the gasoline. The profile which exists as the oil approaches the quarter point section is shown in -9. The last available pump unit of 600 hp at the initial station, shown in -8 as the dashed lines, is not in service in -9, leaving 2000 hp in service at that station.

The allowable line pressure is exceeded at the first and third stations, resulting in throttling energy waste at those two points.

There is also another interesting difference between the situations portrayed in -8 and -9, namely, at the downstream end of the section. With the throttling at stations 1 and 3, the flow is reduced to 2700 bbl/hr in -9 from the 3060 in -8, and one unit at station 4 has been deactivated. Even so, the pressure at the downstream end of the section is 104 psi, which may exceed the requirement at that point and thus constitute a waste.

As the fuel oil approaches station 3 in -10 it is seen that the 1000-hp unit in station 1 has been replaced by the 600-hp unit to avoid throttling from 1422 psi to 1330 psi at station 1 and from 1608 to 1461 at station 2. It is still necessary to throttle station 2 from 1461 psi down to 1328 psi. Flow is now down to 2500 bbl/hr from the previous 2700.

In -11, as the fuel oil is approaching station 3, two profiles are shown. The solid line is the profile that would be required to hold the 2500 bbl/hr flow from the previous figure, i.e., activation of a second 1000-hp unit at station 4. Station 2 would then be throttling from 1553 pis to 1328, and station 3 from 1503 to 1424, while the downstream discharge pressure would be 530 psi. By not activating the additional pump at station 4, only 80 bbl/hr of flow is lost, discharge pressure is held to 34 psi, and some, though of course less, throttling is still necessary at station 2, from 1461 psi to 1328, and at station 3, 1424 to 1330.

The situation when the entire line fill is fuel oil is depicted in -12. Flow is 2500 bbl/hr with eight pumping in service, and throttling is necessary at stations 2, 3, and 4.

Now consider the case in which propane instead of fuel oil follows the gasoline. Earlier, -8 has shown that, with all nine pumps operating, 3100 bbl/hr of gasoline could flow. When propane is introduced at the upstream end, the profile is as shown in -13 as the propane approaches station 2, and as in -14 as the propane approaches station 3. The controlling segment is always the one just ahead of the propane. In -15, solid line profile, as propane approaches station 4, that station requires 2000 hp to maintain the flow. However, the dashed line shows that 1000 hp can be dropped from that station at a sacrifice of only 40 bbl/hr. Note also that gasoline has been introduced upstream and has filled the first segment, requiring that 1,000 hp be added at station 1, of which a large part is throttled away.

In -16, the propane is approaching the terminal. The 600-hp unit in station 1 has been dropped and segment 2 is controlling, 130 psi being throttled there. Flow is 2849 bbl/hr. As the gasoline approaches station 3 (shown in -17), segment 1 takes control. The upper line shows the profile that would obtain if all three pumps in station 1 remained on line. The entire pressure contribution of the 600-hp unit would be throttled away, 239 psi at station 1 and 145 (1504 down to 1359 psi) at station 2. Accordingly, that pump is switched off without loss of flow, and segment 2 assumes control.

Figure -18 simply shows the profiles when the linefill is all propane. With its lower density, the full head with all units pumping never exceeds allowable line pressure. The last 600-hp increment of power at station 1 gains 120 bbl/hr of flow.

Figure 4.3.6.1.2-19 shows how the actual unit power cost varies with throughput. Disregarding flows less than 1000 bbl/hr, below which the demand change dominates, it is seen that for a factor of 3.2 in throughput, i.e., 1000 to 3200 bbl/hr, power cost rises by a factor of 4.4, or approximately as the 1.28 power of the flow. Figure 4.3.6.1.2-20 shows barrel-miles and power cost in cents/barrel mile for the 15-month period of the study.

The throttling pressures are plotted in Fig. 4.3.6.1.2-21 for the full-capacity profiles, -8 through -17. In the uppermost plot of energy wasted at the terminal, it has been assumed for simplicity that pressure up to 15 psi could be used in moving the product to storage, but that any pressure above that value represents wasted energy. Since the times between the successive figures are roughly the same in each instance (5-6 hr), if the operations are continuous, then these pressure profiles are approximately equivalent to time traces of the energy wasted per unit of flow. It is clear that the energy wasted in typical operation is not insignificant.

4.3.6.2 Fuel Cells with DC motors

The reader will have anticipated the principal conclusion of this section: direct-current motors, with their capability for infinitely variable speed control, offer a means to avoid the throttling waste discussed above. They are not used, of course, because of the expense of the converter to convert the AC power provided by electric companies to DC. But the fuel cell is a source of direct current and thus is compatible with the variable speed DC motor. It will be recalled that the fuel cell development programs that were described in Section 4.3.4 above included the development of an inverter to convert the fuel cell DC current to AC. If fuel cell development is justified for the electric utility situations which require the inverter, then it should be even more attractive for the pipeline application where the inverter is not needed. Qualitatively at least, one can say that pipeline service offers one the most attractive application for the initial demonstration and commercialization of large fuel cells. To render this statement quantitative requires some discussion of the economics of pipeline energy consumption and of prospective fuel cell efficiency.

Referring again to Fig. 4.3.6.1.2-21, it has been observed that the abscissa also roughly represents a time scale, if the operations described above are continuous, with each spacing representing approximately six hours. A further step in the analysis would be to calculate these time steps exactly and convert the throttling pressures to horsepower and replot the results, giving time profiles of power wasted at each station. Integration of these curves would then yield the energy wasted. However, for this calculation to yield an estimate of the energy actually wasted in the industry, it would be necessary to write a computer program, characterize each pipeline individually, and input that data, along with the full-year duty cycle of each pipeline into the computer, a task which is clearly impractical.

A very rough estimate of the energy wasted may be developed in the following way. The total operations expenses of the 105 interstate oil pipelines in 1974 was $357,122,000. The ICC statistics do not further subdivide this figure, although in the individual company reports the cost of fuel and energy is reported. Reference to a few of those reports reveals that a figure of 40% for that cost, fuel and energy is not atypical. If that figure were valid, the energy cost for all companies would have been approximately $140 million. For the set of full capacity profiles presented above in Figs.4.3.6.1.2-8 through -17, the average throttling head loss is approximately 6 to 8%. If this fraction were valid nationally, the total wastage would be around $10 million annually. This order of potential saving is clearly enough to justify a R&D program of several million dollars.

Another question of course is that of fuel cell efficiency *vs* utility electric power. The latter seldom reaches 25% after all power generation and transmission losses are taken into account. It was noted in Section 4.3.4 above that fuel cell efficiencies of 45 to 47% are expected in the large second-generation units presently under development. When losses in conversion and transport of the fuel are accounted for, the net efficiency of the fuel cell power source will be reduced somewhat but will still far exceed the electric utility power source.

It is not suggested that the ready adaptability of fuel cell-DC motor power sources to pipeline applications is an established conclusion. Any experienced operator of motorized equipment will testify to the large maintenance burdens which are imposed by DC motors. This fact is very important in pipeline operations, where the motor locations are often remote and/or unattended, and where round-the-clock service is often required. However, the potential clearly exists for significant energy saving, and it is therefore strongly recommended that further work be done to define an appropriate R, D&D program for the applications of fuel cells in pipeline service.

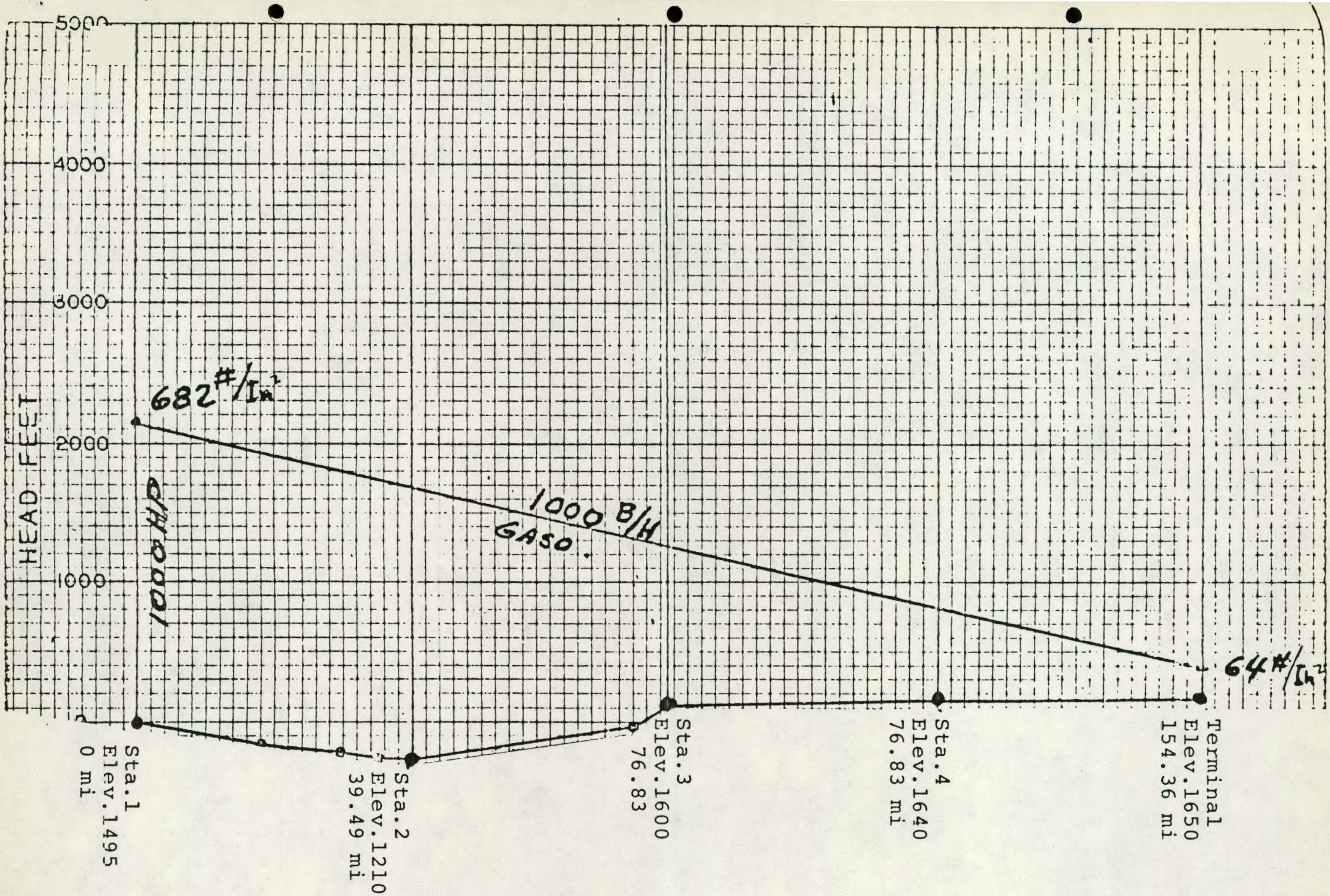

Fig. 4.3.6.1.2-1 - Hydraulic gradient, one pump, all gasoline

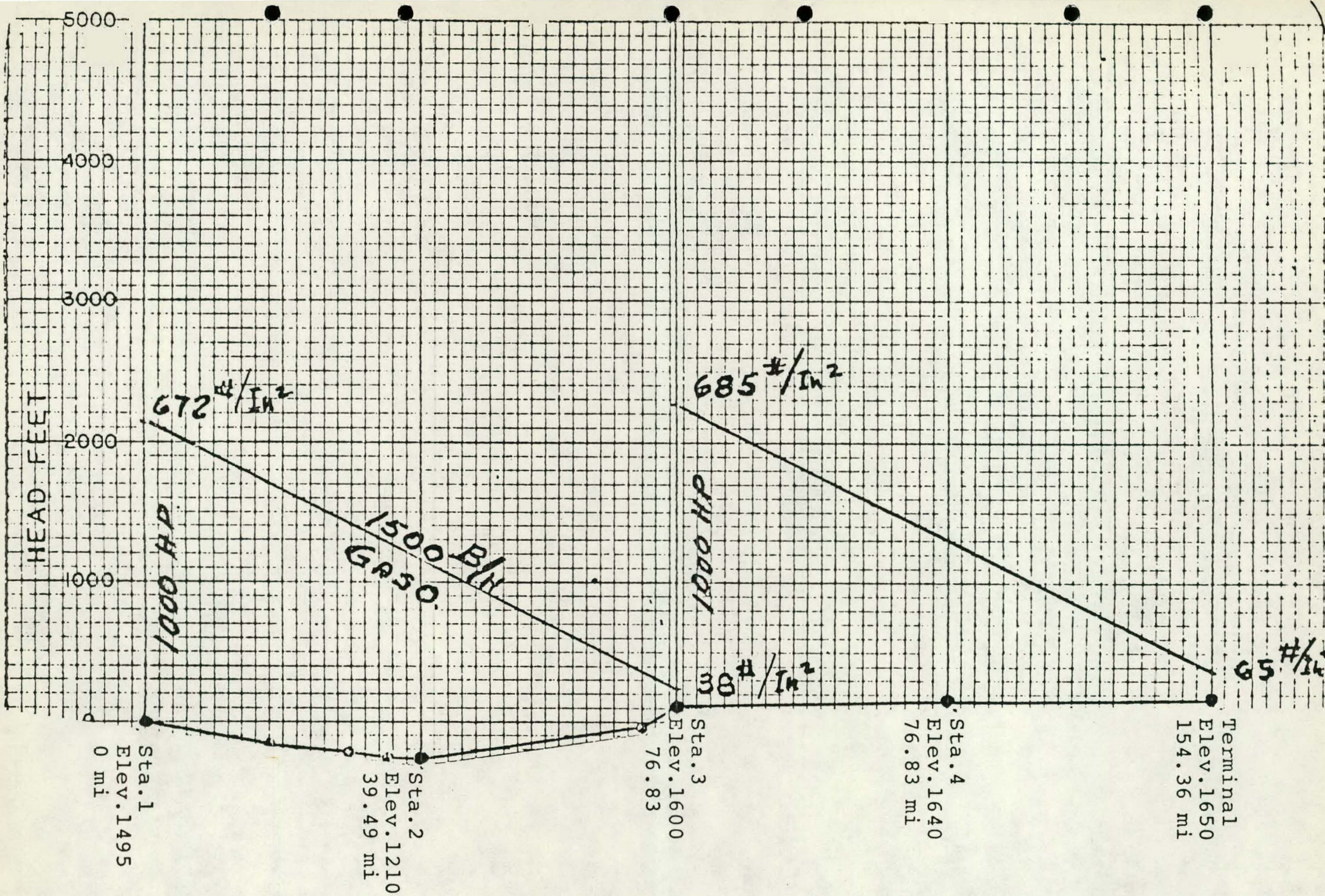

Fig. 4.3.6.1.2-2 - Hydraulic gradient, two pumps, all gasoline

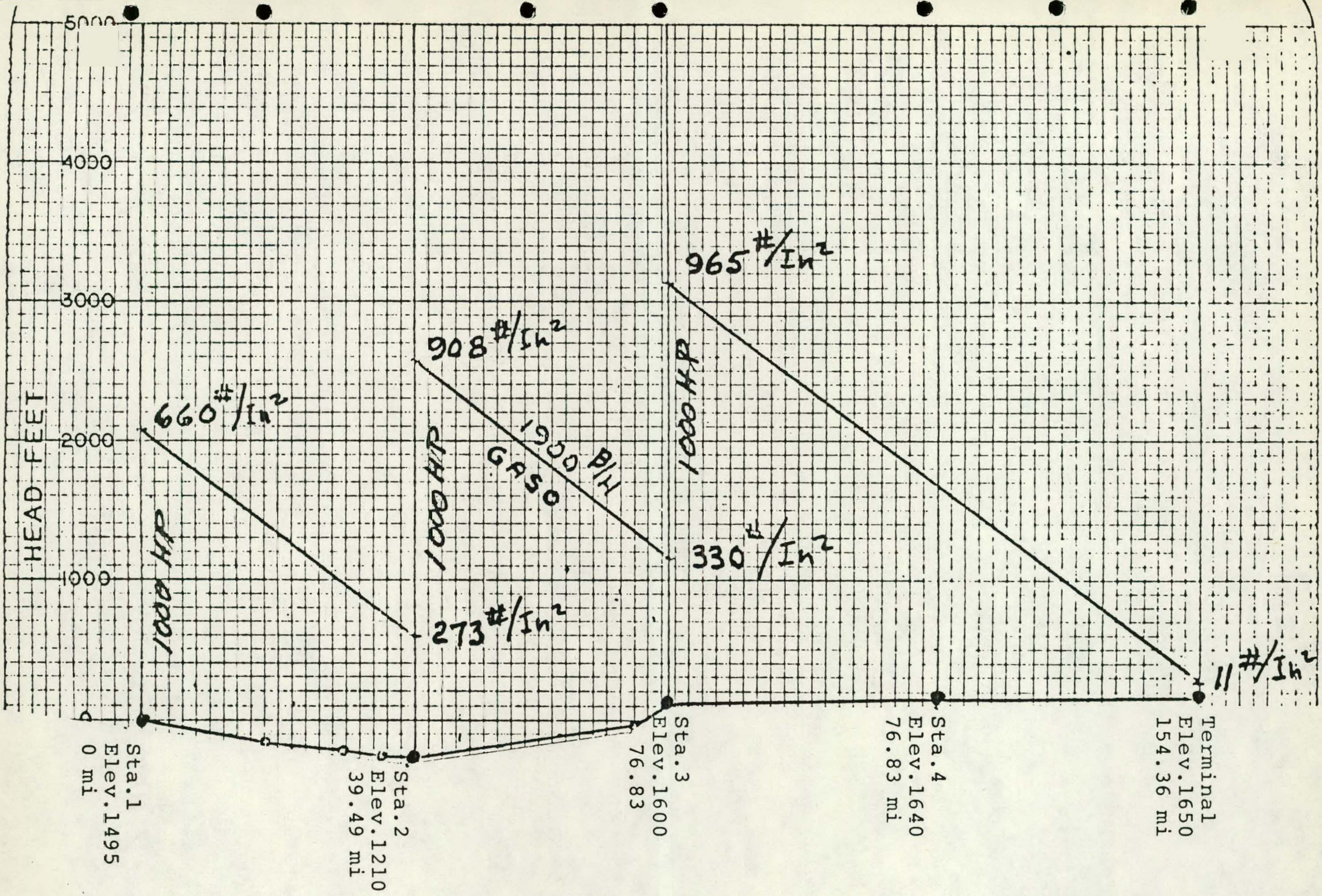

Fig. 4.3.6.1.2-3 - Hydraulic gradient, three pumps, all gasoline

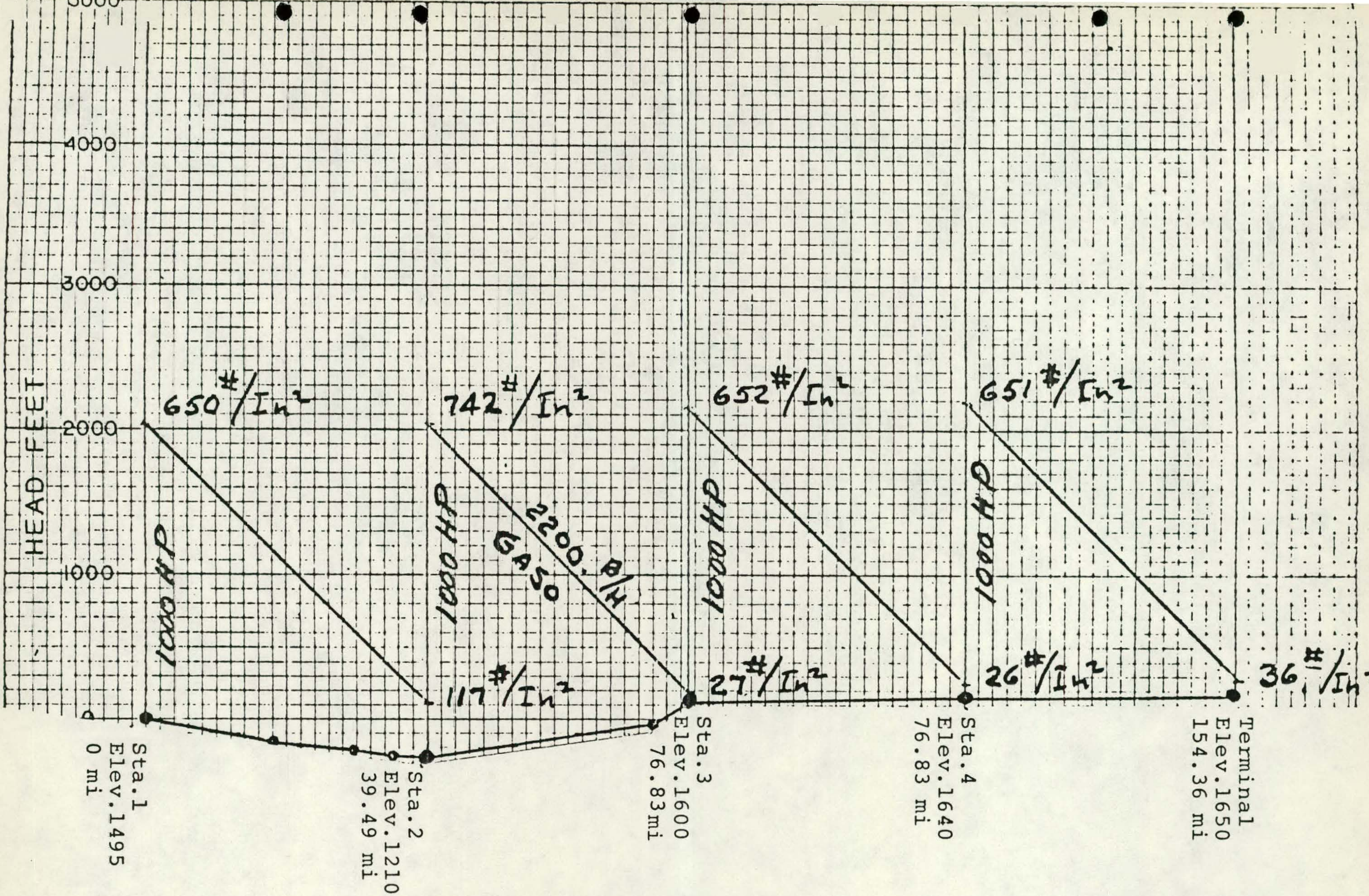

Fig. 4.3.6.1.2-4 - Hydraulic gradient, four pumps, all gasoline

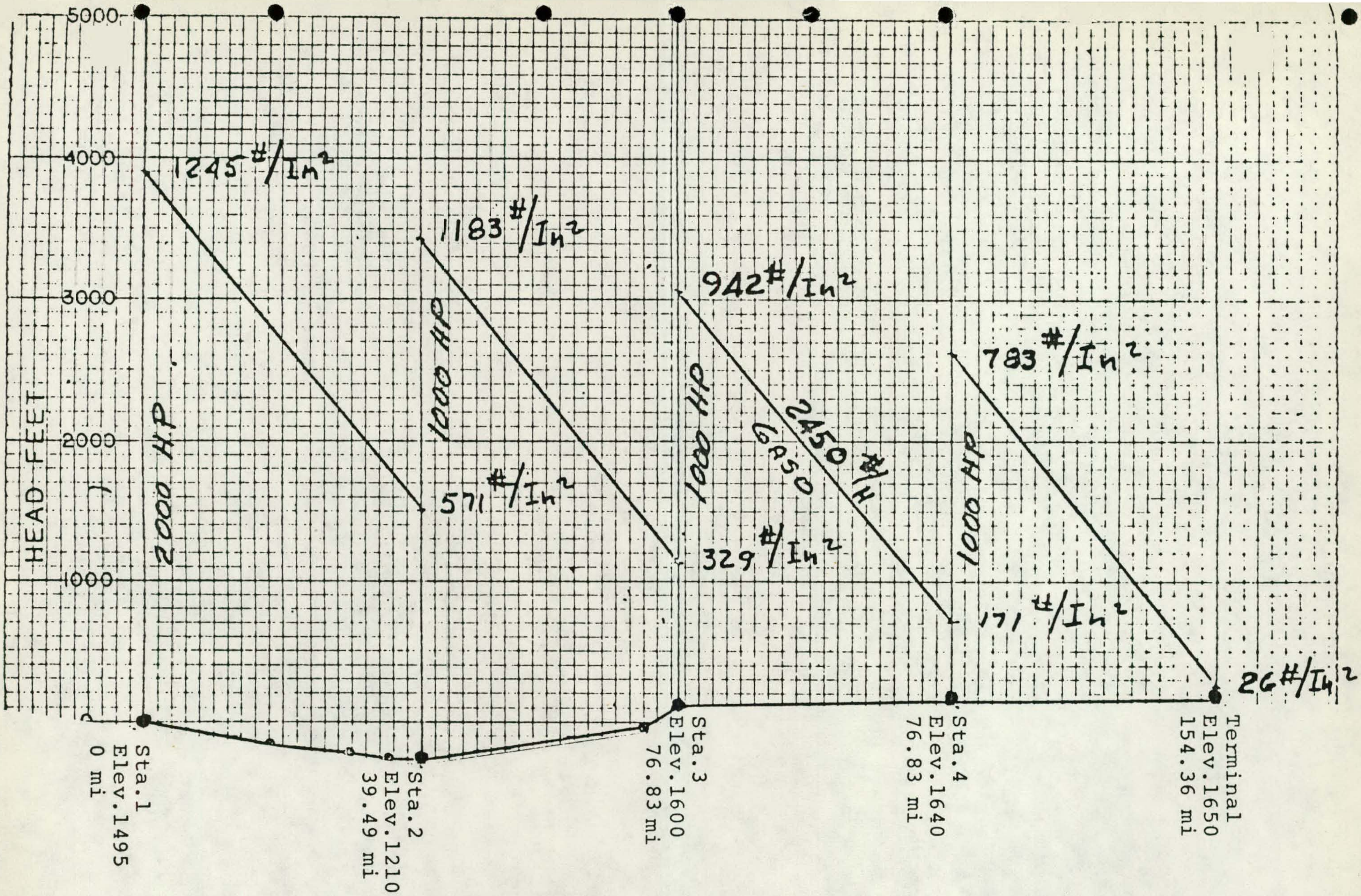

Fig. 4.3.6.1.2-5- Hydraulic gradient, five pumps, all gasoline

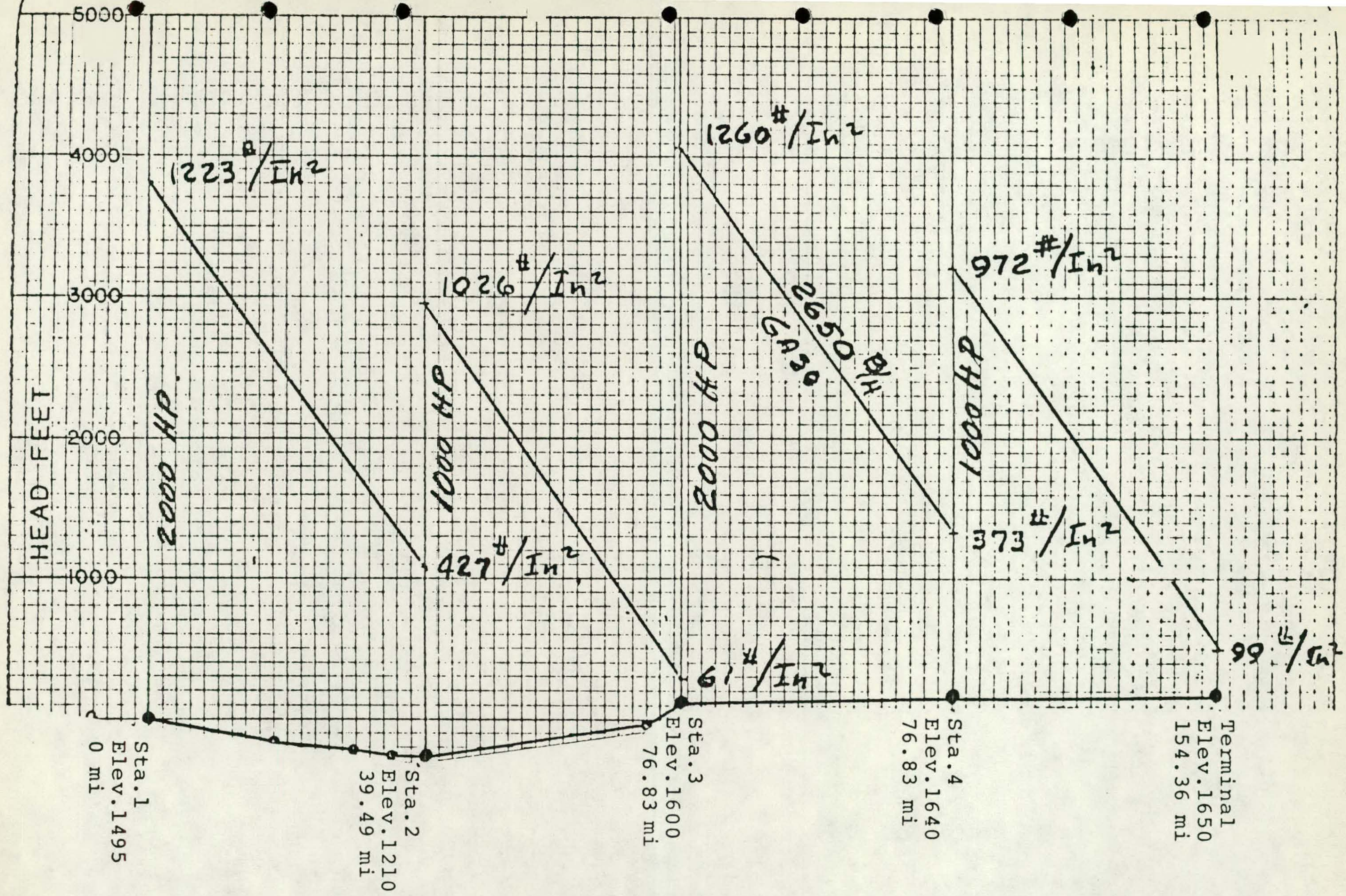

Fig. 4.3.6.1.2-6 - Hydraulic gradient, six pumps, all gasoline

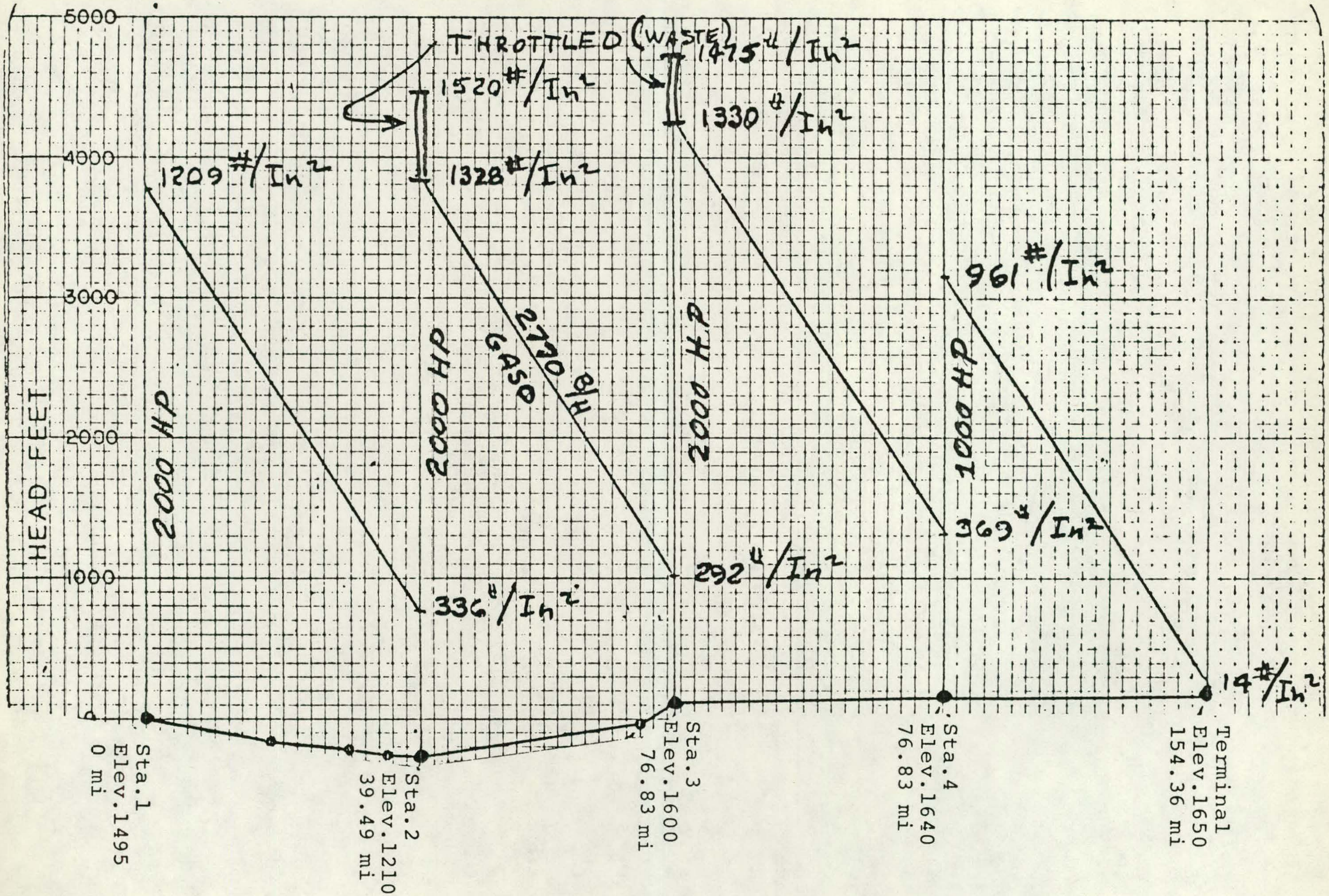

Fig. 4.3.6.1.2-7 - Hydraulic gradient, seven pumps, all gasoline

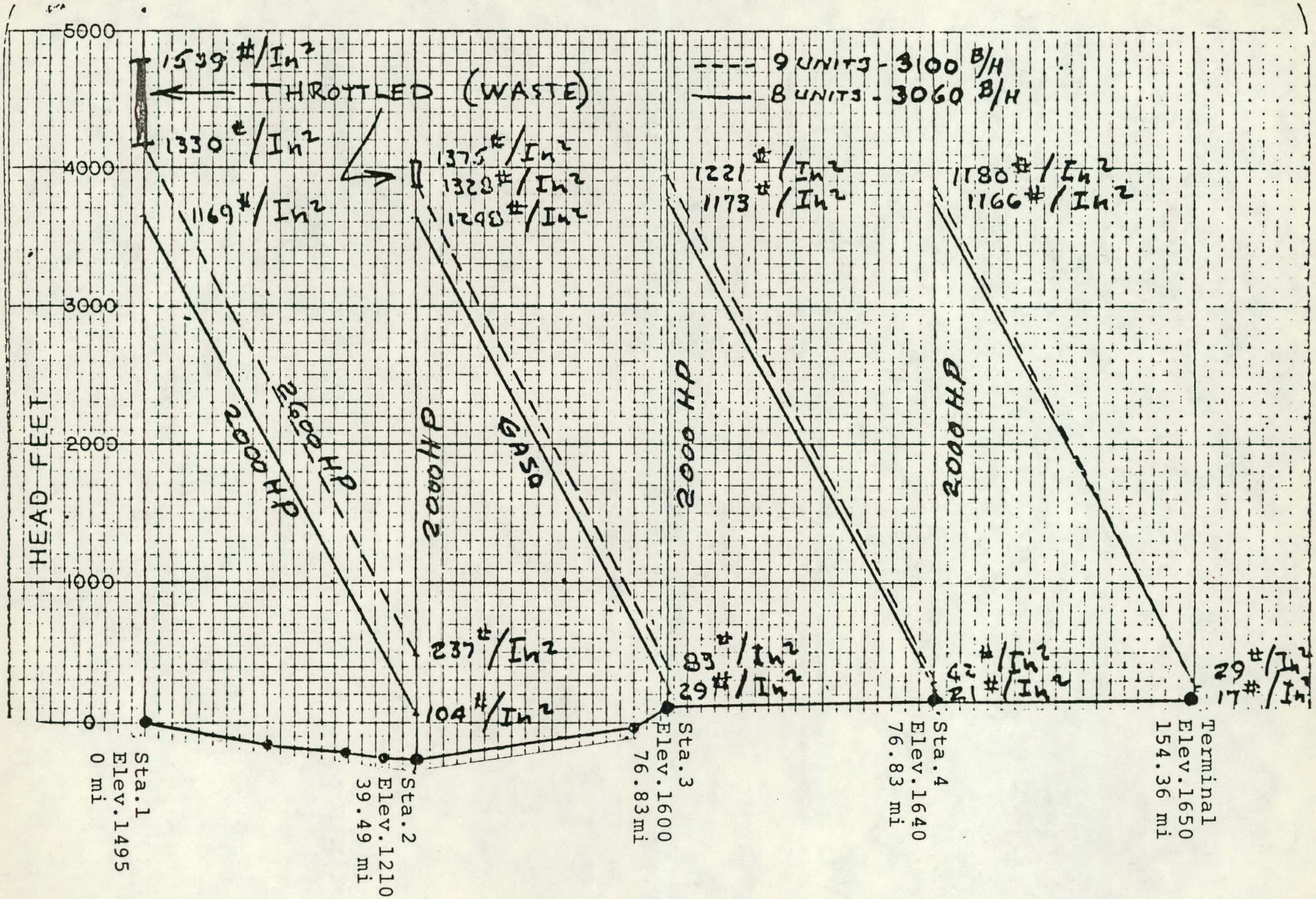

Fig. 4.3.6.1.2-8 - Hydraulic gradient, eight/nine pumps, all gasoline

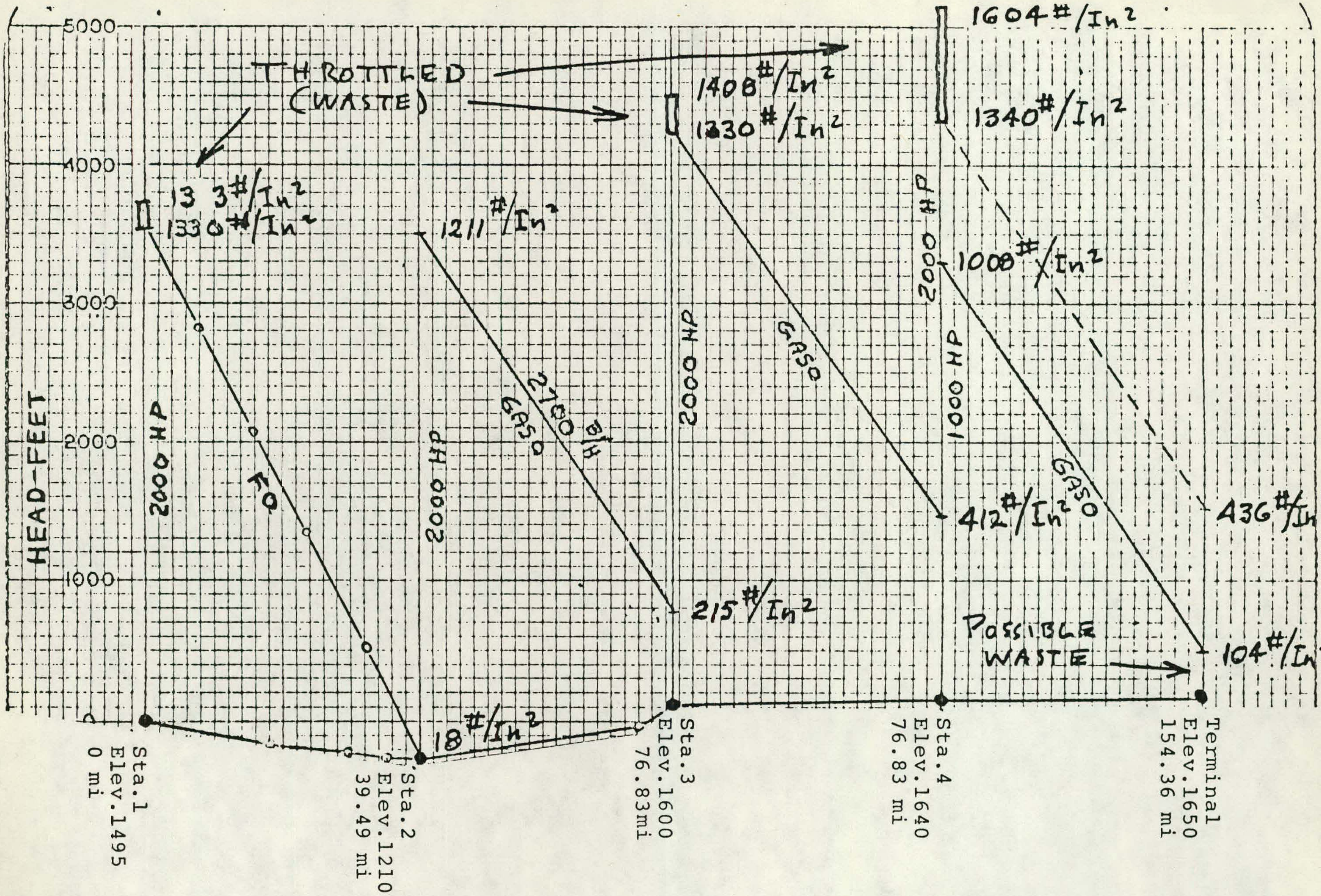

Fig. 4.3.6.1.2-9 - Hydraulic gradient, seven/eight pumps, one segment fuel oil, three segments gasoline

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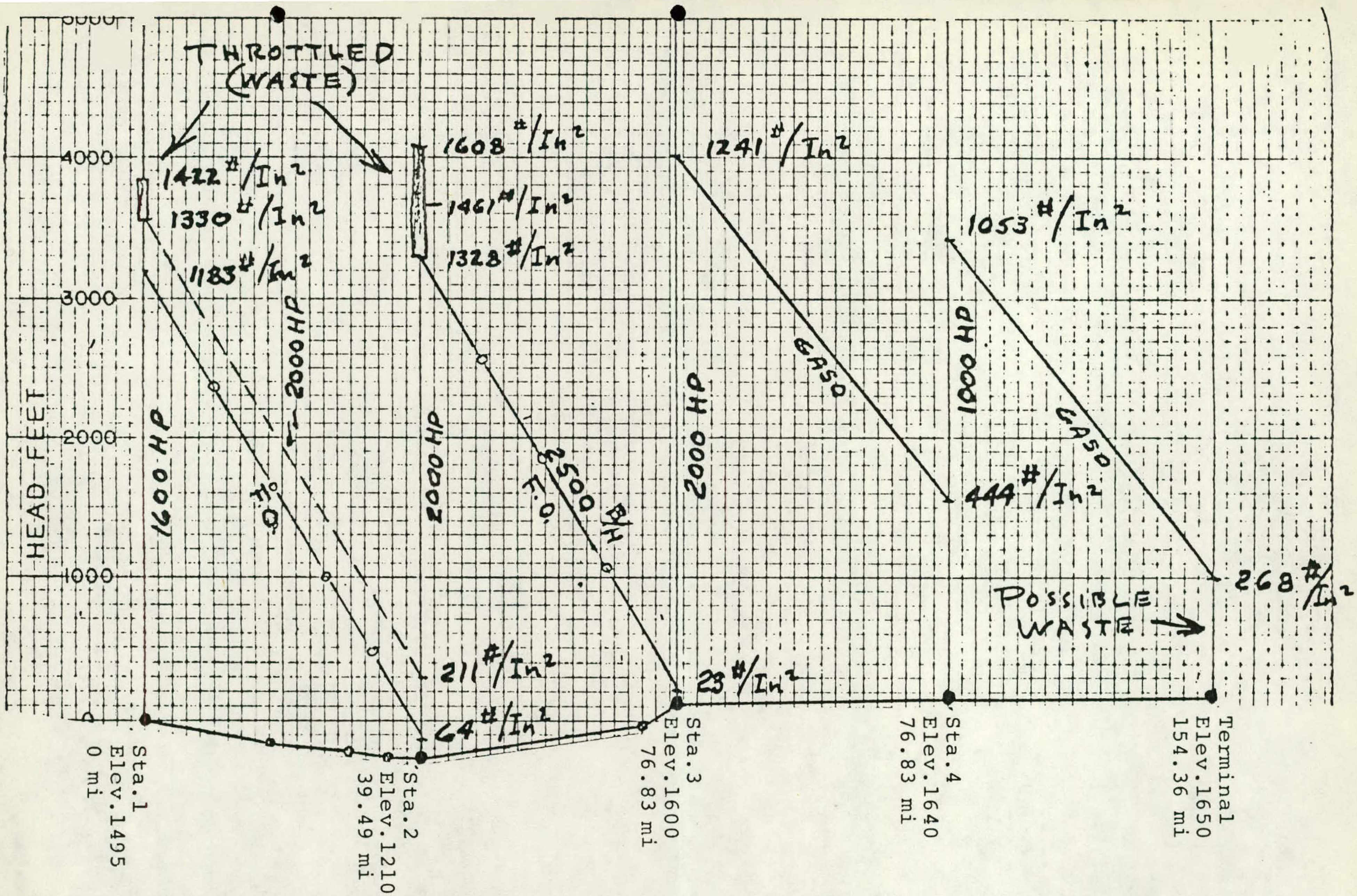

Fig. 4.3.6.1.2-10 - Hydraulic gradient, seven pumps, two segments fuel oil, two segments gasoline

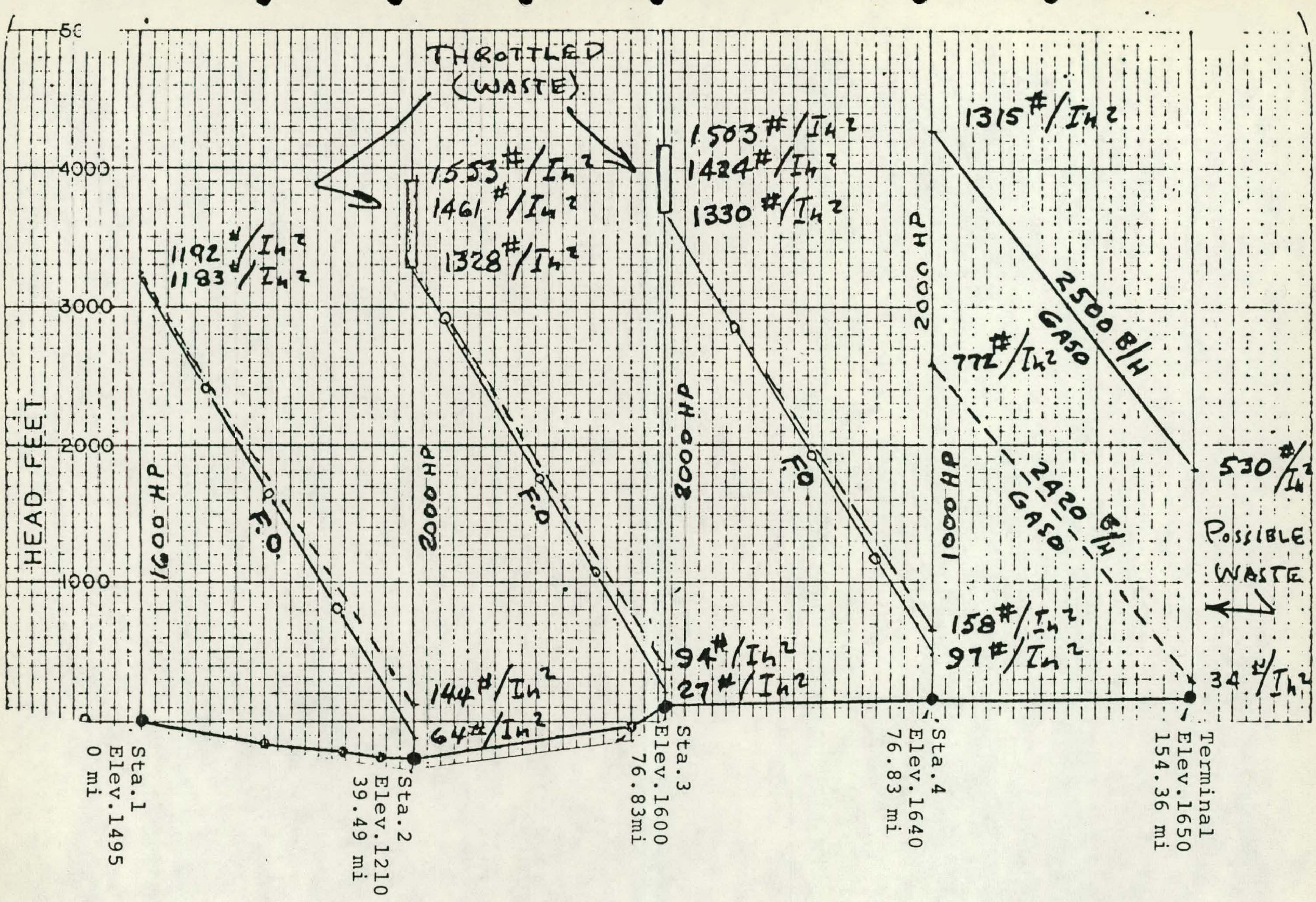

Fig. 4.3.6.1.2-11 - Hydraulic gradient, seven/eight pumps, three segments fuel oil, one segment gasoline

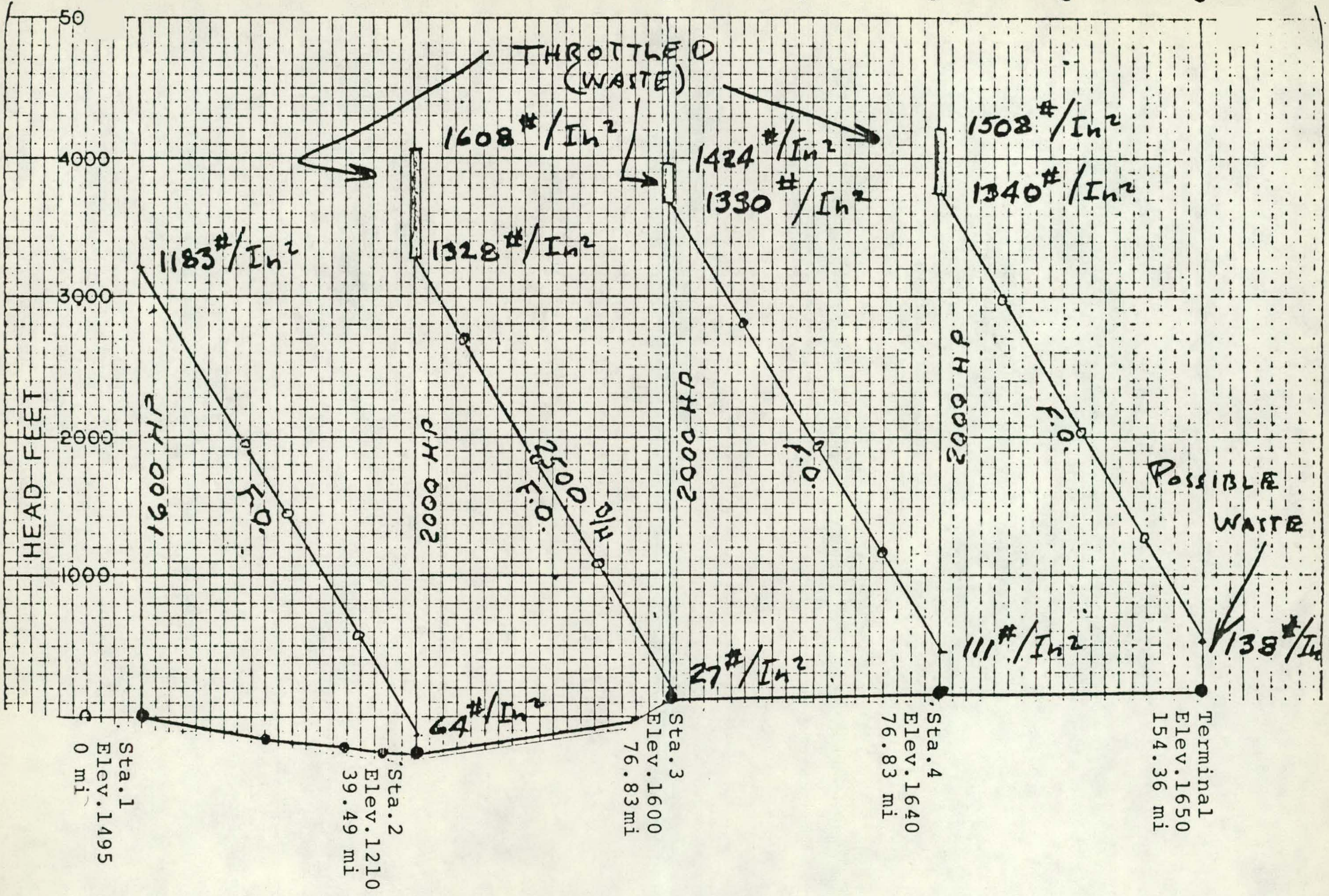

Fig. 4.3.6.1.2-12 - Hydraulic gradient, eight pumps, all fuel oil

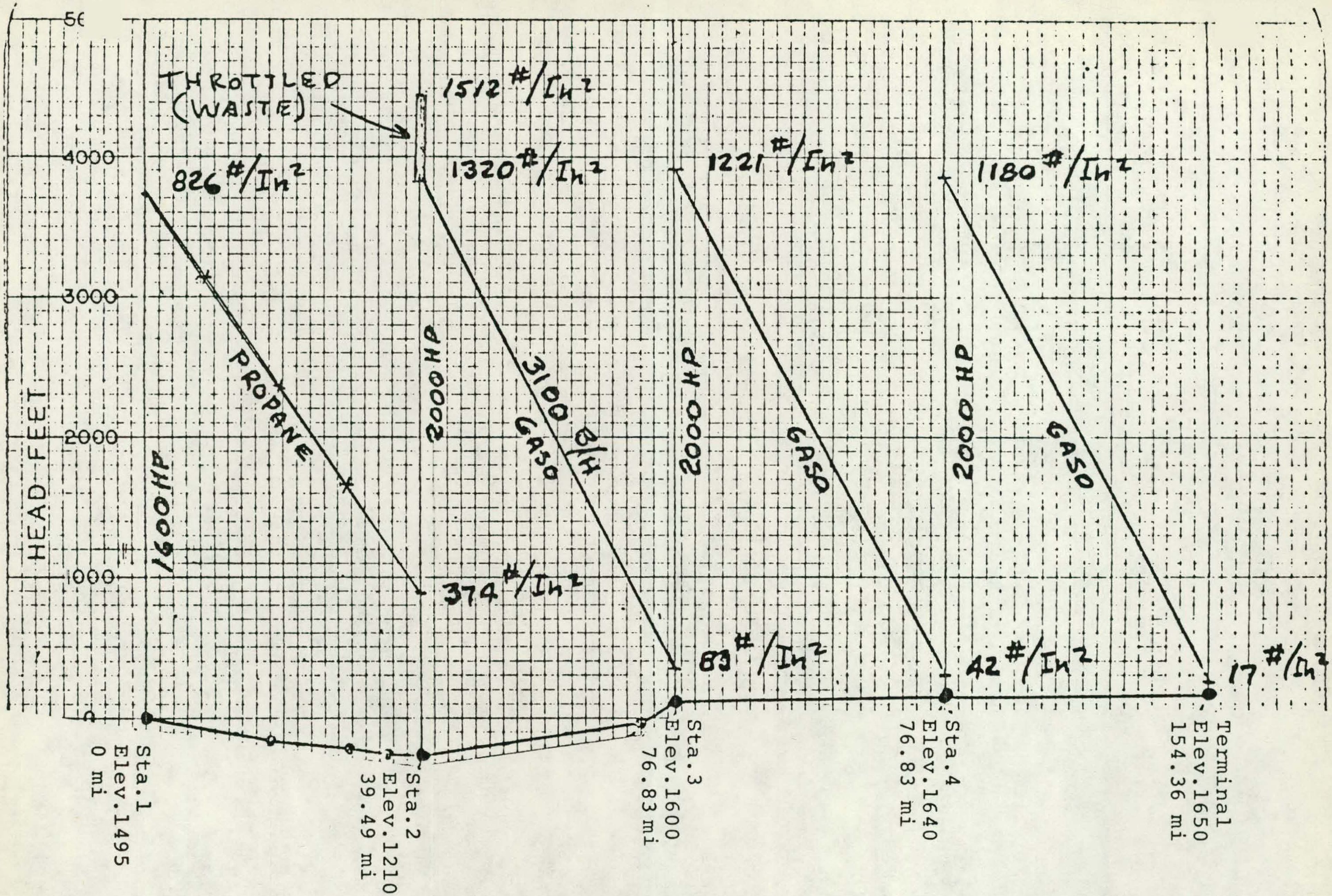

Fig. 4.3.6.1.2-13 - Hydraulic gradient, eight pumps, one segment propane, three segments gasoline

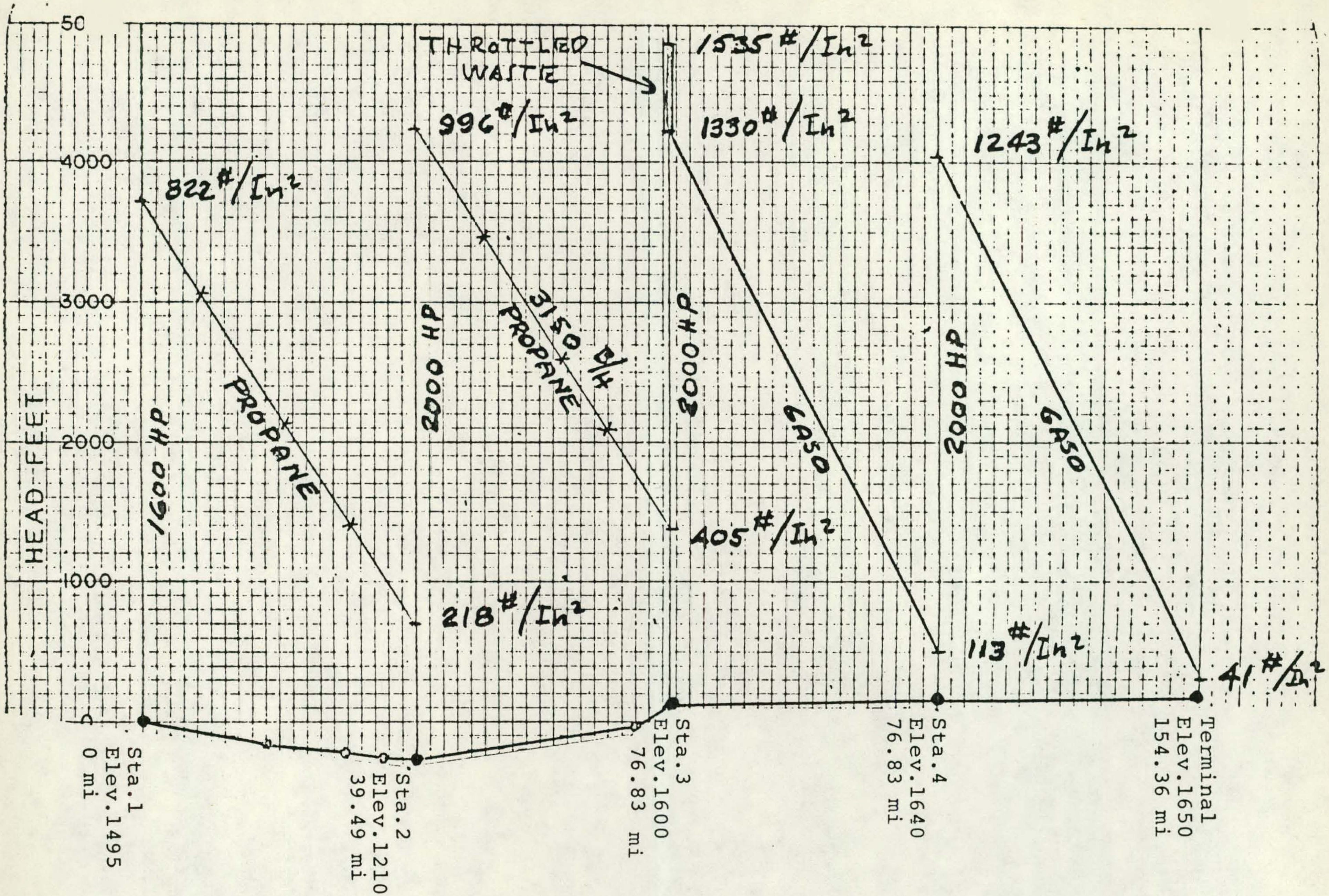

Fig. 4.3.6.1.1-14- Hydraulic gradient, eight pumps, two initial segments propane, two segments gasoline

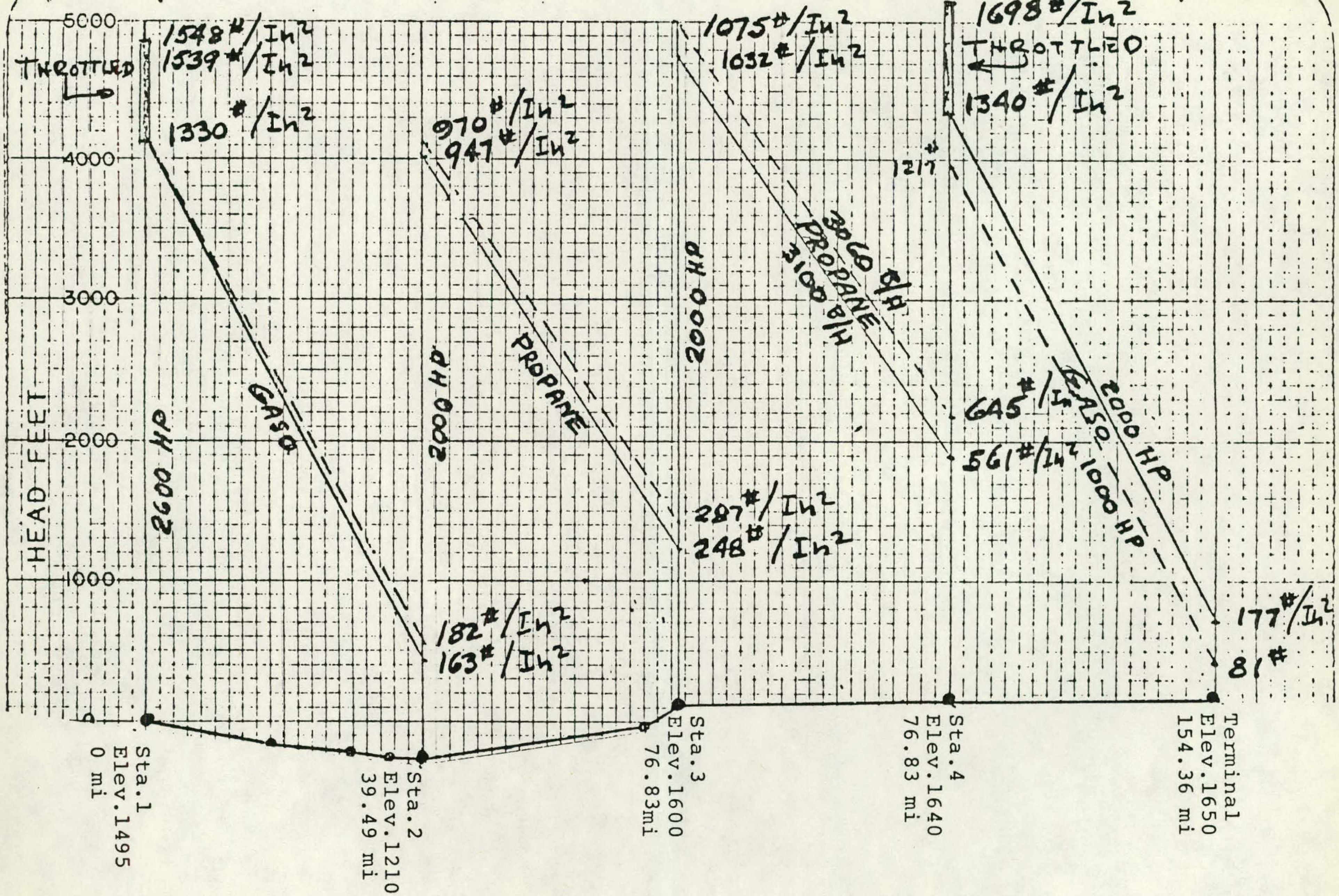

Fig. 4.3.6.1.2-15 - Hydraulic gradient, seven pumps, two initial segments propane, two segments gasoline

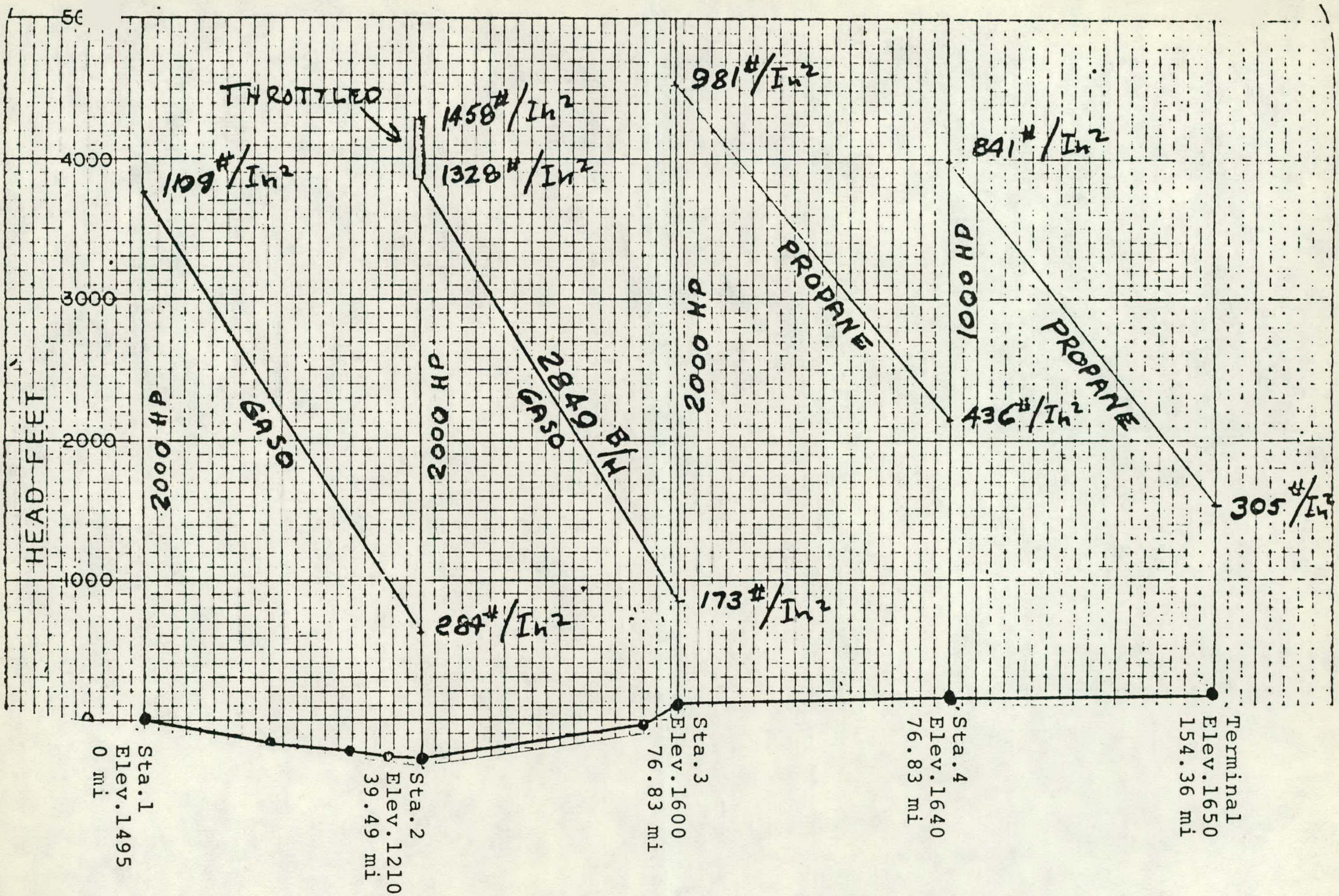

Fig. 4.3.6.1.2-16 - Hydraulic gradient, eight/nine pumps, two initial segments gasoline, two segments propane

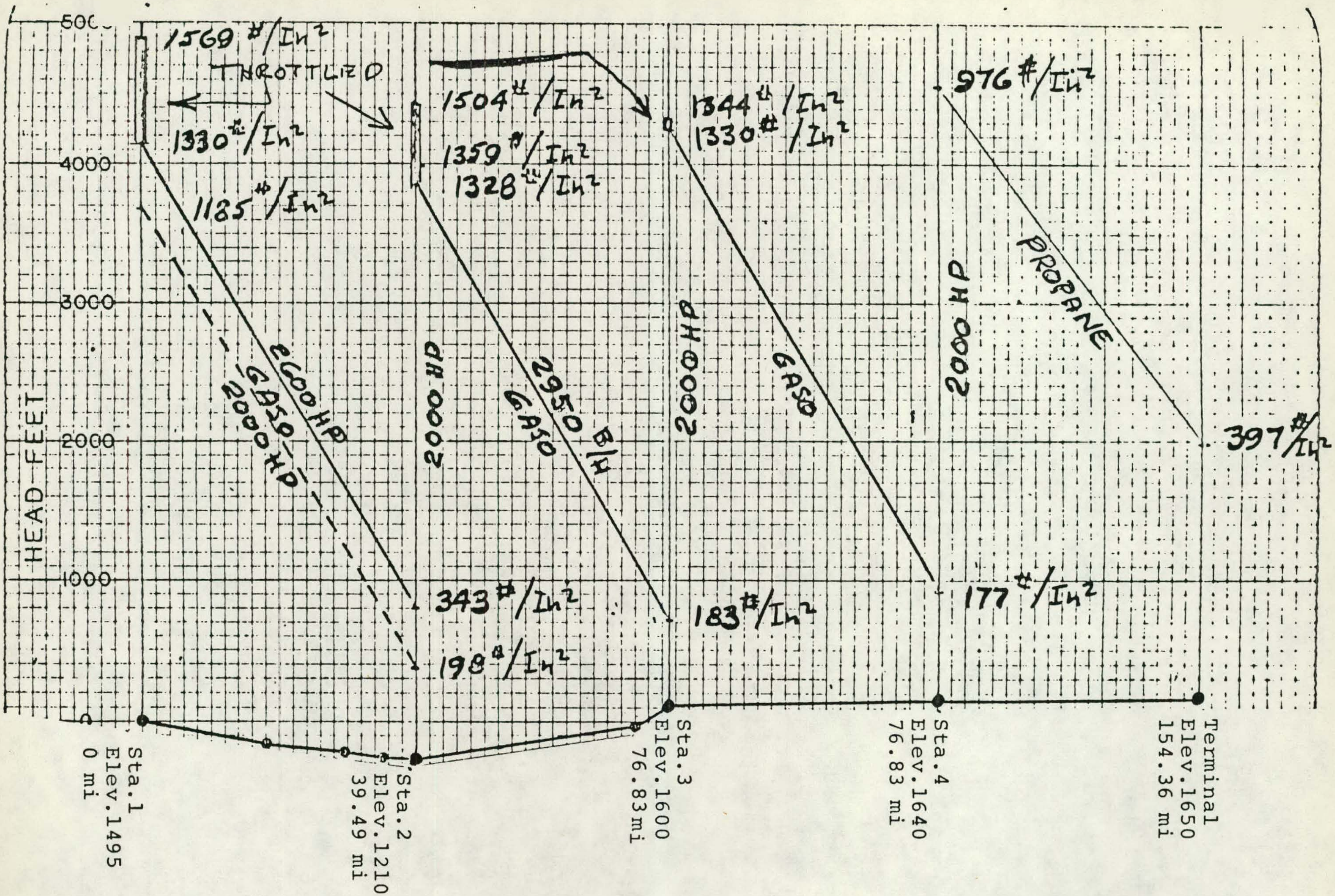

Fig. 4.3.6.1.2-17 - Hydraulic gradient, eight/nine pumps, three initial segments gasoline, last segment propane

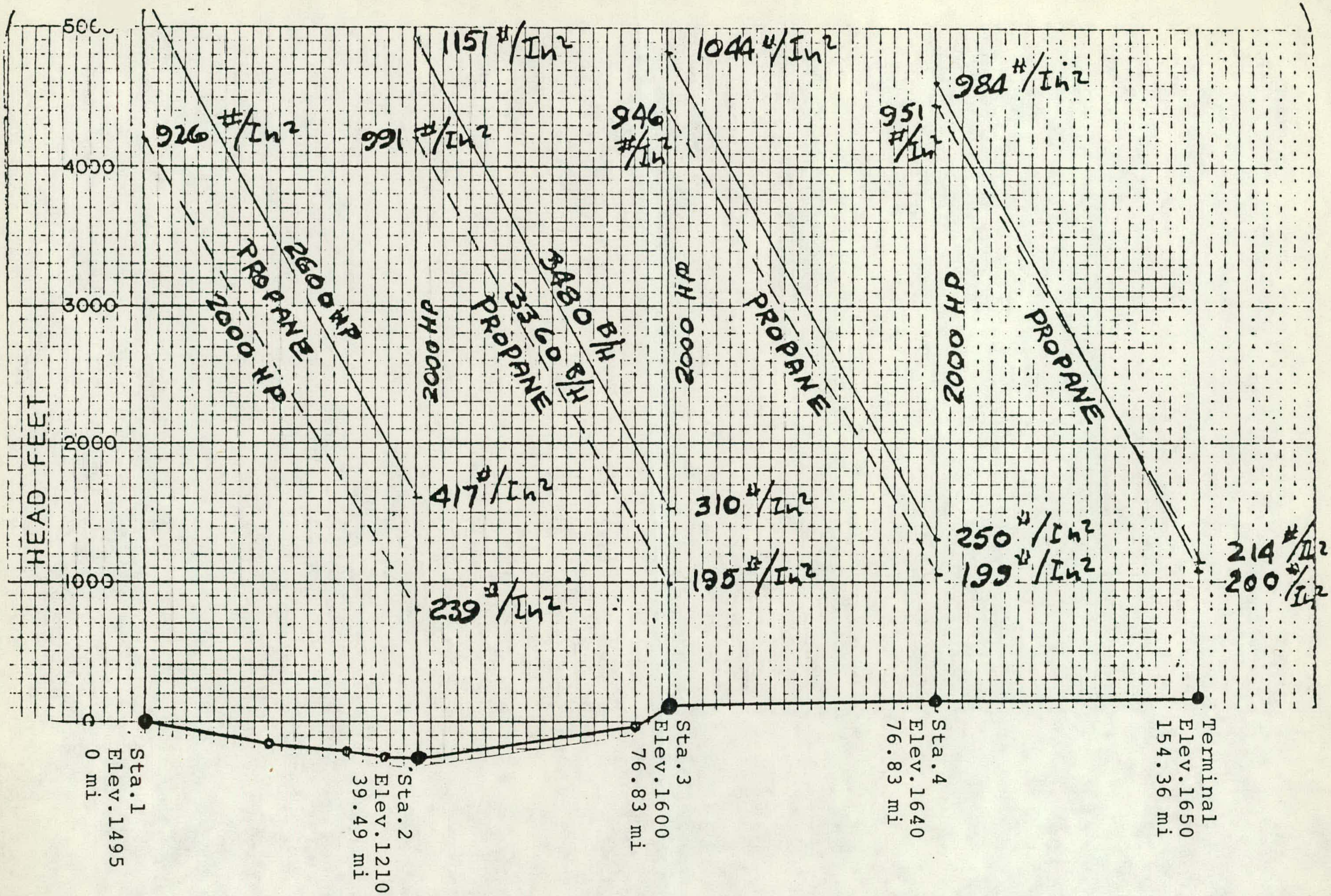

Fig. 4.3.6.1.2-18- Hydraulic gradient, eight/nine pumps, all propane

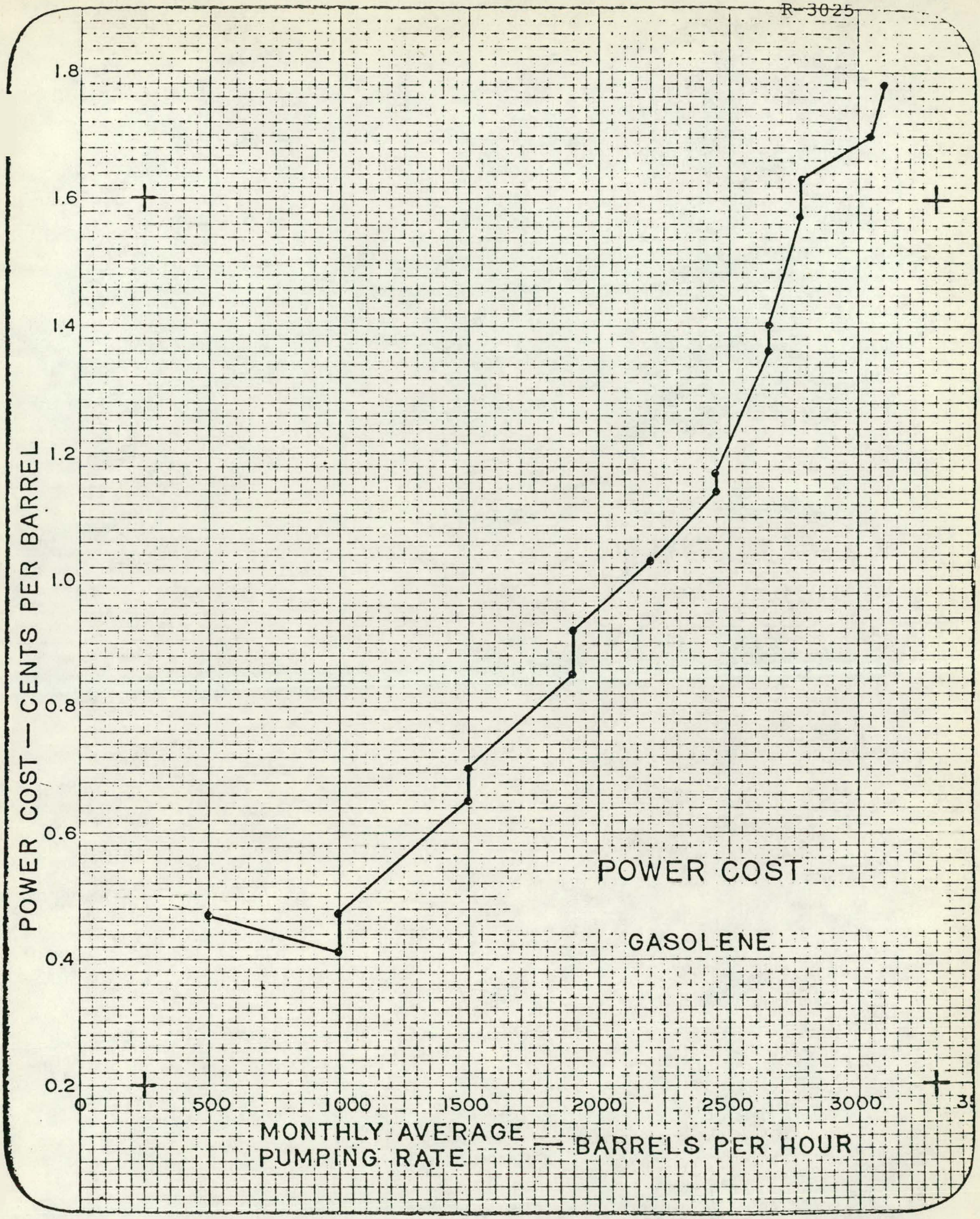

Fig. 4.3.6.1.2-19 - Power cost vs. throughput

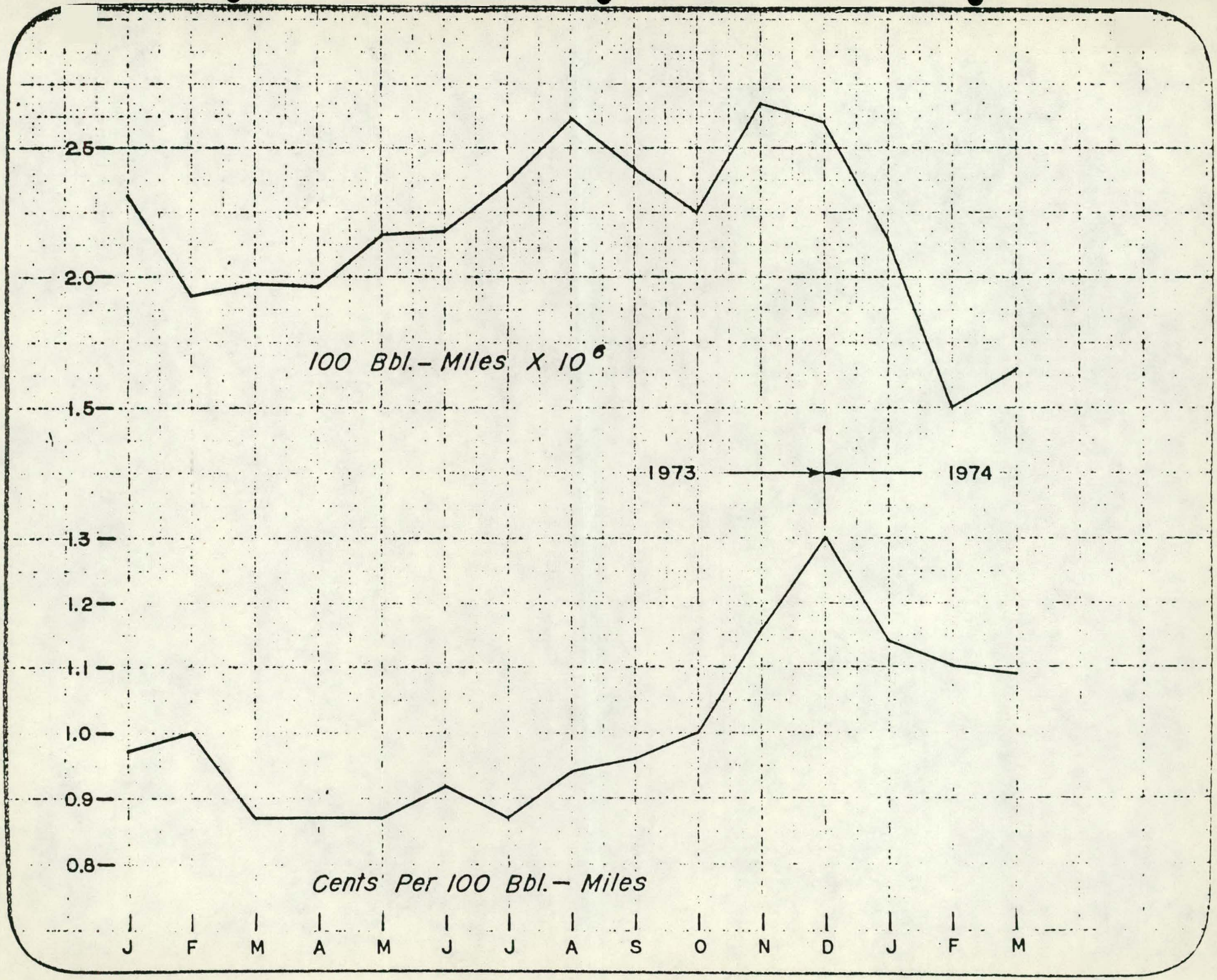

Fig. 4.3.6.1.2-20 - Unit energy cost and total throughput

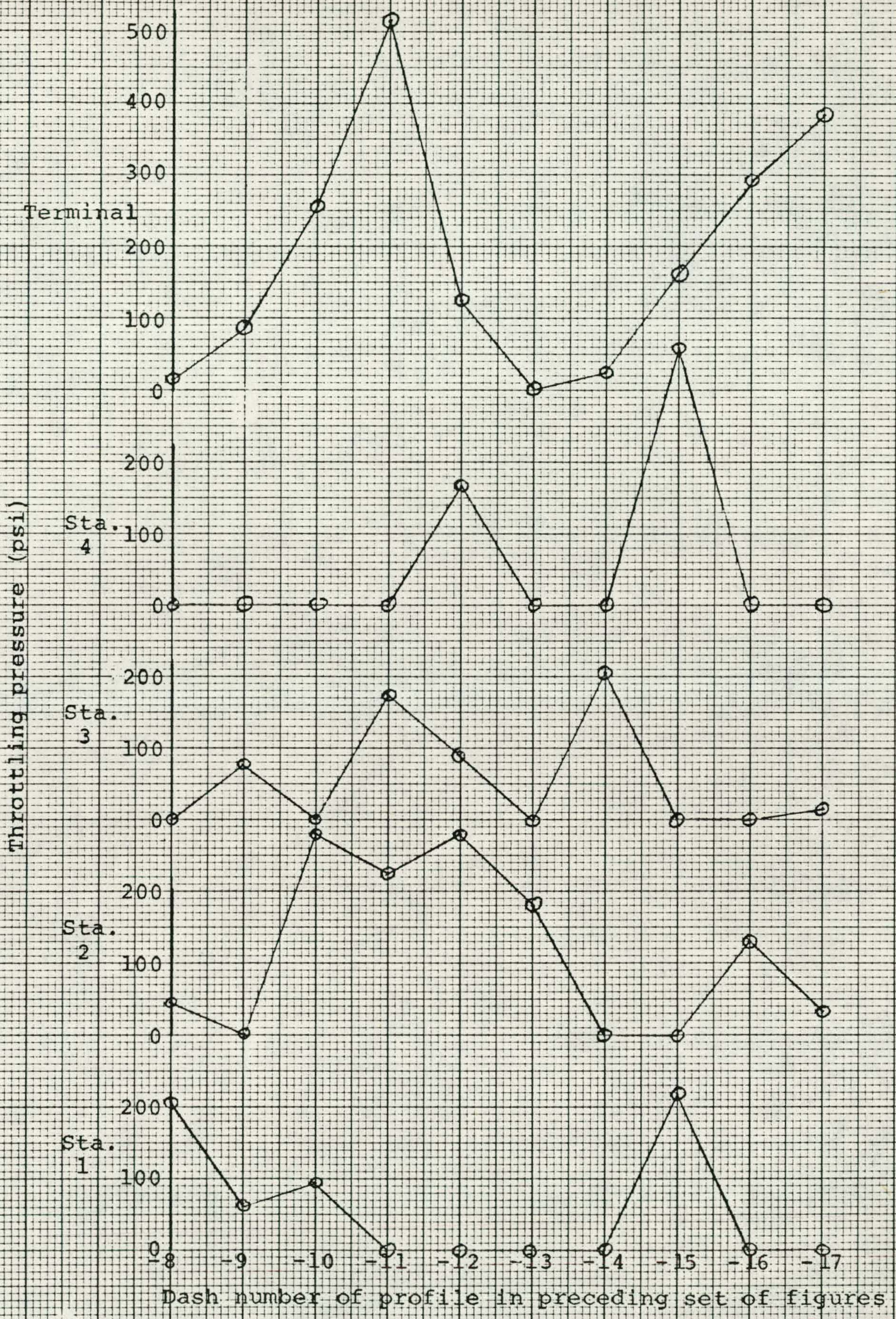

Fig. 4.3.6.1.2-21 - Throttling energy wasted in typical full capacity products pipeline duty cycle

5.0 SLURRY SYSTEM IMPROVEMENTS

5.1 Technological Status of Coal Slurry Pipelines

The technology of coal slurry pipelines is solidly established. Slurry pipelines first gained recognition as a viable method of transporting solids in 1957 when the 108-mile Consolidation Coal pipeline entered service in Ohio, with a capacity of 1.3 million tons/year. This slurry line moved 7 million tons of coal from 1957 to 1963, when it closed down, not because of technical problems, but because of rail rate reductions for large tonnages. The economic and political forces involved have been examined under Task 2 of this project, and are discussed in Section 4.2.1 of Report SSS-77-R-3023 of this series, "Slurry Pipelines - Economic and Political Issues - A Review."

The longest and largest slurry system yet built is the Black Mesa line. The history and description of the line have been presented [Montfort, 1975] and updated [Montfort, 1977], and are summarized below.

Black Mesa Pipeline, Inc., was organized in 1966 to construct, own, and operate a coal slurry line connecting the Black Mesa, Arizona, coal field to the proposed Mohave Generating Station in southern Nevada, over the route shown in Fig. 5.1-1. Black Mesa Pipeline, Inc. is a wholly owned subsidiary of Southern Pacific Pipe Lines, which operates 2400 miles of petroleum products pipelines. These pipeline systems in turn are a part of Southern Pacific Company. This is the longest and largest slurry line yet built. In 1967, a 35-year transportation contract was signed with Peabody Coal Company and engineering design began. Shakedown began in August 1970, and by November, commercial operation was in process.

Black Mesa Pipeline receives coal at a central point in Peabody's strip mine located on the Black Mesa of the Navajo

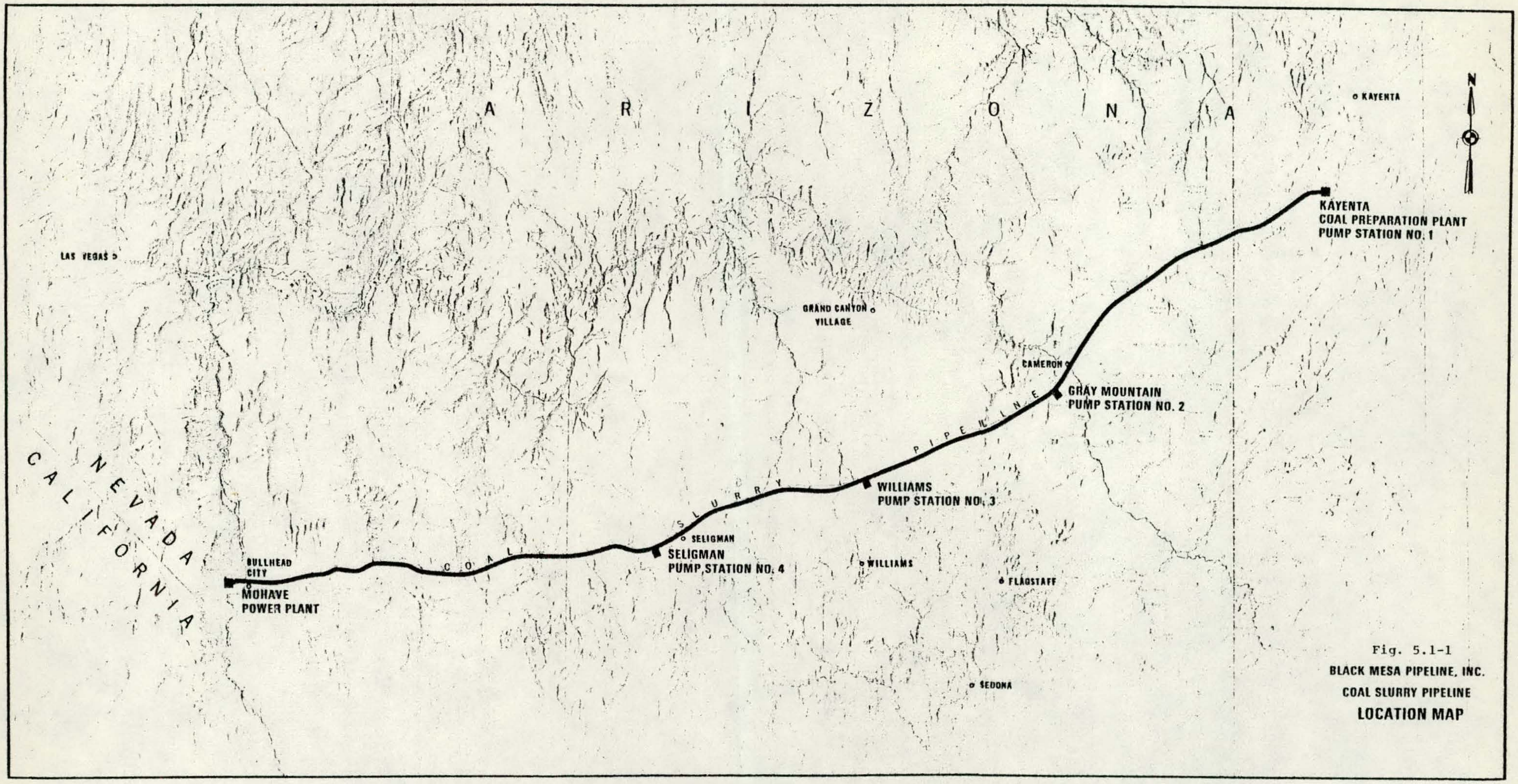

Fig. 5.1-1
BLACK MESA PIPELINE, INC.
COAL SLURRY PIPELINE
LOCATION MAP

and Hopi Reservations in northeastern Arizona. The pipeline system includes slurry preparation plant, pump stations, pipeline, test loops, control and communication facilities, but not terminal storage and dewatering facilities, which are part of the power plant (Fig. 5.1-2). Slurry process water is furnished by the coal company from deep wells. The Black Mesa system cost approximately $39 million.

Coal is received from the coal company at the preparation plant by a conveyor belt, which delivers 2" x 0" coal into three elevated raw coal bins. Each bin feeds a process line consisting of an impact crusher, a rod mill, a sump and centrifugal sump pump, shown in Fig. 5.1-3. Impactors reduce the coal to 1/4" x 0" by dry crushing, and rod mills pulverize the coal by wet grinding to 8 mesh x 0. Slurry is formed in the rod mills,where water is introduced. From rod mill, sump slurry is pumped into one of four 630,000-gallon storage tanks, which are open top and equipped with mechanical agitators to maintain slurry suspension. The slurry is transferred from the storage tanks by a centrifugal charge pump into the suction of the mainline high-pressure pumps.

The pipeline system is capable of pumping 660 tons of coal per hour. At 48% solids by weight, flow is 4200 GPM and velocity is 5.8 fps. When lower delivery rates are required, flow is reduced, down to about 5 fps. Any necessary further reduction is accomplished by inserting water slugs between batches of slurry. Transit time is three days, and line fill includes 45,000 tons of coal.

Four pump stations are required. These are sizeable installations, utilizing the largest piston pumps with electric motor drives and hydraulic couplings for speed control. Three stations have three pumping units installed, with one a spare. The fourth station has four units, with a spare. The three-unit stations operate at about 1000 psi, and the four-

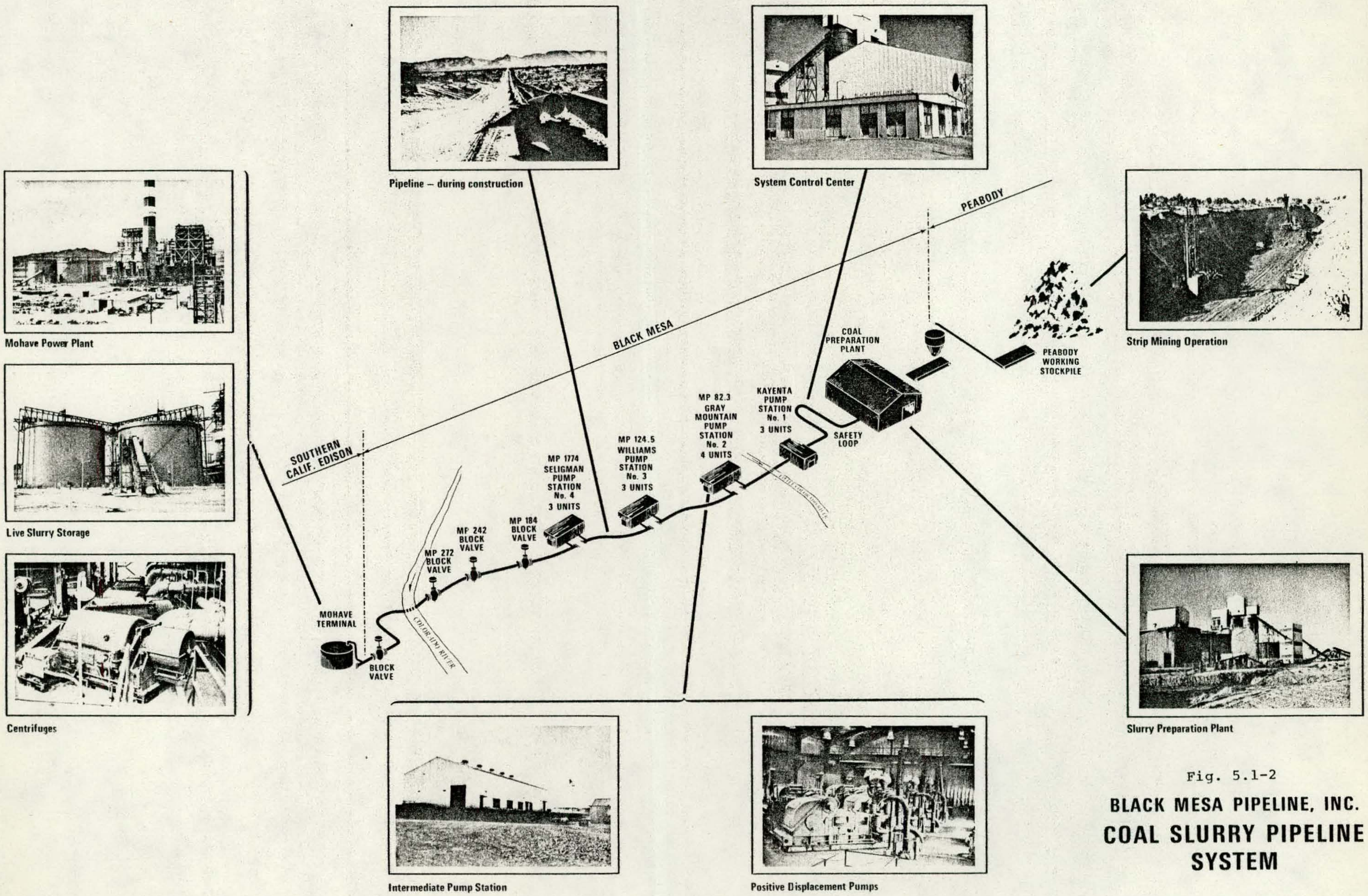

Fig. 5.1-2

BLACK MESA PIPELINE, INC.
COAL SLURRY PIPELINE SYSTEM

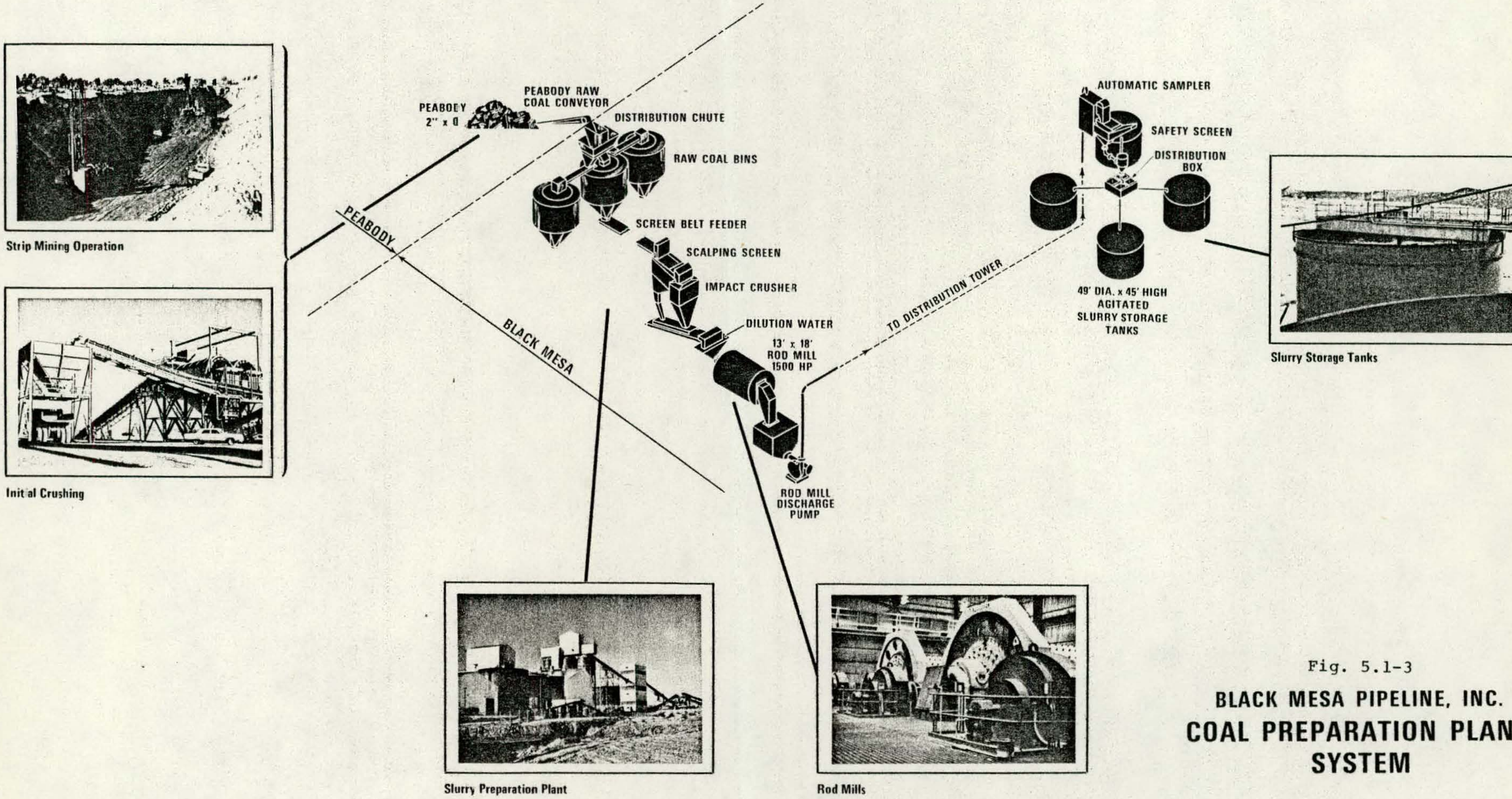

Fig. 5.1-3

BLACK MESA PIPELINE, INC.

COAL PREPARATION PLANT SYSTEM

unit station at about 1500 psi. Driver horsepower is 1500 and 1750 respectively.

The pipe is 18" diameter and traverses mountainous northern Arizona terrain with elevations varying from 6500 ft to 500 ft (Fig. 5.1-1). The last 12 miles drop 3000 ft and in this stretch, diameter is reduced to 12". The pipe is a high-pressure steel, welded pipeline and is conventional except that a 16% maximum slope limitation was observed for hydraulic reasons. The line is extensively telescoped with wall weights ranging from .469" to .219". Even though most of the ditch was rocky, a tape coating system was successfully used, with extensive dirt padding.

At the time of construction, it was not known whether the line could be restarted after an extended shutdown. Accordingly, ponds were provided into which the contents could be dumped in case of shutdown.

Early in 1971 the pipeline was shut down twice due to power failures. Particle size gradation (consist) was relatively coarse and difficulties were encountered in restarting sections of the line. On each shutdown, one hard plug was encountered and had to be located and removed. Plugs were specific - about 40-ft in length. To be on the safe side, other sections were restarted and line fills were dumped into station ponds. In the first instance the affected line section was down 4-1/2 days, but subsequent to removing the plug the section was restarted without difficulty. The consist was adjusted to provide a finer grind and since then, numerous restarts have been handled without difficulty. It is not expected that extensive dumping will occur again. The two important keys to a restartable slurry are a fine grind and a well-graded consist.

Pump stations and pipeline operation are supervised from origin point. Stations, which are complex, are operated unattended; however, there are two resident employees at each booster station to perform maintenance and to provide on-call availability. Pumping is "tight-lined," i.e., without surge tankage at booster stations. Locations are linked by a solid state, hot, standby, microwave system operating at ~2100 MHz. One link of 109 miles involves use of space diversity antennas to cope with atmospheric fades.

The slurry is delivered at the power plant into large (7,800,000 gallon) storage tanks. The tanks, with 500-hp agitators, though much larger than any such tanks previously built, and have operated satisfactorily. Slurry is withdrawn from tankage by a centrifugal pump and transferred to a battery of 40 centrifuges, where 75% of the water is removed. The resulting wet cake is conveyed to 20 bowl mills for drying and further pulverizing. Dried coal is carried pneumatically to the combustion chambers.

Fifty-two permanent employees are required to operate the pipeline. Of these, about 36 are at the Black Mesa preparation plant. The remainder are technical and administrative personnel located in Flagstaff and at the booster stations. At Black Mesa, Navajo and Hopi Indians are given employment preference under terms of the mining lease. At both Peabody and Black Mesa Pipeline, Indian personnel are in the majority.

The system is operated by a four-man shift at Black Mesa. The group includes a Shift Supervisor, a Pipeline Operator in the System Control Room, an Operator-Repairman handling preparation plant and other facilities, and an Operator-Repairman Assistant primarily doing slurry quality control testing in the Testing Laboratory.

It has been noted earlier that pipelining solids requires considerable testing to determine minimum operating velocities with respect to deposition or bedding and to assure good restart characteristics. Varying the percent solids and the size consist affects these important operating characteristics. High capacity test loops are useful here, and Black Mesa has several well instrumented loops, including one 1000' long and 24" in diameter. After determining the desired specification, it is necessary to monitor carefully to assure that specification is consistently met. Percent solids or density is extremely important and is monitored by a nuclear density meter mounted outside the pipe. Size consist must be laboriously obtained by sieve analysis run every hour or two.

The slurry is sometimes corrosive and requires some inhibitor. Water batches especially require use of an inhibitor. Careful corrosion monitoring is necessary.

While the pipeline can be shut down and slurry allowed to settle, pump cylinders and valve chambers as well as convolved station piping must be promptly flushed upon shutdown. The slurry tends to pack into dead spots except for those on top of the pipe, which affects valve selection. A conventional gate valve will not function well in slurry service. Ball valves and lubricated plug valves seem best suited for slurry.

Wear is a continuing process in the mainline pumps. It is of course to be expected, and is therefore not a problem. Valves, seats, pistons, rods, and packing are expendable. However, these parts have met and, in most cases, exceeded expected service life.

The system has successfully completed more than six years of operation. While the line is capable of transporting over 5 million tons of coal per year, it has actually averaged only about 4 million tons per year because of lesser burn requirements at the power plant. Pipeline availability has exceeded 99%. The plant engineer at the Mohave Generating Station reports that, of the three major subsystems, coal mine, pipeline, and power plant, virtually all system downtime is due to either the mine or the power station. The pipeline "is always running."

Because minimum operating velocity and design throughput velocity are fairly close together, the pipeline has a relatively small turndown. When the power plant requirement is less than pipeline turndown, batching is necessary and a dispatching model has been just as useful in this application as on a petroleum line. Flow is measured in gallons per minute using magnetic flow meters and by timed stroke counts. Gallons per minute and tons per hour are related by specific gravity of solids and percent solids.

In closing, it is well to mention planned future systems. The line that is in the most advanced planning stage appears to be the 1036-mile pipeline of Energy Transportation Systems, Inc. (ETSI), owned by Bechtel, Lehman Bros., and the Kansas Nebraska Natural Gas Co. The proposal to build this billion-dollar line has evoked strong opposition (see Report SSS-77-3023 under this contract). The issues may not be fully resolved until the results of a coal slurry pipeline study, approved by the Congress Office of Technology Assessment in July 1976, are available. The study, scheduled for completion in mid-1977, will cover technological, energy, and legal issues; environmental effects (especially water and land use); evaluation of costs and returns; and impact on the railroads.

5.2 Methanol-Coal Slurries

A promising new concept in coal slurry technology which appears to have potential for overcoming some of the problems and limitations associated with coal-water slurries is the methanol-coal slurry system. Leonard Keller, President of the Methacoal Corporation, Dallas, Texas, holds patents, including Ref. 5.1C, relating to the methanol-coal slurry, which he calls Methacoal. He describes Methacoal as a pseudo-thixotropic, or shear-thinning, mechanically stabilized suspensoid. At rest it appears to be a moist, solid mass of black mud, but when subjected to stirring or agitation, it becomes thinner and flows easily. The flow characteristics are such that it is typified viscous or laminar flow, rather than the turbulent flow characteristics typical of coal-water slurries.

5.2.1 Methanol Carrying Capacity

5.2.1.1 Experimental Measurements of Carrying Capacity

It appears that methanol has somewhat greater carrying capacity than water. To acquire some first-order verification of this hypothesis, as a part of this project rheology tests were conducted under subcontract by Prof. R. R. Faddick of the Colorado School of Mines. His report is condensed and summarized below.

In conjunction with rheological tests, additional bench tests are necessary to determine solids specific gravity, solids screen analysis, density and viscosity of the liquid carrier and, where coal is concerned, a proximate analysis. A Western coal was selected for which all this information was readily available. A relatively fresh, unused sample of Utah coal was available in the Rheology Lab from a previous study.

Three concentrations of coal, from maximum possible down to -5%, -10% from maximum were studied at room temperature. Commercially available methanol was used with one particle size distribution of coal.

Rheology is the science of the flow deformation of liquidlike substances subjected to shearing stresses. These stresses are measured over a range of shearing rates. The resulting relationship of shear stress vs. shear rate depicts the flow deformation characteristics of the slurry for a specific solids concentration, size distribution, and temperature. These data, when plotted, form the slurry rheogram. The slope of the rheogram is called the dynamic, or absolute, viscosity of the slurry.

Newtonian liquids, such as water and most hydrocarbons, bear a linear shear stress-shear rate relationship, the slope of which is a constant. The deviation of the slurry rheograms from linearity is a measure of their non-Newtonian characteristics or variation in viscosity with shearing rate. Translated to pipeline flow, this means that the viscosity of the slurry varies with the flow and is not a constant as it would be for a Newtonian liquid.

The purpose of the rheology meausrements, then, is to determine a slurry's shear stress-shear rate relationship. A mathematical model of the shear stress-shear rate curve (rheogram) is then defined, and this model is used to predict slurry friction head losses for pipe flow. The model may also be used to scale actual, measured, pilot loop friction losses to pipe sizes other than that used in the loop.

The rheological measurements were made with a Brookfield Model RVF viscometer over a range of slurry concentrations and at a temperature of 25°C. The slurries were placed in a blender jar containing a heating coil which was connected to a constant temperature bath. The viscometer spindle was lowered into the slurry, which was then agitated as necessary to suspend the solids. With the blender off and the slurry in a quiescent state, the shear stress-shear rate relationships were obtained. Measurements were made with both ascending and descending spindle movements, and all measurements were repeated four to eight times. Yield stresses were obtained by recording the static torque reading when the spindle was allowed to come to rest in the slurry.

A three-parameter yield-pseudoplastic model was fitted to the data. This model does not fit all of the rheology data as well as a four- or five-parameter model would, but it is much easier to deal with mathematically. The model was adjusted to fit the rheology data over the range of shear rates to be encountered in the prototype pipeline operation.

The model selected was a yield-pseudoplastic having the form

$$\tau - \tau_y = K\dot{\gamma}^n$$

where

τ = shear stress (dyne/cm^2)

τ_y = yield shear stress (dyne/cm^2 (TAUY)*)

K = flow consistency index (dyne-sec^n/cm^2 (KYP)*)

$\dot{\gamma}$ = shearing rate (sec^{-1})

n = flow behavior index (dimensionless (NYP)*)

(*Computer notation, used in Fig. 5.2.1.1-1)

Utah Carbon King coal from Braztah Corp. in Helper, Utah, was crushed in a laboratory-type hammermill (Holmes) in several passes.

Specific gravity (measured) = 1.4294 @ 25°C

Moisture content - 1.16%

PROGRAM

THIS PROGRAM SUMMARIZES AND ANALYZES RHEOLOGY DATA FOR SLURRIES OR LIQUIDS MEASURED
WITH A BROOKFIELD VISCOMETER (MODEL LVT OR RVF) USING CYLINDRICAL SPINDLES
SUBROUTINE OPMR PERFORMS A LINEAR REGRESSION ON TAUBOB = K*OMEGA**N
SUBROUTINE PIPE PREDICTS PIPELINE ENERGY REQUIREMENTS

NOTE: PROGRAM COMPUTES IN SI UNITS. SOME DATA OUTPUT IS IN ENGLISH UNITS.

NOMENCLATURE

```
RC      = RADIUS OF CONTAINER MEASURED AT SPINDLE LOCATION, CM
RS      = RADIUS OF SPINDLE, CM
RAT(I)  = RC/RS WHERE I = SPINDLE NUMBER
CW      = GRAVIMETRIC CONCENTRATION (DECIMAL FRACTION)
CV      = VOLUMETRIC CONVENTRATION (DECIMAL FRACTION)
TEMP    = SLURRY TEMPERATURE IN DEGREES CENTIGRADE
S       = SPECIFIC GRAVITY OF SOLIDS
SL      = SPECIFIC GRAVITY OF LIQUID
TAUK(I)= SPINDLE COEFFICIENT FOR STRESS WHERE I = SPINDLE
NUMBER
SM      = SPECIFIC GRAVITY OF SLURRY
YIELD   = SLURRY YIELD STRESS, DYNES/CM/CM
RPM     = SPINDLE SPEED, REV/MIN
DIAL    = BROOKFIELD DIAL READING
STRESS  = STRESS MEASURED AT WALL OF SPINDLE, DYNES/CM/CM
KEMGAM  = SHEAR RATE (1/SEC) FROM KRIEGER, ELROD, MARON (KEM)
 EQUATION
BROGAM  = SHEAR RATE (1/SEC) FROM BROOKEY EQUATION
ALVGAM  = SHEAR RATE (1/SEC) FROM ALVES ET AL. EQUATION
OMEGA   = SPINDLE ANGULAR VELOCITY, RADIANS/SEC
K       = CONSISTENCY INDEX,  DYNE-SEC**N/CM**2

N       = FLOW BEHAVIOR INDEX, DIMENSIONLESS
KYP     = INTERCEPT OF DELTAU VS. ALOGAM (LOGARITHMIC PLOT)
NYP     = SLOPE OF DELTAU VS. ALVGAM  (LOGARITHMIC PLOT)
DELTAU  = TAU - TAUY
TAU     = SHEAR STRESS  DYNES/CM/CM
TAUY    = YIELD
RYP     = COEFFICIENT OF CORRELATION FOR LEAST SQUARES FIT
DIA     = PIPE INNER DIAMETER, INCHES
RELRUF  = RELATIVE ROUGHNESS (ABSOLUTE ROUGHNESS/PIPE ID)
VCL     = CRITICAL VELOCITY DENOTING LAMINAR FLOW
VCT     = CRITICAL VELOCITY DENOTING TRANSITION FLOW
VLCTY   = VELOCITY OF FLOW,  FPS
SHRATE  = SHEAR RATE,  1/SEC
THROUGHPUT IS IN DRY SHORT TONS PER HOUR
RE      = GENERALIZED REYNOLDS NUMBER BASED ON SLURRY' PROPERTIES
F M     = FANNING FRICTION FACTOR
PDROP   = PRESSURE DROP DUE TO WALL FRICTION' ONLY,  PSI/MILE
KILW-HR/TON-MI = SPECIFIC POWER
```

Fig.5.2.1.1, p.1

```
FOUR DATA POINTS
SLURRY: SSI METHACOAL  (64.3X)          3-12-77
                                    I RC(CM) RS(CM)  RAT(I) LN(RAT(I)) CW(X) CV(X) TEMP(C)   S     SL   TAUK(I)   SM  YIELD(DYNES/CM/CM)
                                    2 4.0000 0.5128  7.8003   2.0542   64.3  50.0   25.5   1.429 0.791  0.1407 1.110        6.40

RPM DIAL  STRESS  KEMGAM  BROGAM ALVGAM OMEGA  LN(OMEGA) LN(STRESS)                                                  TAU-TAUY
 10. 10.5   74.632.49E+01   4.50   4.50 1.0472   0.0461    4.3125   STRESS = 75.08(OMEGA )** 0.465  R = 0.9933  68.23
 20. 15.9  113.014.98E+01   9.01   9.01 2.0944   0.7393    4.7274   STRESS = 17.19(KEMGAM)** 0.465  R = 0.9933 106.61
 50. 21.5  152.811.25E+02  22.52  22.51 5.2360   1.6556    5.0292   STRESS = 38.09(BROGAM)** 0.465  R = 0.9933 146.41
100. 32.2  228.862.49E+02  45.03  45.0310.4720   2.3487    5.4331   STRESS = 38.10(ALVGAM)** 0.465  R = 0.9933 222.46

K = 38.10  N = 0.465  KYP = 33.73  NYP = 0.490  RYP = 0.9931
```

Fig. 5.2.1.1, p.2

SYSTEM PROPERTIES

```
MINERAL --- SSI METHACOAL  (64.3%)          3-12-77
AVERAGE SOLID          SPECIFIC GRAVITY (S ) --- 1.429
LIQUID PHASE           SPECIFIC GRAVITY (S_) --- 0.791
SLURRY                 SPECIFIC GRAVITY (SM) --- 1.110
SLURRY CONCENTRATION BY WEIGHT               --- 0.643
SLURRY CONCENTRATION BY VOLUME               --- 0.500
ABSOLUTE PIPEWALL ROUGHNESS (E), FEET        --- 0.00015000
GRAVITATIONAL ACCELERATION                   --- 32.1573 FEET/SEC/SEC
SLURRY TEMPERATURE (TEMP), DEGREES CELSIUS   --- 25.5
PIPE TYPE                                    ---
PIPE SLOPE                                   --- HORIZONTAL
```

MESH	PERCENT
0.007/100	0.30
100/200	13.70
200/325	65.60
325/PAN	20.40
TOTAL =	100.0

```
WEIGHTED MEAN DIAMETER =5.8349E-02 MM        THROUGHPUT (TONS/HR) =  SHORT
COEFFICIENT OF VARIATION =    44.68801       SETTLING REGIME = LAMINAR
DRAG COEFF OF WEIGHTED MEAN DIA = 221.865    VISCOSITY FACTOR =  1.0
REYNOLDS NUMBER OF SETTLING =      0.11
ROSIN - RAMMLER EQUATION!  R = 100 * EXP(-(D/  0.068649) **  2.585615)
SLOPE =  2.585615      INTERCEPT B =  0.06864908340  MILLIMETERS
CORRELATION COEFF. = 0.985528      D50 =  0.06 MILLIMETERS

KYP =  33.73      NYP =  0.490      TAUY =    6.396
SM THEORY

  PH = N.A.
```


Fig.5.2.1.1,p.3

DIA IN.	RELRUF E/DIA	VCL FPS	VCT FPS	VLCTY FPS	THROUGHPUT TONS/HR	REYNOLDS NUMBER	FM MIX	PSI MI.	KILW-HR TON-MI
6.000	0.00030	11.02	16.89	4.00	62.8	455.	.0000	182.930	0.4465
8.000	0.00022	10.04	15.39	4.00	111.7	523.	.0000	119.567	0.2918
10.000	0.00018	9.34	14.31	4.00	174.5	584.	.0000	86.049	0.2100
12.000	0.00015	8.80	13.49	4.00	251.3	639.	.0000	65.778	0.1606
14.000	0.00013	8.37	12.83	4.00	342.0	689.	.0000	52.464	0.1281
16.000	0.00011	8.02	12.28	4.00	446.7	735.	.0000	43.068	0.1051
18.000	0.00010	7.72	11.82	4.00	565.4	779.	.0000	36.254	0.0885
20.000	0.00009	7.46	11.43	4.00	698.0	820.	.0000	31.054	0.0758
22.000	0.00008	7.23	11.08	4.00	844.6	860.	.0000	26.978	0.0659
24.000	0.00007	7.03	10.77	4.00	1005.1	897.	.0000	23.776	0.0580
6.000	0.00030	11.02	16.89	5.00	78.5	637.	.0000	203.465	0.4966
8.000	0.00022	10.04	15.39	5.00	139.6	733.	.0000	132.991	0.3246
10.000	0.00018	9.34	14.31	5.00	218.1	818.	.0000	95.654	0.2335
12.000	0.00015	8.80	13.49	5.00	314.1	894.	.0000	73.151	0.1786
14.000	0.00013	8.37	12.83	5.00	427.5	965.	.0000	58.284	0.1423
16.000	0.00011	8.02	12.28	5.00	558.4	1030.	.0000	47.878	0.1169
18.000	0.00010	7.72	11.82	5.00	706.7	1091.	.0000	40.226	0.0982
20.000	0.00009	7.46	11.43	5.00	872.5	1149.	.0000	34.454	0.0841
22.000	0.00008	7.23	11.08	5.00	1055.7	1204.	.0000	29.963	0.0731
24.000	0.00007	7.03	10.77	5.00	1256.4	1256.	.0000	26.360	0.0643
6.000	0.00030	11.02	16.89	6.00	94.2	838.	.0000	222.044	0.5420
8.000	0.00022	10.04	15.39	6.00	167.5	965.	.0000	145.115	0.3542
10.000	0.00018	9.34	14.31	6.00	261.8	1077.	.0000	104.345	0.2547
12.000	0.00015	8.80	13.49	6.00	376.9	1178.	.0000	79.711	0.1946
14.000	0.00013	8.37	12.83	6.00	513.0	1270.	.0000	63.496	0.1550
16.000	0.00011	8.02	12.28	6.00	670.1	1356.	.0000	52.197	0.1273
18.000	0.00010	7.72	11.82	6.00	848.1	1437.	.0000	43.852	0.1070
20.000	0.00009	7.46	11.43	6.00	1047.0	1513.	.0000	37.576	0.0917
22.000	0.00008	7.23	11.08	6.00	1266.9	1585.	.0000	32.647	0.0797
24.000	0.00007	7.03	10.77	6.00	1507.7	1654.	.0000	28.712	0.0701
6.000	0.00030	11.02	16.89	7.00	109.9	1058.	.0000	239.121	0.5837
8.000	0.00022	10.04	15.39	7.00	195.4	1218.	.0000	156.218	0.3813
10.000	0.00018	9.34	14.31	7.00	305.4	1359.	.0000	112.313	0.2741
12.000	0.00015	8.80	13.49	7.00	439.7	1486.	.0000	85.827	0.2095
14.000	0.00013	8.37	12.83	7.00	598.5	1603.	.0000	68.324	0.1668
16.000	0.00011	8.02	12.28	7.00	781.8	1711.	.0000	56.149	0.1371
18.000	0.00010	7.72	11.82	7.00	989.4	1813.	.0000	47.181	0.1152
20.000	0.00009	7.46	11.43	7.00	1221.5	1909.	.0000	40.399	0.0986
22.000	0.00008	7.23	11.08	7.00	1478.0	2001.	.0000	35.109	0.0857
24.000	0.00007	7.03	10.77	7.00	1759.0	2088.	.0000	30.879	0.0754
6.000	0.00030	11.02	16.89	8.00	125.6	1294.	.0000	254.956	0.6224
8.000	0.00022	10.04	15.39	8.00	223.4	1490.	.0000	166.533	0.4065
10.000	0.00018	9.34	14.31	8.00	349.0	1663.	.0000	119.720	0.2922
12.000	0.00015	8.80	13.49	8.00	502.6	1818.	.0000	91.465	0.2233
14.000	0.00013	8.37	12.83	8.00	684.0	1961.	.0000	72.847	0.1778
16.000	0.00011	8.02	12.28	8.00	893.5	2094.	.0000	59.784	0.1459
18.000	0.00010	7.72	11.82	8.00	1130.8	2484.	.0462	77.956	0.1903
20.000	0.00009	7.46	11.43	8.00	1396.0	2616.	.0455	69.024	0.1685
22.000	0.00008	7.23	11.08	8.00	1689.2	2741.	.0448	61.836	0.1509

Fig.5.2.1.1,p.4

DIA IN.	RELRUF E/DIA	VCL FPS	VCT FPS	VLCTY FPS	THROUGHPUT TONS/HR	REYNOLDS NUMBER	FM MIX	PSI MI	KILW-HR TON-MI
24.000	0.00007	7.03	10.77	8.00	2010.3	2861.	.0442	55.934	0.1365
6.000	0.00030	11.02	16.89	9.00	141.3	1546.	.0000	269.877	0.6587
8.000	0.00022	10.04	15.39	9.00	251.3	1780.	.0000	176.229	0.4302
10.000	0.00018	9.34	14.31	9.00	392.6	1986.	.0000	126.697	0.3093
12.000	0.00015	8.80	13.49	9.00	565.4	2432.	.0466	149.127	0.3640
14.000	0.00013	8.37	12.83	9.00	769.6	2623.	.0455	124.777	0.3046
16.000	0.00011	8.02	12.28	9.00	1005.1	2801.	.0446	106.958	0.2611
18.000	0.00010	7.72	11.82	9.00	1272.1	2967.	.0438	93.379	0.2279
20.000	0.00009	7.46	11.43	9.00	1570.5	3125.	.0431	82.710	0.2019
22.000	0.00008	7.23	11.08	9.00	1900.3	3274.	.0425	74.121	0.1809
24.000	0.00007	7.03	10.77	9.00	2261.6	3417.	.0419	67.066	0.1637
6.000	0.00030	11.02	16.89	10.00	157.1	1813.	.0000	203.885	0.6929
8.000	0.00022	10.04	15.39	10.00	279.2	2087.	.0000	185.407	0.4526
10.000	0.00018	9.34	14.31	10.00	436.3	2608.	.0456	216.271	0.5279
12.000	0.00015	8.80	13.49	10.00	628.2	2852.	.0444	175.222	0.4277
14.000	0.00013	8.37	12.83	10.00	855.1	3076.	.0433	146.699	0.3581
16.000	0.00011	8.02	12.28	10.00	1116.8	3284.	.0425	125.798	0.3071
18.000	0.00010	7.72	11.82	10.00	1413.5	3479.	.0417	109.865	0.2682
20.000	0.00009	7.46	11.43	10.00	1745.0	3664.	.0411	97.343	0.2376
22.000	0.00008	7.23	11.08	10.00	2111.5	3839.	.0405	87.257	0.2130
24.000	0.00007	7.03	10.77	10.00	2512.8	4006.	.0400	78.971	0.1928
6.000	0.00030	11.02	16.89	11.00	172.8	2093.	.0000	297.270	0.7256
8.000	0.00022	10.04	15.39	11.00	307.1	2699.	.0452	323.921	0.7907
10.000	0.00018	9.34	14.31	11.00	479.9	3011.	.0436	250.377	0.6111
12.000	0.00015	8.80	13.49	11.00	691.0	3293.	.0425	202.951	0.4954
14.000	0.00013	8.37	12.83	11.00	940.6	3552.	.0415	169.981	0.4149
16.000	0.00011	8.02	12.28	11.00	1228.5	3792.	.0407	145.813	0.3559
18.000	0.00010	7.72	11.82	11.00	1554.8	4017.	.0400	127.383	0.3109
20.000	0.00009	7.46	11.43	11.00	1919.5	4230.	.0394	112.893	0.2756
22.000	0.00008	7.23	11.08	11.00	2322.6	4433.	.0388	101.220	0.2471
24.000	0.00007	7.03	10.77	11.00	2764.1	4626.	.0383	91.628	0.2237
6.000	0.00030	11.02	16.89	12.00	188.5	2673.	.0454	516.266	1.2601
8.000	0.00022	10.04	15.39	12.00	335.0	3078.	.0434	370.340	0.9040
10.000	0.00018	9.34	14.31	12.00	523.5	3434.	.0420	286.405	0.6991
12.000	0.00015	8.80	13.49	12.00	753.9	3755.	.0408	232.251	0.5669
14.000	0.00013	8.37	12.83	12.00	1026.1	4050.	.0399	194.589	0.4750
16.000	0.00011	8.02	12.28	12.00	1340.2	4324.	.0391	166.971	0.4076
18.000	0.00010	7.72	11.82	12.00	1696.2	4581.	.0385	145.925	0.3561
20.000	0.00009	7.46	11.43	12.00	2094.0	4824.	.0379	129.339	0.3157
22.000	0.00008	7.23	11.08	12.00	2533.8	5055.	.0374	115.989	0.2831
24.000	0.00007	7.03	10.77	12.00	3015.4	5276.	.0369	105.017	0.2563
6.000	0.00030	11.02	16.89	13.00	204.2	3016.	.0437	583.982	1.4254
8.000	0.00022	10.04	15.39	13.00	363.0	3473.	.0419	419.164	1.0231
10.000	0.00018	9.34	14.31	13.00	567.1	3875.	.0405	324.310	0.7916
12.000	0.00015	8.80	13.49	13.00	816.7	4237.	.0394	263.085	0.6422
14.000	0.00013	8.37	12.83	13.00	1111.6	4570.	.0385	220.491	0.5382
16.000	0.00011	8.02	12.28	13.00	1451.9	4880.	.0378	189.247	0.4619
18.000	0.00010	7.72	11.82	13.00	1837.5	5170.	.0372	165.409	0.4037
20.000	0.00009	7.46	11.43	13.00	2268.5	5444.	.0366	146.657	0.3580

Fig.5.2.1.1,p.5

DIA IN.	RELRUF E/DIA	VCL FPS	VCT FPS	VLCTY FPS	THROUGHPUT TONS/HR	REYNOLDS NUMBER	FM MIX	PSI MI	KILW-HR TON-MI
22.000	0.00008	7.23	11.08	13.00	2744.9	5704.	.0361	131.545	0.3211
24.000	0.00007	7.03	10.77	13.00	3266.7	5953.	.0357	119.121	0.2908
6.000	0.00030	11.02	16.89	14.00	219.9	3373.	.0423	654.937	1.5986
8.000	0.00022	10.04	15.39	14.00	390.9	3884.	.0405	470.340	1.1480
10.000	0.00018	9.34	14.31	14.00	610.8	4334.	.0392	364.052	0.8886
12.000	0.00015	8.80	13.49	14.00	879.5	4739.	.0381	295.421	0.7211
14.000	0.00013	8.37	12.83	14.00	1197.1	5111.	.0373	247.659	0.6045
16.000	0.00011	8.02	12.28	14.00	1563.5	5457.	.0366	212.615	0.5190
18.000	0.00010	7.72	11.82	14.00	1978.9	5782.	.0360	185.871	0.4537
20.000	0.00009	7.46	11.43	14.00	2443.0	6088.	.0355	164.831	0.4023
22.000	0.00008	7.23	11.08	14.00	2956.1	6380.	.0350	147.870	0.3609
24.000	0.00007	7.03	10.77	14.00	3518.0	6658.	.0346	133.924	0.3269
6.000	0.00030	11.02	16.89	15.00	235.6	3744.	.0410	729.069	1.7796
8.000	0.00022	10.04	15.39	15.00	418.8	4311.	.0393	523.823	1.2786
10.000	0.00018	9.34	14.31	15.00	654.4	4809.	.0380	405.595	0.9900
12.000	0.00015	8.80	13.49	15.00	942.3	5259.	.0370	329.229	0.8036
14.000	0.00013	8.37	12.83	15.00	1282.6	5672.	.0362	276.068	0.6739
16.000	0.00011	8.02	12.28	15.00	1675.2	6056.	.0356	237.055	0.5786
18.000	0.00010	7.72	11.82	15.00	2120.2	6416.	.0350	207.275	0.5059
20.000	0.00009	7.46	11.43	15.00	2617.5	6757.	.0345	183.842	0.4487
22.000	0.00008	7.23	11.08	15.00	3167.2	7080.	.0340	164.949	0.4026
24.000	0.00007	7.03	10.77	15.00	3769.3	7389.	.0336	149.412	0.3647
6.000	0.00030	11.02	16.89	16.00	251.3	4127.	.0399	806.324	1.9681
8.000	0.00022	10.04	15.39	16.00	446.7	4752.	.0382	579.572	1.4147
10.000	0.00018	9.34	14.31	16.00	698.0	5302.	.0370	448.908	1.0957
12.000	0.00015	8.80	13.49	16.00	1005.1	5797.	.0360	364.482	0.8897
14.000	0.00013	8.37	12.83	16.00	1368.1	6253.	.0353	305.697	0.7462
16.000	0.00011	8.02	12.28	16.00	1786.9	6676.	.0346	262.547	0.6408
18.000	0.00010	7.72	11.82	16.00	2261.6	7073.	.0341	229.623	0.5604
20.000	0.00009	7.46	11.43	16.00	2792.0	7448.	.0336	203.675	0.4971
22.000	0.00008	7.23	11.08	16.00	3378.4	7804.	.0331	182.769	0.4461
24.000	0.00007	7.03	10.77	16.00	4020.5	8145.	.0327	165.574	0.4041

Fig.5.2.1.1,p.6

FOUR DATA POINTS

SLURRY: SSI METHACOAL (63.7%) 3-12-77

I	RC(CM)	RS(CM)	RAT(I)	LN(RAT(I))	CW(%)	CV(%)	TEMP(C)	S	SL	TAUK(I)	SM	YIELD(DYNES/CM/CM)
2	4.0000	0.5128	7.8003	2.0542	63.7	49.2	25.1	1.429	0.791	0.1407	1.105	2.35

RPM	DIAL	STRESS	KEMGAM	BROGAM	ALVGAM	OMEGA	LN(OMEGA)	LN(STRESS)			TAU-TAUY
10.	7.0	49.89	4.53E+00	3.44	3.43	1.0472	0.0461	3.9099	STRESS = 44.67(OMEGA)** 0.610	R = 0.9908	47.55
20.	8.9	63.04	9.06E+00	6.87	6.87	2.0944	0.7393	4.1438	STRESS = 10.28(KEMGAM)** 0.610	R = 0.9908	60.70
50.	16.8	119.40	2.26E+01	17.19	17.17	5.2360	1.6556	4.7825	STRESS = 21.63(BROGAM)** 0.610	R = 0.9908	117.06
100.	27.7	196.94	4.53E+01	34.37	34.33	10.4720	2.3487	5.2829	STRESS = 21.65(ALVGAM)** 0.610	R = 0.9908	194.60

K = 21.64 N = 0.610 KYP = 20.22 NYP = 0.626 RYP = 0.9912

Fig.5.2.1.1,p.7

SYSTEM PROPERTIES

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MINERAL --- SSI METHACOAL  (63.7%)          3-12-77
AVERAGE SOLID         SPECIFIC GRAVITY (S ) --- 1.429
LIQUID PHASE          SPECIFIC GRAVITY (SL) --- 0.791
SLURRY                SPECIFIC GRAVITY (SM) --- 1.105
SLURRY CONCENTRATION BY WEIGHT              --- 0.637
SLURRY CONCENTRATION BY VOLUME              --- 0.492
ABSOLUTE PIPEWALL ROUGHNESS (E), FEET       --- 0.00015000
GRAVITATIONAL ACCELERATION                  --- 32.1573 FEET/SEC/SEC
SLURRY TEMPERATURE (TEMP), DEGREES CELSIUS  --- 25.1
PIPE TYPE                                   ---
PIPE SLOPE                                  --- HORIZONTAL
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MESH	PERCENT
0.007/100	0.30
100/200	13.70
200/325	65.60
325/PAN	20.40
TOTAL =	100.0

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WEIGHTED MEAN DIAMETER =5.8349E-02 MM        THROUGHPUT (TONS/HR) =  SHORT
COEFFICIENT OF VARIATION =   44.68001        SETTLING REGIME = LAMINAR
DRAG COEFF OF WEIGHTED MEAN DIA = 221.865    VISCOSITY FACTOR =  1.0
REYNOLDS NUMBER OF SETTLING =    0.11
ROSIN - RAMMLER EQUATION!  R = 100 * EXP(-(D/  0.068649) **  2.585615)
SLOPE =  2.585615     INTERCEPT D =  0.06864908340  MILLIMETERS
CORRELATION COEFF. = 0.965528      D50 =  0.06 MILLIMETERS

KYP =  20.22      NYP =  0.626      TAUY =     2.345
SM THEORY

  PH = N.A.
```

Fig.5.2.1.1,p.8

DIA IN.	RELRUF E/DIA	VCL FPS	VCT FPS	VLCTY FPS	THROUGHPUT TONS/HR	REYNOLDS NUMBER	FM MIX	PSI MI	KILW-HR TON-MI
6.000	0.00030	12.47	19.93	4.00	61.9	440.	.0000	184.843	0.4576
8.000	0.00022	10.94	17.48	4.00	110.1	527.	.0000	116.035	0.2873
10.000	0.00018	9.88	15.79	4.00	172.1	607.	.0000	80.827	0.2001
12.000	0.00015	9.09	14.53	4.00	247.8	680.	.0000	60.225	0.1491
14.000	0.00013	8.47	13.55	4.00	337.2	749.	.0000	46.915	0.1161
16.000	0.00011	7.97	12.75	4.00	440.5	814.	.0000	37.775	0.0935
13.000	0.00010	7.56	12.08	4.00	557.5	876.	.0000	31.267	0.0774
23.000	0.00009	7.20	11.51	4.00	638.2	936.	.0000	26.388	0.0653
22.000	0.00008	6.90	11.02	4.00	832.8	994.	.0000	22.622	0.0560
24.000	0.00007	6.63	10.59	4.00	991.0	1049.	.0000	19.662	0.0487
6.000	0.00030	12.47	19.93	5.00	77.4	598.	.0000	212.262	0.5255
8.000	0.00022	10.94	17.48	5.00	137.6	717.	.0000	133.129	0.3296
10.000	0.00018	9.88	15.79	5.00	215.1	824.	.0000	92.828	0.2298
12.000	0.00015	9.09	14.53	5.00	309.7	924.	.0000	69.096	0.1711
14.000	0.00013	8.47	13.55	5.00	421.5	1017.	.0000	53.859	0.1333
16.000	0.00011	7.97	12.75	5.00	550.6	1106.	.0000	43.405	0.1075
18.000	0.00010	7.56	12.08	5.00	696.8	1191.	.0000	35.829	0.0887
20.000	0.00009	7.20	11.51	5.00	860.3	1272.	.0000	30.220	0.0748
22.000	0.00008	6.90	11.02	5.00	1040.9	1350.	.0000	25.939	0.0642
24.000	0.00007	6.63	10.59	5.00	1238.8	1426.	.0000	22.561	0.0559
6.000	0.00030	12.47	19.93	6.00	92.9	769.	.0000	237.754	0.5886
8.000	0.00022	10.94	17.48	6.00	165.2	921.	.0000	149.122	0.3692
10.000	0.00018	9.88	15.79	6.00	258.1	1059.	.0000	103.827	0.2570
12.000	0.00015	9.09	14.53	6.00	371.6	1187.	.0000	77.357	0.1915
14.000	0.00013	8.47	13.55	6.00	505.8	1307.	.0000	60.269	0.1492
16.000	0.00011	7.97	12.75	6.00	660.7	1421.	.0000	48.562	0.1202
18.000	0.00010	7.56	12.08	6.00	836.2	1530.	.0000	40.190	0.0994
20.000	0.00009	7.20	11.51	6.00	1032.3	1634.	.0000	33.834	0.0838
22.000	0.00008	6.90	11.02	6.00	1249.1	1735.	.0000	29.007	0.0718
24.000	0.00007	6.63	10.59	6.00	1486.6	1832.	.0000	25.183	0.0623
6.000	0.00030	12.47	19.93	7.00	108.4	950.	.0000	261.617	0.6477
8.000	0.00022	10.94	17.48	7.00	192.7	1138.	.0000	164.085	0.4062
10.000	0.00018	9.88	15.79	7.00	301.1	1308.	.0000	114.297	0.2830
12.000	0.00015	9.09	14.53	7.00	433.6	1467.	.0000	85.060	0.2106
14.000	0.00013	8.47	13.55	7.00	590.2	1615.	.0000	66.306	0.1641
16.000	0.00011	7.97	12.75	7.00	770.8	1756.	.0000	53.386	0.1322
18.000	0.00010	7.56	12.08	7.00	975.6	1890.	.0000	44.111	0.1092
20.000	0.00009	7.20	11.51	7.00	1204.4	2019.	.0000	37.231	0.0922
22.000	0.00008	6.90	11.02	7.00	1457.3	2339.	.0471	49.541	0.1226
24.000	0.00007	6.63	10.59	7.00	1734.3	2470.	.0463	44.643	0.1105
6.000	0.00030	12.47	19.93	8.00	123.9	1141.	.0000	284.298	0.7038
8.000	0.00022	10.94	17.48	8.00	220.2	1367.	.0000	178.316	0.4414
10.000	0.00018	9.88	15.79	8.00	344.1	1572.	.0003	124.160	0.3074
12.000	0.00015	9.09	14.53	8.00	495.5	1762.	.0000	92.421	0.2288
14.000	0.00013	8.47	13.55	8.00	674.5	1940.	.0000	71.996	0.1782
16.000	0.00011	7.97	12.75	8.00	880.9	2302.	.0474	89.471	0.2215
18.000	0.00010	7.56	12.08	8.00	1114.9	2478.	.0463	77.702	0.1924
20.000	0.00009	7.20	11.51	8.00	1376.5	2647.	.0453	68.494	0.1696
22.000	0.00008	6.90	11.02	8.00	1665.5	2810.	.0445	61.117	0.1513

Fig.5.2.1.1,p.9

DIA IN.	RELRUF E/DIA	VCL FPS	VCT FPS	VLCTY FPS	THROUGHPUT TONS/HR	REYNOLDS NUMBER	FM MIX	PSI MI	KILW-HR TON-MI
24.000	0.00007	6.63	10.59	8.00	1982.1	2967.	.0437	55.086	0.1364
6.000	0.00030	12.47	19.93	9.00	139.4	1342.	.0000	305.940	0.7574
8.000	0.00022	10.94	17.48	9.00	247.8	1607.	.0000	191.854	0.4750
10.000	0.00018	9.88	15.79	9.00	387.1	1848.	.0000	133.597	0.3307
12.000	0.00015	9.09	14.53	9.00	557.5	2071.	.0003	99.415	0.2461
14.000	0.00013	8.47	13.55	9.00	758.8	2489.	.0462	126.330	0.3127
16.000	0.00011	7.97	12.75	9.00	991.0	2706.	.0450	107.667	0.2665
18.000	0.00010	7.56	12.08	9.00	1254.3	2913.	.0440	93.531	0.2315
20.000	0.00009	7.20	11.51	9.00	1548.5	3112.	.0431	82.482	0.2042
22.000	0.00008	6.90	11.02	9.00	1873.7	3303.	.0424	73.626	0.1823
24.000	0.00007	6.63	10.59	9.00	2229.9	3488.	.0417	66.383	0.1643
6.000	0.00030	12.47	19.93	10.00	154.9	1551.	.0000	326.629	0.8086
8.000	0.00022	10.94	17.48	10.00	275.3	1857.	.0000	204.822	0.5071
10.000	0.00018	9.88	15.79	10.00	430.1	2330.	.0472	223.078	0.5523
12.000	0.00015	9.09	14.53	10.00	619.4	2611.	.0456	179.316	0.4439
14.000	0.00013	8.47	13.55	10.00	843.1	2876.	.0442	149.126	0.3692
16.000	0.00011	7.97	12.75	10.00	1101.2	3127.	.0431	127.155	0.3148
18.000	0.00010	7.56	12.08	10.00	1393.7	3366.	.0421	110.525	0.2736
20.000	0.00009	7.20	11.51	10.00	1720.6	3596.	.0413	97.486	0.2413
22.000	0.00008	6.90	11.02	10.00	2081.9	3817.	.0406	87.048	0.2155
24.000	0.00007	6.63	10.59	10.00	2477.6	4031.	.0399	78.527	0.1944
6.000	0.00030	12.47	19.93	11.00	170.3	1768.	.0000	346.598	0.8580
8.000	0.00022	10.94	17.48	11.00	302.8	2309.	.0474	338.607	0.8383
10.000	0.00018	9.88	15.79	11.00	473.2	2656.	.0454	259.166	0.6416
12.000	0.00015	9.09	14.53	11.00	681.5	2977.	.0438	208.404	0.5159
14.000	0.00013	8.47	13.55	11.00	927.4	3279.	.0425	173.400	0.4293
16.000	0.00011	7.97	12.75	11.00	1211.3	3564.	.0414	147.914	0.3662
18.000	0.00010	7.56	12.08	11.00	1533.0	3837.	.0405	128.592	0.3183
20.000	0.00009	7.20	11.51	11.00	1892.6	4099.	.0397	113.478	0.2809
22.000	0.00008	6.90	11.02	11.00	2290.1	4351.	.0390	101.356	0.2509
24.000	0.00007	6.63	10.59	11.00	2725.4	4595.	.0384	91.435	0.2264
6.000	0.00030	12.47	19.93	12.00	185.8	1992.	.0000	365.940	0.9059
8.000	0.00022	10.94	17.48	12.00	330.3	2602.	.0457	388.285	0.9612
10.000	0.00018	9.88	15.79	12.00	516.2	2993.	.0437	297.321	0.7360
12.000	0.00015	9.09	14.53	12.00	743.3	3355.	.0422	239.210	0.5922
14.000	0.00013	8.47	13.55	12.00	1011.7	3695.	.0410	199.117	0.4929
16.000	0.00011	7.97	12.75	12.00	1321.4	4017.	.0400	169.913	0.4206
18.000	0.00010	7.56	12.08	12.00	1672.4	4325.	.0391	147.764	0.3658
20.000	0.00009	7.20	11.51	12.00	2064.7	4620.	.0384	130.432	0.3229
22.000	0.00008	6.90	11.02	12.00	2498.3	4904.	.0377	116.528	0.2885
24.000	0.00007	6.63	10.59	12.00	2973.1	5178.	.0371	105.145	0.2603
6.000	0.00030	12.47	19.93	13.00	201.3	2426.	.0467	621.717	1.5391
8.002	0.00022	10.94	17.48	13.00	357.9	2905.	.0442	440.555	1.2906
10.000	0.00018	9.88	15.79	13.00	559.2	3341.	.0423	337.541	0.8356
12.000	0.00015	9.09	14.53	13.00	805.2	3745.	.0409	271.693	0.6726
14.000	0.00013	8.47	13.55	13.00	1096.0	4124.	.0397	226.242	0.5601
16.000	0.00011	7.97	12.75	13.00	1431.5	4484.	.0387	193.122	0.4781
18.000	0.00010	7.56	12.08	13.00	1811.8	4827.	.0379	167.994	0.4159
20.000	0.00009	7.20	11.51	13.00	2236.7	5157.	.0372	148.327	0.3672

Fig.5.2.1.1,p10

DIA IN.	RELRUF E/DIA	VCL FPS	VCT FPS	VLCTY FPS	THROUGHPUT TONS/HR	REYNOLDS NUMBER	FM MIX	PSI MI	KILW-HR TON-MI
22.000	0.00008	6.90	11.02	13.00	2706.4	5474.	.0365	132.545	0.3281
24.000	0.00007	6.63	10.59	13.00	3220.9	5780.	.0360	119.621	0.2961
6.000	0.00030	12.47	19.93	14.00	216.8	2686.	.0453	698.817	1.7300
8.000	0.00022	10.94	17.48	14.00	385.4	3216.	.0428	495.430	1.2265
10.000	0.00018	9.88	15.79	14.00	602.2	3698.	.0410	379.781	0.9402
12.000	0.00015	9.09	14.53	14.00	867.2	4146.	.0397	305.819	0.7571
14.000	0.00013	8.47	13.55	14.00	1180.3	4566.	.0385	254.746	0.6306
16.000	0.00011	7.97	12.75	14.00	1541.6	4964.	.0376	217.516	0.5385
18.000	0.00010	7.56	12.08	14.00	1951.1	5345.	.0368	189.263	0.4685
20.000	0.00009	7.20	11.51	14.00	2408.8	5709.	.0361	167.142	0.4138
22.000	0.00008	6.90	11.02	14.00	2914.6	6060.	.0355	149.388	0.3698
24.000	0.00007	6.63	10.59	14.00	3468.7	6400.	.0350	134.847	0.3338
6.000	0.00030	12.47	19.93	15.00	232.3	2953.	.0440	779.339	1.9293
8.000	0.00022	10.94	17.48	15.00	412.9	3536.	.0416	552.860	1.3687
10.000	0.00018	9.88	15.79	15.00	645.2	4066.	.0399	424.002	1.0497
12.000	0.00015	9.09	14.53	15.00	929.1	4558.	.0386	341.596	0.8456
14.000	0.00013	8.47	13.55	15.00	1264.6	5020.	.0375	284.602	0.7046
16.000	0.00011	7.97	12.75	15.00	1651.7	5458.	.0366	243.074	0.6018
18.000	0.00010	7.56	12.08	15.00	2090.5	5876.	.0358	211.549	0.5237
20.000	0.00009	7.20	11.51	15.00	2580.8	6277.	.0352	186.861	0.4626
22.000	0.00008	6.90	11.02	15.00	3122.8	6663.	.0346	167.042	0.4135
24.000	0.00007	6.63	10.59	15.00	3716.4	7036.	.0341	150.827	0.3733
6.000	0.00030	12.47	19.93	16.00	247.8	3227.	.0429	863.345	2.1373
8.000	0.00022	10.94	17.48	16.00	440.5	3864.	.0406	612.798	1.5170
10.000	0.00018	9.88	15.79	16.00	688.2	4443.	.0389	470.170	1.1639
12.000	0.00015	9.09	14.53	16.00	991.0	4981.	.0376	378.874	0.9379
14.000	0.00013	8.47	13.55	16.00	1348.9	5485.	.0366	315.787	0.7818
16.000	0.00011	7.97	12.75	16.00	1761.9	5964.	.0357	269.773	0.6678
18.000	0.00010	7.56	12.08	16.00	2229.9	6421.	.0350	234.834	0.5814
20.000	0.00009	7.20	11.51	16.00	2752.9	6858.	.0343	207.467	0.5136
22.000	0.00008	6.90	11.02	16.00	3331.0	7280.	.0338	185.494	0.4592
24.000	0.00007	6.63	10.59	16.00	3964.2	7688.	.0333	167.490	0.4146

Fig.5.2.1.1,p.11

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FOUR DATA POINTS
SLURRY:  SSI METHACOAL  (59.2%)          3-12-77
                                        I RC(CM) RS(CM)  RAT(I) LN(RAT(I)) CW(X) CV(X) TEMP(C)    S      SL  TAUK(I)    SM  YIELD(DYNES/CM/CM)
                                        1 4.0000 0.9421  4.2458   1.4459   59.2  44.6   25.9   1.429 0.791  0.5814 1.076         7.28

RPM DIAL  STRESS  KEMGAM  BROGAM ALVGAM OMEGA  LN(OMEGA) LN(STRESS)                                                       TAU-TAUY
 10. 32.6    56.049.68E+00   5.36   5.36 1.0472   0.0461    4.0260   STRESS = 59.65(OMEGA )** 0.391  R = 0.9785  48.76
 20. 51.1    87.961.94E+01  10.72  10.72 2.0944   0.7393    4.4769   STRESS = 25.01(KEMGAM)** 0.391  R = 0.9785  80.68
 50. 68.7   118.204.84E+01  26.81  26.79 5.2360   1.6556    4.7724   STRESS = 31.51(BROGAM)** 0.391  R = 0.9785 110.92
100. 82.2   141.319.68E+01  53.61  53.5810.4720   2.3487    4.9510   STRESS = 31.51(ALVGAM)** 0.391  R = 0.9785 134.04

K = 31.51   N = 0.391   KYP = 26.15   NYP = 0.427   RYP = 0.9761


LAST THREE DATA POINTS
SLURRY:  SSI METHACOAL  (59.2%)          3-12-77
                                        I RC(CM) RS(CM)  RAT(I) LN(RAT(I)) CW(X) CV(X) TEMP(C)    S      SL  TAUK(I)    SM  YIELD(DYNES/CM/CM)
                                        1 4.0000 0.9421  4.2458   1.4459   59.2  44.6   25.9   1.429 0.791  0.5814 1.076         7.28

RPM DIAL  STRESS  KEMGAM  BROGAM ALVGAM OMEGA  LN(OMEGA) LN(STRESS)                                                       TAU-TAUY
 10. 32.6    56.049.68E+00   5.36   5.36 1.0472   0.0461    4.0260   STRESS = 59.65(OMEGA )** 0.391  R = 0.9785  48.76
 20. 51.1    87.961.94E+01  10.72  10.72 2.0944   0.7393    4.4769   STRESS = 25.01(KEMGAM)** 0.391  R = 0.9785  80.68
 50. 68.7   118.204.84E+01  26.81  26.79 5.2360   1.6556    4.7724   STRESS = 31.51(BROGAM)** 0.391  R = 0.9785 110.92
100. 82.2   141.319.68E+01  53.61  53.5810.4720   2.3487    4.9510   STRESS = 31.51(ALVGAM)** 0.391  R = 0.9785 134.04

K = 31.51   N = 0.391   KYP = 38.36   NYP = 0.317   RYP = 0.9978
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Fig.5.2.1.1,p.12

SYSTEM PROPERTIES

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MINERAL --- SSI METHACOAL  (59.2%)             3-12-77
AVERAGE SOLID          SPECIFIC GRAVITY (S )  --- 1.429
LIQUID PHASE           SPECIFIC GRAVITY (SL)  --- 0.791
SLURRY                 SPECIFIC GRAVITY (SM)  --- 1.076
SLURRY CONCENTRATION BY WEIGHT                --- 0.592
SLURRY CONCENTRATION BY VOLUME                --- 0.446
ABSOLUTE PIPEWALL ROUGHNESS (E), FEET         --- 0.00015000
GRAVITATIONAL ACCELERATION                    --- 32.1573 FEET/SEC/SEC
SLURRY TEMPERATURE (TEMP), DEGREES CELSIUS    --- 25.9
PIPE TYPE                                     ---
PIPE SLOPE                                    --- HORIZONTAL


   MESH        PERCENT

0.007/100        0.30
  100/200       13.70
  200/325       65.60
  325/PAN       20.40
 TOTAL =   100.0

WEIGHTED MEAN DIAMETER =5.8349E-02 MM       THROUGHPUT (TONS/HR) =  SHORT
COEFFICIENT OF VARIATION =    44.68801      SETTLING REGIME = LAMINAR
DRAG COEFF OF WEIGHTED MEAN DIA = 215.939   VISCOSITY FACTOR =  1.0
REYNOLDS NUMBER OF SETTLING =     0.11
ROSIN - RAMMLER EQUATION:  R = 100 * EXP(-(D/  0.068649) **  2.585615)
SLOPE =  2.585615     INTERCEPT B =  0.06864908340  MILLIMETERS
CORRELATION COEFF. = 0.965928     D50 =  0.05 MILLIMETERS

KYP =  38.36      NYP =  0.317      TAUY =    7.275
SM THEORY

  PH = N.A.
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Fig.5.2.1.1,p.13

DIA IN.	RELRUF E/DIA	VCL FPS	VCT FPS	VLCTY FPS	THROUGHPUT TONS/HR	REYNOLDS NUMBER	FM MIX	PSI MI	KILW-HR TON-MI
6.000	0.00030	7.22	10.58	4.00	56.1	778.	.0000	106.102	0.2902
8.000	0.00022	6.83	10.02	4.00	99.7	852.	.0000	72.981	0.1996
10.000	0.00018	6.55	9.61	4.00	155.7	915.	.0000	54.592	0.1493
12.000	0.00015	6.33	9.29	4.00	224.2	969.	.0000	43.100	0.1179
14.000	0.00013	6.15	9.02	4.00	305.2	1018.	.0000	35.284	0.0965
16.000	0.00011	6.00	8.80	4.00	398.6	1062.	.0000	29.687	0.0812
18.000	0.00010	5.87	8.60	4.00	504.5	1102.	.0000	25.492	0.0697
20.000	0.00009	5.75	8.43	4.00	622.8	1140.	.0000	22.217	0.0608
22.000	0.00008	5.65	8.28	4.00	753.6	1175.	.0000	19.662	0.0538
24.000	0.00007	5.56	8.15	4.00	896.9	1208.	.0000	17.547	0.0480
6.000	0.00030	7.22	10.58	5.00	70.1	1133.	.0000	113.458	0.3104
8.000	0.00022	6.83	10.02	5.00	124.6	1241.	.0000	78.028	0.2134
10.000	0.00018	6.55	9.61	5.00	194.6	1332.	.0000	58.385	0.1597
12.000	0.00015	6.33	9.29	5.00	280.3	1411.	.0000	46.077	0.1260
14.000	0.00013	6.15	9.02	5.00	381.5	1482.	.0000	37.713	0.1032
16.000	0.00011	6.00	8.80	5.00	498.3	1546.	.0000	31.721	0.0868
18.000	0.00010	5.87	8.60	5.00	630.6	1605.	.0000	27.229	0.0745
20.000	0.00009	5.75	8.43	5.00	778.6	1659.	.0000	23.750	0.0650
22.000	0.00008	5.65	8.28	5.00	942.1	1710.	.0000	20.979	0.0574
24.000	0.00007	5.56	8.15	5.00	1121.1	1758.	.0000	18.752	0.0513
6.000	0.00030	7.22	10.58	6.00	84.1	1540.	.0000	119.915	0.3280
8.000	0.00022	6.83	10.02	6.00	149.5	1687.	.0000	82.445	0.2255
10.000	0.00018	6.55	9.61	6.00	233.6	1810.	.0000	61.655	0.1687
12.000	0.00015	6.33	9.29	6.00	336.3	1918.	.0000	48.654	0.1331
14.000	0.00013	6.15	9.02	6.00	457.8	2014.	.0000	39.826	0.1089
16.000	0.00011	6.00	8.80	6.00	597.9	2409.	.0467	48.282	0.1321
18.000	0.00010	5.87	8.60	6.00	756.8	2500.	.0462	42.408	0.1160
20.000	0.00009	5.75	8.43	6.00	934.3	2585.	.0457	37.764	0.1033
22.000	0.00008	5.65	8.28	6.00	1130.5	2665.	.0452	34.004	0.0930
24.000	0.00007	5.56	8.15	6.00	1345.3	2739.	.0448	30.921	0.0845
6.000	0.00030	7.22	10.58	7.00	98.1	1996.	.0000	125.649	0.3437
8.000	0.00022	6.83	10.02	7.00	174.4	2506.	.0462	130.073	0.3558
10.000	0.00018	6.55	9.61	7.00	272.5	2690.	.0452	101.716	0.2782
12.000	0.00015	6.33	9.29	7.00	392.4	2850.	.0444	83.220	0.2276
14.000	0.00013	6.15	9.02	7.00	534.1	2993.	.0437	70.242	0.1921
16.000	0.00011	6.00	8.80	7.00	697.6	3122.	.0431	60.655	0.1659
18.000	0.00010	5.87	8.60	7.00	882.9	3241.	.0426	53.294	0.1458
20.000	0.00009	5.75	8.43	7.00	1090.0	3351.	.0422	47.474	0.1299
22.000	0.00008	5.65	8.28	7.00	1318.9	3454.	.0418	42.760	0.1170
24.000	0.00007	5.56	8.15	7.00	1569.6	3551.	.0414	38.868	0.1063
6.000	0.00030	7.22	10.58	8.00	112.1	2864.	.0444	217.707	0.5955
8.000	0.00022	6.83	10.02	8.00	199.3	3138.	.0431	158.584	0.4338
10.000	0.00018	6.55	9.61	8.00	311.4	3368.	.0422	124.077	0.3394
12.000	0.00015	6.33	9.29	8.00	448.4	3568.	.0414	101.559	0.2778
14.000	0.00013	6.15	9.02	8.00	610.4	3747.	.0408	85.753	0.2346
16.000	0.00011	6.00	8.80	8.00	797.2	3909.	.0403	74.073	0.2026
18.000	0.00010	5.87	8.60	8.00	1009.0	4058.	.0399	65.103	0.1781
20.000	0.00009	5.75	8.43	8.00	1245.7	4195.	.0395	58.007	0.1587
22.000	0.00008	5.65	8.28	8.00	1507.3	4324.	.0391	52.259	0.1430

Fig.5.2.1.1,p.14

DIA IN.	RELRUF E/DIA	VCL FPS	VCT FPS	VLCTY FPS	THROUGHPUT TONS/HR	REYNOLDS NUMBER	FM MIX	PSI MI	KILW-HR TON-MI
24.000	0.00007	5.56	8.15	8.00	1793.8	4445.	.0388	47.513	0.1300
6.000	0.00030	7.22	10.58	9.00	126.1	3492.	.0419	259.615	0.7102
8.000	0.00022	6.83	10.02	9.00	224.2	3826.	.0407	189.211	0.5176
10.000	0.00018	6.55	9.61	9.00	350.4	4106.	.0398	148.122	0.4051
12.000	0.00015	6.33	9.29	9.00	504.5	4350.	.0391	121.266	0.3317
14.000	0.00013	6.15	9.02	9.00	686.7	4568.	.0385	102.424	0.2802
16.000	0.00011	6.00	8.80	9.00	896.9	4766.	.0380	88.496	0.2421
18.000	0.00010	5.87	8.60	9.00	1135.1	4947.	.0376	77.798	0.2128
20.000	0.00009	5.75	8.43	9.00	1401.4	5115.	.0373	69.333	0.1897
22.000	0.00008	5.65	8.28	9.00	1695.7	5272.	.0369	62.475	0.1709
24.000	0.00007	5.56	8.15	9.00	2018.0	5420.	.0366	56.810	0.1554
6.000	0.00030	7.22	10.58	10.00	140.1	4170.	.0397	304.322	0.8325
8.000	0.00022	6.83	10.02	10.00	249.1	4568.	.0386	221.888	0.6070
10.000	0.00018	6.55	9.61	10.00	389.3	4903.	.0378	173.738	0.4753
12.000	0.00015	6.33	9.29	10.00	560.5	5194.	.0372	142.299	0.3893
14.000	0.00013	6.15	9.02	10.00	763.0	5455.	.0366	120.218	0.3289
16.000	0.00011	6.00	8.80	10.00	996.5	5691.	.0362	103.893	0.2842
18.000	0.00010	5.87	8.60	10.00	1261.3	5907.	.0358	91.351	0.2499
20.000	0.00009	5.75	8.43	10.00	1557.1	6108.	.0354	81.425	0.2227
22.000	0.00008	5.65	8.28	10.00	1884.1	6295.	.0351	73.383	0.2007
24.000	0.00007	5.56	8.15	10.00	2242.2	6471.	.0349	66.739	0.1826
6.000	0.00030	7.22	10.58	11.00	154.2	4895.	.0380	351.753	0.9622
8.000	0.00022	6.83	10.02	11.00	274.1	5363.	.0369	256.558	0.7018
10.000	0.00018	6.55	9.61	11.00	428.2	5756.	.0361	200.941	0.5497
12.000	0.00015	6.33	9.29	11.00	616.6	6098.	.0355	164.618	0.4503
14.000	0.00013	6.15	9.02	11.00	839.3	6404.	.0350	139.103	0.3805
16.000	0.00011	6.00	8.80	11.00	1096.2	6681.	.0346	120.235	0.3289
18.000	0.00010	5.87	8.60	11.00	1387.4	6935.	.0342	105.737	0.2892
20.000	0.00009	5.75	8.43	11.00	1712.8	7170.	.0339	94.262	0.2579
22.000	0.00008	5.65	8.28	11.00	2072.5	7390.	.0336	84.963	0.2324
24.000	0.00007	5.56	8.15	11.00	2466.5	7597.	.0334	77.280	0.2114
6.000	0.00030	7.22	10.58	12.00	168.2	5667.	.0364	401.845	1.0992
8.000	0.00022	6.83	10.02	12.00	299.0	6208.	.0354	293.174	0.8020
10.000	0.00018	6.55	9.61	12.00	467.1	6663.	.0347	229.672	0.6283
12.000	0.00015	6.33	9.29	12.00	672.7	7060.	.0341	188.193	0.5148
14.000	0.00013	6.15	9.02	12.00	915.6	7414.	.0337	159.052	0.4351
16.000	0.00011	6.00	8.80	12.00	1195.9	7734.	.0333	137.499	0.3761
18.000	0.00010	5.87	8.60	12.00	1513.5	8028.	.0329	120.936	0.3308
20.000	0.00009	5.75	8.43	12.00	1868.5	8301.	.0326	107.825	0.2950
22.000	0.00008	5.65	8.28	12.00	2260.9	8556.	.0323	97.199	0.2659
24.000	0.00007	5.56	8.15	12.00	2690.7	8795.	.0321	88.419	0.2419
6.000	0.00030	7.22	10.58	13.00	182.2	6484.	.0351	454.543	1.2434
8.000	0.00022	6.83	10.02	13.00	323.9	7104.	.0342	331.694	0.9073
10.000	0.00018	6.55	9.61	13.00	506.1	7624.	.0335	259.898	0.7109
12.000	0.00015	6.33	9.29	13.00	728.7	8078.	.0329	212.995	0.5826
14.000	0.00013	6.15	9.02	13.00	991.9	8483.	.0325	180.040	0.4925
16.000	0.00011	6.00	8.80	13.00	1295.5	8849.	.0321	155.663	0.4258
18.000	0.00010	5.87	8.60	13.00	1639.6	9186.	.0317	136.928	0.3746
20.000	0.00009	5.75	8.43	13.00	2024.2	9498.	.0315	122.126	0.3341

Fig.5.2.1.1,p.15

DIA IN.	RELRUF E/DIA	VCL FPS	VCT FPS	VLCTY FPS	THROUGHPUT TONS/HR	REYNOLDS NUMBER	FM MIX	PSI MI	KILW-HR TON-MI
22.000	0.00008	5.65	8.28	13.00	2449.3	9790.	.0312	110.100	0.3012
24.000	0.00007	5.56	8.15	13.00	2914.9	10063.	.0310	100.163	0.2740
6.000	0.00030	7.22	10.58	14.00	196.2	7346.	.0340	509.801	1.3945
8.000	0.00022	6.83	10.02	14.00	348.8	8047.	.0331	372.002	1.0170
10.000	0.00018	6.55	9.61	14.00	545.0	8637.	.0324	291.590	0.7976
12.000	0.00015	6.33	9.29	14.00	784.8	9151.	.0318	239.001	0.6538
14.000	0.00013	6.15	9.02	14.00	1068.2	9609.	.0314	202.094	0.5528
16.000	0.00011	6.00	8.80	14.00	1395.2	10025.	.0310	174.749	0.4780
18.000	0.00010	5.87	8.60	14.00	1765.8	10406.	.0307	153.731	0.4205
20.000	0.00009	5.75	8.43	14.00	2180.0	10760.	.0304	137.091	0.3750
22.000	0.00008	5.65	8.28	14.00	2637.8	11090.	.0302	123.602	0.3381
24.000	0.00007	5.56	8.15	14.00	3139.1	11400.	.0300	112.455	0.3076
6.000	0.00030	7.22	10.58	15.00	210.2	8250.	.0329	567.575	1.5526
8.000	0.00022	6.83	10.02	15.00	373.7	9038.	.0321	414.306	1.1333
10.000	0.00018	6.55	9.61	15.00	583.9	9700.	.0314	324.797	0.8885
12.000	0.00015	6.33	9.29	15.00	840.8	10278.	.0309	266.246	0.7283
14.000	0.00013	6.15	9.02	15.00	1144.5	10792.	.0305	225.100	0.6157
16.000	0.00011	6.00	8.80	15.00	1494.8	11259.	.0301	194.660	0.5325
18.000	0.00010	5.87	8.60	15.00	1891.9	11688.	.0298	171.263	0.4685
20.000	0.00009	5.75	8.43	15.00	2335.7	12085.	.0295	152.737	0.4178
22.000	0.00008	5.65	8.28	15.00	2826.2	12455.	.0293	137.719	0.3767
24.000	0.00007	5.56	8.15	15.00	3363.4	12804.	.0291	125.308	0.3428
6.000	0.00030	7.22	10.58	16.00	224.2	9197.	.0320	627.829	1.7174
8.000	0.00022	6.83	10.02	16.00	398.6	10075.	.0312	458.441	1.2540
10.000	0.00018	6.55	9.61	16.00	622.8	10813.	.0306	359.344	0.9830
12.000	0.00015	6.33	9.29	16.00	896.9	11457.	.0301	294.593	0.8058
14.000	0.00013	6.15	9.02	16.00	1220.8	12031.	.0296	249.088	0.6814
16.000	0.00011	6.00	8.80	16.00	1594.5	12551.	.0293	215.422	0.5893
18.000	0.00010	5.87	8.60	16.00	2018.0	13029.	.0290	189.543	0.5185
20.000	0.00009	5.75	8.43	16.00	2491.4	13471.	.0287	169.052	0.4624
22.000	0.00008	5.65	8.28	16.00	3014.6	13884.	.0285	152.440	0.4170
24.000	0.00007	5.56	8.15	16.00	3587.6	14273.	.0283	138.710	0.3794

FOUR DATA POINTS

SLURRY: SSI METHACOAL (54.6%) 3-12-77

I	RC(CM)	RS(CM)	RAT(I)	LN(RAT(I))	CW(%)	CV(%)	TEMP(C)	S	SL	TAUK(I)	SM	YIELD(DYNES/CM/CM)
1	4.0000	0.9421	4.2458	1.4459	54.6	39.9	26.0	1.429	0.791	0.5814	1.046	2.32

RPM	DIAL	STRESS	KEMGAM	BROGAM	ALVGAM	OMEGA	LN(OMEGA)	LN(STRESS)			TAU-TAUY
10.	15.2	26.11	3.06E+04	12.47	12.47	1.0472	0.0461	3.2623	STRESS = 25.46(OMEGA)** 0.168	R = 0.9932	23.79
20.	16.5	28.33	6.13E+04	24.95	24.95	2.0944	0.7393	3.3439	STRESS = 4.53(KEMGAM)** 0.168	R = 0.9932	26.01
50.	19.2	33.06	1.53E+05	62.37	62.37	5.2360	1.6556	3.4983	STRESS = 16.80(BROGAM)** 0.168	R = 0.9932	30.74
100.	22.3	38.42	3.06E+05	124.73	124.73	10.4720	2.3487	3.6487	STRESS = 16.80(ALVGAM)** 0.168	R = 0.9932	36.10

K = 16.80 N = 0.168 KYP = 14.78 NYP = 0.181 RYP = 0.9937

Fig.5.2.1.1,p.17

SYSTEM PROPERTIES

```
MINERAL --- SSI METHACOAL  (54.6%)          3-12-77
AVERAGE SOLID         SPECIFIC GRAVITY (S ) --- 1.429
LIQUID PHASE          SPECIFIC GRAVITY (SL) --- 0.791
SLURRY                SPECIFIC GRAVITY (SM) --- 1.046
SLURRY CONCENTRATION BY WEIGHT              --- 0.546
SLURRY CONCENTRATION BY VOLUME              --- 0.399
ABSOLUTE PIPEWALL ROUGHNESS (E), FEET       --- 0.00015000
GRAVITATIONAL ACCELERATION                  --- 32.1573 FEET/SEC/SEC
SLURRY TEMPERATURE (TEMP), DEGREES CELSIUS  --- 26.0
PIPE TYPE                                   ---
PIPE SLOPE                                  --- HORIZONTAL
```

MESH	PERCENT
0.007/100	0.30
100/200	13.70
200/325	65.60
325/PAN	20.40
TOTAL =	100.0

```
WEIGHTED MEAN DIAMETER =5.8349E-02 MM       THROUGHPUT (TONS/HR) =  SHORT
COEFFICIENT OF VARIATION =    44.68801      SETTLING REGIME = LAMINAR
DRAG COEFF OF WEIGHTED MEAN DIA = 215.939   VISCOSITY FACTOR =  1.0
REYNOLDS NUMBER OF SETTLING =     0.11
ROSIN - RAMMLER EQUATION!  R = 100 * EXP(-(D/  0.068649) **  2.585615)
SLOPE =  2.585615      INTERCEPT B =  0.06864908340  MILLIMETERS
CORRELATION COEFF. = 0.965528       D50 =  0.06 MILLIMETERS

KYP =  14.78      NYP =  0.181      TAUY =     2.322
SM THEORY

  PH = N.A.
```

Fig.5.2.1.1,p.18

DIA IN.	RELRUF E/DIA	VCL FPS	VCT FPS	VLCTY FPS	THROUGHPUT TONS/HR	REYNOLDS NUMBER	FM MIX	PSI MI	KILW-HR TON-MI
6.000	0.00030	3.04	4.34	4.00	50.2	3958.	.0403	48.069	0.1468
8.000	0.00022	2.96	4.22	4.00	89.3	4170.	.0397	35.442	0.1082
10.000	0.00018	2.89	4.12	4.00	139.5	4342.	.0392	27.990	0.0855
12.000	0.00015	2.84	4.05	4.00	200.8	4488.	.0388	23.084	0.0705
14.000	0.00013	2.80	3.99	4.00	273.4	4615.	.0384	19.616	0.0599
16.000	0.00011	2.76	3.93	4.00	357.1	4728.	.0381	17.037	0.0520
18.000	0.00010	2.73	3.89	4.00	451.9	4830.	.0379	15.046	0.0459
20.000	0.00009	2.70	3.85	4.00	557.9	4924.	.0377	13.464	0.0411
22.000	0.00008	2.67	3.81	4.00	675.1	5009.	.0375	12.176	0.0372
24.000	0.00007	2.65	3.78	4.00	803.4	5089.	.0373	11.109	0.0339
6.000	0.00030	3.04	4.34	5.00	62.8	5939.	.0360	66.971	0.2045
8.000	0.00022	2.96	4.22	5.00	111.6	6257.	.0354	49.388	0.1508
10.000	0.00018	2.89	4.12	5.00	174.3	6515.	.0349	39.013	0.1191
12.000	0.00015	2.84	4.05	5.00	251.1	6734.	.0346	32.183	0.0983
14.000	0.00013	2.80	3.99	5.00	341.7	6925.	.0343	27.353	0.0835
16.000	0.00011	2.76	3.93	5.00	446.3	7095.	.0340	23.762	0.0726
18.000	0.00010	2.73	3.89	5.00	564.9	7248.	.0338	20.989	0.0641
20.000	0.00009	2.70	3.85	5.00	697.4	7388.	.0336	18.785	0.0574
22.000	0.00008	2.67	3.81	5.00	843.8	7517.	.0335	16.992	0.0519
24.000	0.00007	2.65	3.78	5.00	1004.2	7636.	.0333	15.505	0.0473
6.000	0.00030	3.04	4.34	6.00	75.3	8274.	.0329	88.241	0.2695
8.000	0.00022	2.96	4.22	6.00	133.9	8717.	.0324	65.072	0.1987
10.000	0.00018	2.89	4.12	6.00	209.2	9077.	.0320	51.405	0.1570
12.000	0.00015	2.84	4.05	6.00	301.3	9382.	.0316	42.409	0.1295
14.000	0.00013	2.80	3.99	6.00	410.1	9648.	.0314	36.058	0.1101
16.000	0.00011	2.76	3.93	6.00	535.6	9885.	.0312	31.327	0.0957
18.000	0.00010	2.73	3.89	6.00	677.9	10098.	.0310	27.674	0.0845
20.000	0.00009	2.70	3.85	6.00	836.9	10293.	.0308	24.770	0.0756
22.000	0.00008	2.67	3.81	6.00	1012.6	10472.	.0306	22.407	0.0684
24.000	0.00007	2.65	3.78	6.00	1205.1	10639.	.0305	20.449	0.0624
6.000	0.00030	3.04	4.34	7.00	87.9	10951.	.0306	111.810	0.3415
8.000	0.00022	2.96	4.22	7.00	156.2	11538.	.0301	82.437	0.2517
10.000	0.00018	2.89	4.12	7.00	244.1	12014.	.0297	65.118	0.1989
12.000	0.00015	2.84	4.05	7.00	351.5	12418.	.0294	53.723	0.1641
14.000	0.00013	2.80	3.99	7.00	478.4	12770.	.0292	45.667	0.1395
16.000	0.00011	2.76	3.93	7.00	624.9	13083.	.0290	39.677	0.1212
18.000	0.00010	2.73	3.89	7.00	790.8	13366.	.0288	35.052	0.1070
20.000	0.00009	2.70	3.85	7.00	976.3	13623.	.0287	31.376	0.0958
22.000	0.00008	2.67	3.81	7.00	1181.4	13861.	.0285	28.305	0.0867
24.000	0.00007	2.65	3.78	7.00	1405.9	14081.	.0284	25.905	0.0791
6.000	0.00030	3.04	4.34	8.00	100.4	13961.	.0289	137.563	0.4201
8.000	0.00022	2.96	4.22	8.00	178.5	14709.	.0284	101.395	0.3096
10.000	0.00018	2.89	4.12	8.00	279.0	15317.	.0280	80.083	0.2446
12.000	0.00015	2.84	4.05	8.00	401.7	15832.	.0277	66.064	0.2018
14.000	0.00013	2.80	3.99	8.00	546.7	16280.	.0275	56.171	0.1715
16.000	0.00011	2.76	3.93	8.00	714.1	16679.	.0273	48.802	0.1490
18.000	0.00010	2.73	3.89	8.00	903.8	17039.	.0271	43.114	0.1317
20.000	0.00009	2.70	3.85	8.00	1115.8	17368.	.0270	38.593	0.1179
22.000	0.00008	2.67	3.81	8.00	1350.1	17671.	.0269	34.914	0.1066

Fig.5.2.1.1,p.19

DIA IN.	RELRUF E/DIA	VCL FPS	VCT FPS	VLCTY FPS	THROUGHPUT TONS/HR	REYNOLDS NUMBER	FM MIX	PSI MI	KILWHR TON-MI
24.000	0.00007	2.65	3.78	8.00	1606.8	17952.	.0267	31.865	0.0973
6.000	0.00030	3.04	4.34	9.00	113.0	17297.	.0274	165.512	0.5054
8.000	0.00022	2.96	4.22	9.00	200.8	18223.	.0270	121.945	0.3724
10.000	0.00018	2.89	4.12	9.00	313.8	18975.	.0266	96.292	0.2941
12.000	0.00015	2.84	4.05	9.00	451.9	19613.	.0263	79.426	0.2426
14.000	0.00013	2.80	3.99	9.00	615.1	20169.	.0261	67.509	0.2062
16.000	0.00011	2.76	3.93	9.00	803.4	20664.	.0259	58.664	0.1792
18.000	0.00010	2.73	3.89	9.00	1016.8	21110.	.0258	51.824	0.1583
20.000	0.00009	2.70	3.85	9.00	1255.3	21517.	.0256	46.387	0.1417
22.000	0.00008	2.67	3.81	9.00	1518.9	21892.	.0255	41.965	0.1282
24.000	0.00007	2.65	3.78	9.00	1807.6	22240.	.0254	38.308	0.1170
6.000	0.00030	3.04	4.34	10.00	125.5	20950.	.0263	195.530	0.5971
8.000	0.00022	2.96	4.22	10.00	223.2	22072.	.0258	144.025	0.4398
10.000	0.00018	2.89	4.12	10.00	348.7	22983.	.0255	113.717	0.3473
12.000	0.00015	2.84	4.05	10.00	502.1	23756.	.0252	93.787	0.2864
14.000	0.00013	2.80	3.99	10.00	683.4	24429.	.0250	79.666	0.2433
16.000	0.00011	2.76	3.93	10.00	892.7	25028.	.0248	69.209	0.2114
18.000	0.00010	2.73	3.89	10.00	1129.8	25568.	.0246	61.131	0.1867
20.000	0.00009	2.70	3.85	10.00	1394.8	26062.	.0245	54.728	0.1671
22.000	0.00008	2.67	3.81	10.00	1687.7	26516.	.0244	49.510	0.1512
24.000	0.00007	2.65	3.78	10.00	2008.5	26938.	.0243	45.185	0.1380
6.000	0.00030	3.04	4.34	11.00	138.1	24915.	.0253	227.681	0.6953
8.000	0.00022	2.96	4.22	11.00	245.5	26249.	.0248	167.493	0.5115
10.000	0.00018	2.89	4.12	11.00	383.6	27333.	.0245	132.225	0.4038
12.000	0.00015	2.84	4.05	11.00	552.3	28252.	.0242	109.022	0.3329
14.000	0.00013	2.80	3.99	11.00	751.8	29053.	.0240	92.640	0.2829
16.000	0.00011	2.76	3.93	11.00	981.9	29765.	.0238	80.494	0.2458
18.000	0.00010	2.73	3.89	11.00	1242.7	30407.	.0237	71.104	0.2171
20.000	0.00009	2.70	3.85	11.00	1534.2	30994.	.0235	63.643	0.1944
22.000	0.00008	2.67	3.81	11.00	1856.4	31534.	.0234	57.575	0.1758
24.000	0.00007	2.65	3.78	11.00	2209.3	32036.	.0233	52.545	0.1605
6.000	0.00030	3.04	4.34	12.00	150.6	29186.	.0244	261.774	0.7994
8.000	0.00022	2.96	4.22	12.00	267.8	30749.	.0240	192.614	0.5882
10.000	0.00018	2.89	4.12	12.00	418.4	32019.	.0236	151.966	0.4641
12.000	0.00015	2.84	4.05	12.00	602.5	33096.	.0234	125.281	0.3826
14.000	0.00013	2.80	3.99	12.00	820.1	34034.	.0232	106.448	0.3251
16.000	0.00011	2.76	3.93	12.00	1071.2	34868.	.0230	92.459	0.2824
18.000	0.00010	2.73	3.89	12.00	1355.7	35621.	.0228	81.667	0.2494
20.000	0.00009	2.70	3.85	12.00	1673.7	36308.	.0227	73.094	0.2232
22.000	0.00008	2.67	3.81	12.00	2025.2	36941.	.0226	66.123	0.2019
24.000	0.00007	2.65	3.78	12.00	2410.2	37529.	.0225	60.362	0.1843
6.000	0.00030	3.04	4.34	13.00	163.2	33760.	.0237	298.035	0.9102
8.000	0.00022	2.96	4.22	13.00	290.1	35568.	.0232	219.142	0.6692
10.000	0.00018	2.89	4.12	13.00	453.3	37037.	.0229	172.837	0.5278
12.000	0.00015	2.84	4.05	13.00	652.8	38282.	.0226	142.493	0.4352
14.000	0.00013	2.80	3.99	13.00	888.5	39367.	.0224	121.056	0.3697
16.000	0.00011	2.76	3.93	13.00	1160.4	40332.	.0223	105.138	0.3211
18.000	0.00010	2.73	3.89	13.00	1468.7	41203.	.0221	92.860	0.2836
20.000	0.00009	2.70	3.85	13.00	1813.2	41997.	.0220	83.108	0.2538

Fig.5.2.1.1,p.20

DIA IN.	RELRUF E/DIA	VCL FPS	VCT FPS	VLCTY FPS	THROUGHPUT TONS/HR	REYNOLDS NUMBER	FM MIX	PSI MI	KILW-HR TON-MI
22.000	0.00008	2.67	3.81	13.00	2194.0	42730.	.0219	75.179	0.2296
24.000	0.00007	2.65	3.78	13.00	2611.0	43409.	.0218	68.608	0.2095
6.000	0.00030	3.04	4.34	14.00	175.7	38631.	.0230	336.301	1.0270
8.000	0.00022	2.96	4.22	14.00	312.4	40699.	.0226	247.198	0.7549
10.000	0.00018	2.89	4.12	14.00	488.2	42380.	.0223	194.902	0.5952
12.000	0.00015	2.84	4.05	14.00	703.0	43805.	.0220	160.608	0.4905
14.000	0.00013	2.80	3.99	14.00	956.8	45047.	.0218	136.423	0.4166
16.000	0.00011	2.76	3.93	14.00	1249.7	46151.	.0216	118.469	0.3618
18.000	0.00010	2.73	3.89	14.00	1581.7	47147.	.0215	104.625	0.3195
20.000	0.00009	2.70	3.85	14.00	1952.7	48057.	.0214	93.631	0.2859
22.000	0.00008	2.67	3.81	14.00	2362.7	48895.	.0213	84.693	0.2586
24.000	0.00007	2.65	3.78	14.00	2811.9	49672.	.0212	77.286	0.2360
6.000	0.00030	3.04	4.34	15.00	188.3	43795.	.0225	376.699	1.1504
8.000	0.00022	2.96	4.22	15.00	334.7	46140.	.0220	276.668	0.8449
10.000	0.00018	2.89	4.12	15.00	523.0	48046.	.0217	218.056	0.6659
12.000	0.00015	2.84	4.05	15.00	753.2	49661.	.0214	179.642	0.5486
14.000	0.00013	2.80	3.99	15.00	1025.2	51069.	.0212	152.562	0.4659
16.000	0.00011	2.76	3.93	15.00	1339.0	52321.	.0211	132.467	0.4045
18.000	0.00010	2.73	3.89	15.00	1694.6	53450.	.0209	116.975	0.3572
20.000	0.00009	2.70	3.85	15.00	2092.2	54481.	.0208	104.674	0.3197
22.000	0.00008	2.67	3.81	15.00	2531.5	55431.	.0207	94.676	0.2891
24.000	0.00007	2.65	3.78	15.00	3012.7	56313.	.0206	86.392	0.2638
6.000	0.00030	3.04	4.34	16.00	200.8	49249.	.0220	419.059	1.2797
8.000	0.00022	2.96	4.22	16.00	357.1	51887.	.0215	307.588	0.9393
10.000	0.00018	2.89	4.12	16.00	557.9	54029.	.0212	242.330	0.7400
12.000	0.00015	2.84	4.05	16.00	803.4	55846.	.0209	199.587	0.6095
14.000	0.00013	2.80	3.99	16.00	1093.5	57429.	.0207	169.468	0.5175
16.000	0.00011	2.76	3.93	16.00	1428.2	58837.	.0206	147.124	0.4493
18.000	0.00010	2.73	3.89	16.00	1807.6	60107.	.0204	129.903	0.3967
20.000	0.00009	2.70	3.85	16.00	2231.6	61266.	.0203	116.233	0.3550
22.000	0.00008	2.67	3.81	16.00	2700.3	62335.	.0202	105.123	0.3210
24.000	0.00007	2.65	3.78	16.00	3213.5	63326.	.0201	95.920	0.2929

Fig.5.2.1.1,p.21

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SPOOLER RUNTIME 8 SECONDS, 27 KCS, 21 PAGES
**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**

**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**

**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**

**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**

**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**

**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**

**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**

**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**

**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**

**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**

**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**

**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**

**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**

**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**

**END** USER ORS 352 [1213,254] JOB  FOR09 SEQ. 8032 DATE 15-MAR-77 16:59:47 MONITOR CSM DECSYSTEM-10 602A7 **END**
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Fig.5.2.1.1,p.22

Screen Analysis

+100 mesh	0.3%
+200	13.7
+325	65.60
-325	20.4
	100.0%

Analysis of Coal

	As received	Dry basis	Moisture & ash-free basis
Volatile matter	37.4%	40.2%	47.5
Fixed carbon	41.4%	44.5%	52.5
Ash	14.2%	15.3%	-
Btu/lb	11,355	12,210	14,420
Sulfur	0.7%	0.7%	-

Rheological data are listed on the computer printouts. Four concentrations were measured, two at 64% solids concentration by weight, and one each at 60% and 55%. It was concluded that higher concentrations, possibly above 70% by weight, could probably be pumped, although of course pipewall friction would be high.

Nomenclature is given in Fig. 5.2.2.1-1. Along with the rheological data are included the predicted pressure drops to overcome pipewall friction in psi per mile. The specific power (kilowatts to transport one dry short ton of coal per hour one mile) is listed also. In both cases, pressure drop and specific power are for pipewall friction only and do not include pipeline bends, elevation differences, fitting losses, or pump motor efficiencies.

As a point of comparison, the Black Mesa pipeline transports 660 dry tons of coal per hour at 48% solids by weight at less than 6 fps in an 18" diameter pipe. Pressure drop is approximately 21 psi/mile. The closest predicted methanol data point is 27.6 psi/mile for 675 tph at 6 fps and a solids concentration of 55% by weight. Thus, the methacoal slurry does not appear to be substantially more viscous than a coal-water slurry. It does of course require considerably more power to move the greater amount of payload.

A range of pressure drops for pipe inside-diameters from 6" to 24" in 2" increments is listed in Fig. 5.2.1.1-1 for velocities from 4 to 16 fps in 1-fps increments. Pipe roughness is assumed to be that of new commercial steel (0.00015 ft).

The slurry is essentially a nonsettling or homogeneous slurry for minus-100 mesh coal but it is on verge of settling. Thus a coarser particle size is likely to produce a settling slurry. Thus, the experiment has produced two very significant pieces of information, both of which are important in the economics of slurry pipelining - carrying capacity and coarseness limit.

5.2.1.2 Importance of Carrying Capacity

The carrying capacity of the coal is important for the several reasons which are discussed below.

5.2.1.2.1 Reduction of Methanol Conversion Loss

The greater the carrying capacity of the methanol, the smaller is the total system energy loss to the methanol conversion process. Coal can be converted to methanol at only 40 to 50% efficiency. However, if the slurry itself is recognized as a fuel and compared to pure methanol, then the conversion loss is only taken on the portion converted to methanol, and that loss can be spread over the entire mixture.

Consider, for example, a slurry which is two-thirds coal and one-third methanol, the coal being 60% carbon and 30% moisture. Then a ton of coal yields about 3200 lb of methanol, which in turn transports 6400 lb of coal. The loss of the additional ton of coal that is consumed in the conversion process can now be charged to the full 9600 lb of fuel delivered to the power plant. The overall conversion efficiency of the process is

$$\frac{9600}{9600 + 2000} \times 100 = 83\% \ .$$

If the carrying ratio is increased from two to three, the overall conversion efficiency increases to 86%.

This conclusion is sufficiently important to merit repetition. While the direct conversion efficiency of coal to methanol is at most 50%, the equivalent efficiency of conversion for the entire system potentially approaches 90%. It will be seen in Section 5.2.1.3.2 below that this efficiency can be further increased, possibly exceeding 90%.

5.2.1.2.2 Reduction of System Water Requirement

The ton of coal, which in the above example is converted to methanol, requires about 3000 lb of water for conversion, if the natural moisture in the coal can be used in the process, and yields about 3200 lb of methanol. The carrying capacity of this 3000 lb of water, after its conversion to methanol, may be compared with the carrying capacity of an equal amount of water at Black Mesa, where 52 lb of water carries 48 lb of coal. For a slurry which is two-thirds coal, the 3200 lb of methanol carries 6400 lb of coal. But the 3200 lb of methanol itself is equivalent, on a heating-value basis, to approximately the same amount of coal, so that the equivalent of 6400 + 3200 = 9600 lb of coal is carried by 3200 lb of methanol. At Black Mesa, the same 3000 lb of water entering the system carries

$$3000 \times \frac{48}{52} = 2769 \text{ lb of coal.}$$

Now consider the removal of water from the coal-water slurry. In Report R-3022 of this series (see Table 1.1 above), it was seen that at the Mohave Generating Station, 1,176,000 Btu/ton are required to separate the coal and water and dry the coal to the contract moisture value of 10.74%. About 1,420,000 Btu would be required to yield bone-dry coal, which for the coal used in the preceding numerical examples (not the same as Black Mesa coal) would be equivalent to 166 lb of coal consumed.

In the case of the methanol slurry, of the 50% energy lost in conversion, it seems reasonable to expect that some can be used, as low-grade waste heat from the process, to dry the incoming coal. Therefore, it is not necessary to again charge the methanol system with the energy loss in obtaining dry coal.

Thus, for the water system, 2769 lb of coal and 3000 lb of water enter the system, and 2769 - 166 = 2603 lb of dry coal eventually appears in the power plant boiler. The transportation efficiency may then be said to be

$$\frac{2769 - 166}{2769} = 94.0\% \ .$$

With the one-third - two-thirds methanol system, along with the 3000 lb of entering water, there is a ton of coal to be converted, a ton to be lost, and 2x3200 = 6400 lb to be transported. That is, 10,400 lb of raw coal enter and the equivalent of 9600 lb of dry coal emerge, for a transportation efficiency of

$$\frac{9600}{10{,}400} = 92.3\% \ .$$

The advantage of methanol as a slurry carrier may now be portrayed as the ratio of its carrying capacity to that of water, i.e.,

$$\frac{10,400 \times 0.923}{2769 \times 0.940} = 3.7 \ .$$

For the one-fourth - three-fourths methanol system, the advantage is

$$\frac{12,800 \times 0.941}{2769 \times 0.940} = 4.6 \ .$$

The foregoing example was based upon using the natural moisture in the coal, which was taken as 30%, or 600 lb/ton. If none of this moisture is used, the ratios calculated above must be reduced by 3000/3600 = 0.833, becoming 3.1 and 3.9 respectively.

A highly important conclusion emerges. The water requirement for the methanol slurry may be three to four times less than that for the water slurry.

In the foregoing discussion, methanol and coal have been treated as equivalent fuels. For the present illustrative purpose, this approach is adequate because the heat contents are approximately the same. More precise calculations would be done on a heat content basis for coals of a specific proximate analysis and for methanol produced by a specific process from that coal.

Clearly, the water problem for the methanol line is greatly less than for the water line. In particular, the use of a return line becomes very interesting when it is only required to accommodate such a small return fraction. It may be further noted that, once it is decided to install the return line, system flexibility is increased. For example, it is no longer necessary to locate the methanol plant at the head of the line; it can be placed at any location along the line where other factors, e.g., existing labor force, availability of cheap power, proximity to other markets for the methanol, etc., are located. However, as will be seen in Section 5.2.1.3.2 below, there are good reasons for having the methanol plant at the pipeline head end.

5.2.1.3 Benefits of Dry Coal

If the coal entering the slurrifier at the head of the pipeline is completely dry, a number of benefits accrue. Some of these would be realized in the pipeline operation, and some in the power plant. They are discussed below, in reverse order.

5.2.1.3.1 Effects of Moisture upon Power Plant Efficiency

Any moisture that enters the furnace must be evaporated, and the latent heat of vaporization thereby absorbed is denied to the power conversion process. For example, if the coal is 30% moisture as in the foregoing example and as is the case with much Western coal, then approximately 350 Btu/lb of coal input is required to vaporize the moisture, representing an energy loss of about 4% for most Western coal.

Now, it is not possible to avoid taking this loss somewhere between the mine mouth and the power plant stack. In principle, this latent heat, along with that of the water formed during combustion, could be recovered by installation of sufficient preheater surface to cool the stack gas below the boiling point. In practice, this is not done because condensing moisture combines with the sulfur dioxide and other stack gases to form acids. The process is complex, and heat recovery in the preheater is generally limited to about 300°F. However, the less moisture in the gas, the lower the temperature to which the stack gas can be economically cooled. And since 30°F in additional stack gas cooling is worth approximately a 1% increase in efficiency of the power plant, the availability of bone-dry coal should have some value by enabling a small increase in power plant efficiency.

5.2.1.3.2 Effects of Moisture on Pipeline Operation

If one now examines the entire mine-pipeline-power plant complex as an integrated system, it is seen that drying the coal along with the methanol conversion process offers several attractions. First, of course, is the possible increase in power plant efficiency just discussed.

There are three large additional benefits which accrue to the pipeline operation from the use of dry coal. The first benefit derives from the increase in pipeline efficiency because it is no longer necessary to transport the water. For Western coals, which generally have high moisture content, this benefit can be very large. Again using the example of coal which has a 30% moisture content, the efficiency of the pipeline is increased by almost 50% if the coal is dried at the head of the line. If the raw coal is only 20% moisture, the pipeline efficiency is still increased 25% by drying. This increase is realized as a direct percentage increase in the number of Btu transported per Btu consumed in the transportation process.

The second benefit of drying lies in the further reduction of the system water requirement, as was seen in the discussion of water requirements in Section 5.2.1.2. The third benefit from drying the coal lies in the possibility of obtaining the drying energy from the methanol conversion process, as was also assumed in the discussion in Section 5.2.1.2.1 above. As noted there, the methanol conversion process is very inefficient, being estimated at 41% [Bodle '75] to 50% [Burke '75]. Most of this inefficiency appears in the form of waste heat. If this otherwise wasted heat is used to dry the coal to be shipped, then that same amount of energy becomes available as sensible heat to the power plant energy conversion cycle instead of being lost as latent heat in the boiler. It is as though low-grade waste heat from the methanation plant were transported without cost and transformed into high-grade heat at the power plant, in defiance of the second law of thermodynamics.

5.2.2 Methanol Consumption and Marketing Options

At the pipeline terminal, many options are available for realizing the value of the methanol.

1. The slurry may be burned directly as a fuel in power plant boilers.

2. The slurry may be separated into powdered coal and alcohols which, in turn, provide fuel for several applications. The suboptions include:

2.1 Powdered coal, after separation from the slurry, may be used as feed stock for low-BTU gas plants in areas where water for the gasification is available, or for synthetic natural gas plants or ammonia plants.

2.2 The alcohols may be returned, by a second pipe laid alongside the main line, to the head end of the pipeline. As has been seen in Sections 5.2.1.2.1 and 5.2.1.3.2, the return line for methanol only needs a fifth or a sixth of the capacity that would be required for a water slurry.

2.3 The alcohols may be marketed as fuel-grade methanol for stationary engines. The market could include natural gas supplement, replacement for propane or butane, gas turbine fuel, additive to gasoline fuel, or used directly as fuel in engines for automotive and industrial applications.

2.4 The alcohols may be marketed as vehicular fuel. If one looks ahead to the time, early in the next century, when petroleum can no longer supply most of the vehicular fuel requirement, there appear to be two preeminent candidates for liquid, vehicular (ultimate) fuels: methanol and hydrogen. There are, of course, many problems and obstacles to the adoption of either of these, which means that a great deal of research and development will be necessary to bring either concept to fruition. The use of methanol in the pipeline in the nearer term offers the opportunity to find early answers to many of the questions relative to its potential as the ultimate vehicular fuel. That is, the principal objectives of two R&D programs can be accomplished by funding only the smaller, more immediate of the two. Moreover, should a methanol pipeline be built, it would constitute a part of the demonstration program for the ultimate fuel.

2.5 The alcohols separated from slurry may be further separated into the basic constituents for subsequent marketing. These could include: methanol, ethanol, n-propanol, and i-butanol.

3. The slurry can be used directly as pipeline fuel. When burned in a gas turbine with a bottoming engine, the overall efficiency of the pumping process would then be approximately 50% greater than that of the electrically driven prime movers. The direct use of the slurry as prime mover fuel would render the slurry pipeline the most energy-efficient of all coal transportation modes insofar as the consumption of mechanical energy of movement is concerned. When these two factors are combined in a system design and subjected to economic analysis, it may well be that the methanol-coal slurry is overall the most energy-efficient mode of long-distance coal transport.

These simple figures are quoted only to show the promising potential of the concept. As has already been stated, it is strongly recommended that further research be performed. Specific recommendations will be presented in Report SSS-77-R-3026, R&D Recommendations.

5.2.3 Problem Areas

It is necessary to recognize disadvantages compared to other approaches, potential pitfalls, limitations, technological uncertainties, and economic constraints.

The brevity with which these factors are treated here is no indication whatever of their anticipated severity. The future work which is being recommended here should begin with a quantitative, in-depth treatment of these factors. The present purpose is to identify potential opportunities, not to assess them in depth. Accordingly, brief mention will be made of only two particularly sensitive questions.

5.2.3.1 Safety

Present-technology coal slurry pipelines, i.e., water-coal slurry, appear to be far safer than any other mode of long-distance coal transport, partly because the water-coal slurry is not flammable. Methanol-coal slurry, of course, does not possess this attraction. However, since it is still far less flammable than some of the fluids presently moved by commercial pipeline, flammability is certainly not a barrier to the introduction of methanol-coal slurry pipelines. However, the safety implications of this new application must be examined.

5.2.3.2 Environmental Impact

Although it has not been analyzed, the environmental disruption resulting from a methanol-coal slurry pipeline spill is almost certain to be more undesirable than that from a water-coal slurry. However, the consequences appear to be much less undesirable than some fluids which are presently moved all over the country by pipeline. Therefore, it seems unlikely that environmental impact will prove to be a decisive negative factor in the competition. Nevertheless, that impact must be examined.

5.3 Slurry-fired Engines

To utilize coal-water slurry as a fuel for engines, it is customary to first dewater the slurry and then utilize the dry, pulverized coal for firing in the engine. Two different systems have been used in the United States to dewater and dry coal at the terminus of the coal pipeline. The original Consolidation Coal pipeline in Ohio used disk filters followed by flash dryers. The Black Mesa line uses centrifuges mounted directly on top of the pulverizer, and the centrifuge cake is dried in the pulverizer. One other methid is the direct combustion of a concentrated, stabilized coal slurry in a cyclone burner. This was done only experimentally, but the test was successful.

Dewatering facilities generally produce environmental effects similar to those of the coal preparation area and, for the most part are similarly solved. One exception is the disposal of water from the slurry stream. Water treatment may or may not be required, depending on intended use of the water. Normally a coal slurry pipeline can be expected to supply a steam power plant. In this case, effluent from the dewatering facility can be used as a part of the cooling water makeup. If the water is used for boiler feed water, treatment is generally required.

The major problems involved in burning pulverized coal in internal combustion engines will be discussed later in para. 6.1. For the reciprocating engines they would primarily involve metering, combustion, deposits, and wear. For open cycle gas turbines, hot corrosion and erosion resulting from high sulfur and ash content of most coals are the principal problems.

The severity of the problems is dependent to some extent on the source and analysis of the coal used. There is a wide variety of coal mined in the United States (see Table 5.3-1). Fixed carbon ranges from 40% to 96%, calorific values from 9300 to 15,700 Btu/lb and ash content from 4% to 22%. Some of the coals, particularly those in the Midwest, are high in sulfur content, creating difficult air pollution problems.

Little is known about the ignition quality of coal as compared to conventional liquid fuels used in internal combustion

Table 5.3-1 Sources and Analyses of Various Ranks of Coal

Classification by rank	State	County	Bed	Condition*	Proximate, percent: Moisture	Proximate, percent: Volatile matter	Proximate, percent: Fixed carbon	Proximate, percent: Ash	Ultimate, percent: Sulfur	Ultimate, percent: Hydrogen	Ultimate, percent: Carbon	Ultimate, percent: Nitrogen	Ultimate, percent: Oxygen	Calorific value, Btu per lb
Meta-anthracite	Rhode Island	Newport	Middle	1	13.2	2.6	65.3	18.9	0.3	1.9	64.2	0.2	14.5	9,310
				2		2.9	75.3	21.8	0.3	0.5	74.1	0.2	3.1	10,740
				3		3.8	96.2		0.4	0.6	94.7	0.3	4.0	13,720
Anthracite	Pennsylvania	Lackawanna	Clark	1	4.3	5.1	81.0	9.6	0.8	2.9	79.7	0.9	6.1	12,880
				2		5.3	84.6	10.1	0.8	2.5	83.3	0.9	2.4	13,470
				3		5.9	94.1		0.9	2.8	92.5	1.0	2.8	14,980
Semianthracite	Arkansas	Johnson	Lower Hartshorne	1	2.6	10.6	79.3	7.5	1.7	3.8	81.4	1.6	4.0	13,880
				2		10.8	81.5	7.7	1.8	3.6	83.6	1.6	1.7	14,240
				3		11.7	88.3		1.9	3.9	90.6	1.8	1.8	15,430
Low-volatile bituminous coal	West Virginia	Wyoming	Pocahontas No. 3	1	2.9	17.7	74.0	5.4	0.8	4.6	83.2	1.3	4.7	14,400
				2		18.2	76.3	5.5	0.8	4.4	85.7	1.3	2.3	14,830
				3		19.3	80.7		0.8	4.6	90.7	1.4	2.5	15,690
Medium-volatile bituminous coal	Pennsylvania	Clearfield	Upper Kittanning	1	2.1	24.4	67.4	6.1	1.0	5.0	81.6	1.4	4.9	14,310
				2		24.9	68.8	6.3	1.1	4.8	83.3	1.5	3.0	14,610
				3		26.5	73.5		1.1	5.2	88.9	1.6	3.2	15,590
High-volatile A bituminous coal	West Virginia	Marion	Pittsburgh	1	2.3	36.5	56.0	5.2	0.8	5.5	78.4	1.6	8.5	14,040
				2		37.4	57.2	5.4	0.8	5.4	80.2	1.6	6.6	14,370
				3		39.5	60.5		0.8	5.7	84.8	1.7	7.0	15,180
High-volatile B bituminous coal	Kentucky, western field	Muhlenburg	No. 9	1	8.5	36.4	44.3	10.8	2.8	5.4	65.1	1.3	14.6	11,680
				2		39.8	48.5	11.7	3.0	4.9	71.2	1.5	7.7	12,760
				3		45.0	55.0		3.4	5.5	80.6	1.7	8.8	14,460
High-volatile C bituminous coal	Illinois	Sangamon	No. 5	1	14.4	35.4	40.6	9.6	3.8	5.8	59.7	1.0	20.1	10,810
				2		41.4	47.4	11.2	4.4	4.9	69.8	1.2	8.5	12,630
				3		46.6	53.4		5.0	5.6	78.6	1.3	9.5	14,230
Subbituminous A coal	Wyoming	Sweetwater	No. 3	1	16.9	34.8	44.7	3.6	1.4	6.0	60.4	1.2	27.4	10,650
				2		41.8	53.8	4.4	1.7	4.9	72.7	1.5	14.8	12,810
				3		43.7	56.3		1.8	5.2	76.0	1.5	15.5	13,390
Subbituminous B coal	Wyoming	Sheridan	Monarch	1	22.2	33.2	40.3	4.3	0.5	6.9	53.9	1.0	33.4	9,610
				2		42.7	51.7	5.6	0.6	5.6	69.3	1.2	17.7	12,350
				3		45.2	54.8		0.6	6.0	73.4	1.3	18.7	13,080
Subbituminous C coal	Colorado	El Paso	Fox Hill	1	25.1	30.4	37.7	6.8	0.3	6.2	50.5	0.7	35.5	8,560
				2		40.6	50.3	9.1	0.4	4.6	67.4	1.0	17.5	11,430
				3		44.6	55.4		0.5	5.0	74.1	1.1	19.3	12,560
Lignite	North Dakota	McLean	Unnamed	1	36.8	27.8	29.5	5.9	0.9	6.9	40.6	0.6	45.1	7,000
				2		43.9	46.7	9.4	1.4	4.5	64.3	1.0	19.4	11,080
				3		48.4	51.6		1.6	5.0	70.9	1.1	21.4	12,230

* 1, sample as received; 2, moisture-free; 3, moisture- and ash-free.

Source: Baumeister & Marks, 1967.

engines. This could be an important factor, particularly in Otto cycle and diesel engines, wherein the basic engine design (compression ratio, ignition or injection timing, etc.) is largely influenced by the fuel properties. Ignition quality is largely dependent on the particle size and volatility of the coal used. It is known from past work that ignition quality and burning rate improve as particle size decreases. Particle sizing, however, has not been reported in enough detail to clearly identify the sizes needed for optimum combustion characteristics. Work needs to be done not only in this area but also to identify the chemical mechanisms by which ignition is initiated. The effort should also include a determination of whether the process of pulverizing changes the chemical as well as the physical properties of coal.

The concept of methanol-coal slurries introduces new possibilities and considerations with respect to utilization as fuel in engines. One approach is to burn the slurry as fuel in its as-received condition. This should definitely be feasible in Rankine cycle and other external combustion engine power plants. It may be feasible in gas turbines, although problems of corrosion and erosion could result from the coal constituents of the fuel, depending on the particular type of coal used. Using a methanol-coal slurry as fuel in reciprocating engines would be highly questionable because of the inherent difficulties associated with burning coal in these engines, as previously described.

The other approach is to separate the pulverized coal from the methanol to provide fuel for different applications. The use of pulverized coal in various types of engines will be discussed in more detail later. A discussion of using methanol as fuel follows.

5.3.1 Reciprocating Engines

Methanol has some significant advantages as a gasoline engine fuel. It burns much cleaner than petroleum fuels or even natural gas. It is a more flexible fuel than gasoline, permitting wider deviations from ideal fuel-air ratios. Although the net heat of combustion of methanol (8550 Btu/lb) is only about half that of gasoline, engines burning methanol can be made more efficient than gasoline engines. Compression ratios can be increased to take advantage of methanol's higher flame speed and good antiknock properties. Because it burns cleaner than gasoline, fewer emission control devices are needed. Since it burns cooler and has a cooling effect as it evaporates in the cylinders, cooling systems can be smaller and consume less power [Bryson, 1974]. It is completely mixable with gasoline in concentrations up to about 15%, an advantage which facilitates its use as a gasoline additive. The addition of 15% methanol to most motor gasolines increases the Research Octane Number (RON) significantly and the Motor Octane Number (MON) slightly. For example, with a typical unleaded gasoline having a RON of 93 and a MON of 84, the addition of 15% methanol having octane blending values of 120 RON and 91 MON would increase the octane numbers of the gasoline-methanol blend to 97 RON and 85 MON [Wigg, 1974].

Methanol has some disadvantages as well. One of its most serious problems associated with methanol-gasoline mixtures is phase separation, which relates to the question of fuel stability. Because of methanol's polar character, its solubility in gasoline is limited to about 15%, as indicated earlier. However, the phase separation problem becomes critical when the blend contacts even very small quantities of water. Rapid phase separation occurs, with the polar water-methanol phase settling out at the bottom. Gasoline containing methanol would therefore have to be stored and distributed under anhydrous conditions, which would be difficult and expensive. As an alternative, it might be possible to blend alcohol with gasoline at the pump, but this would also be expensive and require special equipment. Another problem with methanol is the possibility of vapor lock occurring in the engine fuel system. The addition of methanol to gasoline considerably increases the volatility of the fuel. If current gasoline vapor pressures were to be maintained, the use of 15% methanol blends would require removal of all the butanes and a significant fraction of the pentanes. A further potential disadvantage of using methanol-gasoline blends is the possible adverse effect on road performance of vehicles. Because of emission controls, most new cars are already carbureted near the lean limit for satisfactory performance, and additional leaning by methanol may tend to compound this problem. The wider flammability limits of methanol may partially compensate for the leaner carburetion, but the problem would still exist.

Essentially the only engine applications to date for methanol have been in racing cars and boats and in piston engine engine airacraft where it was injected directly into manifolds for added takeoff power. Some experimental testing has been conducted with methanol-gasoline blends during the past three years in late model and older model passenger cars, and the results suggest that in the area of fuel economy and emissions the benefits are only significant in the case of the older cars which operated with rich carburetor mixtures before emission control standards were imposed.

Burning of methanol is diesel engines presents a considerably more difficult difficult problem than in gasoline engines. The high octane number of methanol, which is an advantage in a gasoline engine, is a detriment in a diesel engine because the ignition delay (a function of the cetane number of the fuel) with methanol is much greater than with diesel fuel. A very high compression ratio, or an auxiliary means of ignition, would be required to properly ignite methanol in a compression ignition engine, particularly for starting and under idling or light load conditions. This would cause some complications in the engine design and no doubt result in greater cost as well as maintenance. Another potential problem would be more rapid wear in the fuel injectors because of the low lubricity of methanol. With some development effort it may be possible to adapt diesel engines to run on blends of methanol-diesel fuel, but the only apparent advantage would be to conserve a small percentage of diesel fuel, and it is questionable whether the compromises involved in engine design and performance would be worthwhile.

5.3.2 Gas Turbines

Methanol has excellent characteristics as a gas turbine fuel, primarily because of its clean burning characteristics which result in a lower level of harmful emissions and should help to ensure long component life and low maintenance. With its continuous combustion process the turbine is not subject to the limitations of reciprocating engines with regard to ignition quality.

The potential of coal-derived methanol as a substitute fuel for natural gas and petroleum-derived liquid fuel for gas turbines provided the incentive for a recent joint test project by AMAX, Inc., Turbo Power and Marine, and Florida Power Corp. (FPC) [Farmer, 1976]. A 12.5-hr run on methanol was conducted at one of FPC's gas turbine generator installations. The power plant was converted to a dual fuel configuration, both to allow direct comparison with standard fuel oil and to provide gas assist starting on methanol and No.2 oil. The only other engine modification was addition of a piston pump at the fuel supply to provide lubrication of the engine fuel pump.

Engine performance on methanol was reported to be excellent. Acceleration was normal, and steady state running was even more stable than on No.2 fuel. Burner can temperature patterns were the same as when burning Jet A or No.2, and there was little carbon buildup on the nozzles. Test data for NO_x emissions while burning methanol (Fig. 5.3.2-1) showed that, over the power range tested, emissions were 74% less than with No.2 oil. CO emissions were somewhat higher with methanol than with

No. 2 fuel, exceeding the projected EPA limit at loads higher than 15 MW (Fig. 5.3.2-2). However, the whole question of meeting proposed regulations is somewhat academic at present, because the regulations have not yet been promulgated.

From this and other programs (AMAX is also working with General Motors on vehicular turbine tests), it is concluded that methanol has excellent potential as a turbine fuel, provided it can be produced at a price competitive with petroleum-based turbine fuel. With coal-derived methanol, this appears to be a distinct possibility.

5.3.3 Boilers

The potential of methanol as a fuel for industrial boilers appears to be equally as good, if not better than, for gas turbines. Modifications should for the most part be confined to fuel pumps and burner nozzles to handle the larger volume of methanol required for providing the same output as regular fuel oil. With the continuous combustion process and a clean burning fuel, control should be relatively simple, and maintenance should be low.

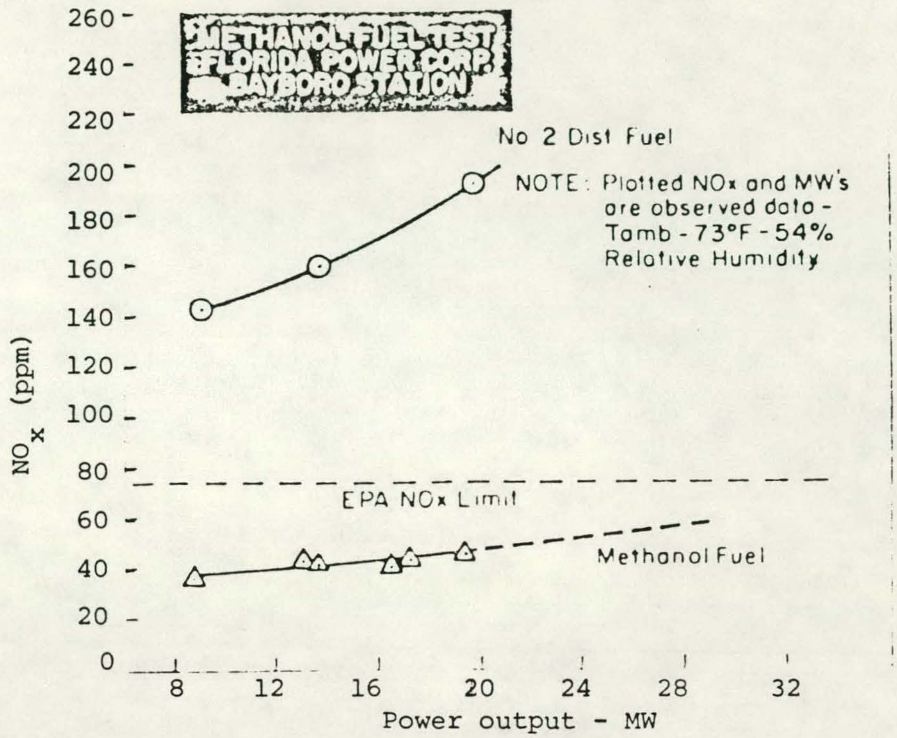


Fig. 5.3.2-1 - NO_x Emissions - Gas Turbine Test with Methanol Fuel

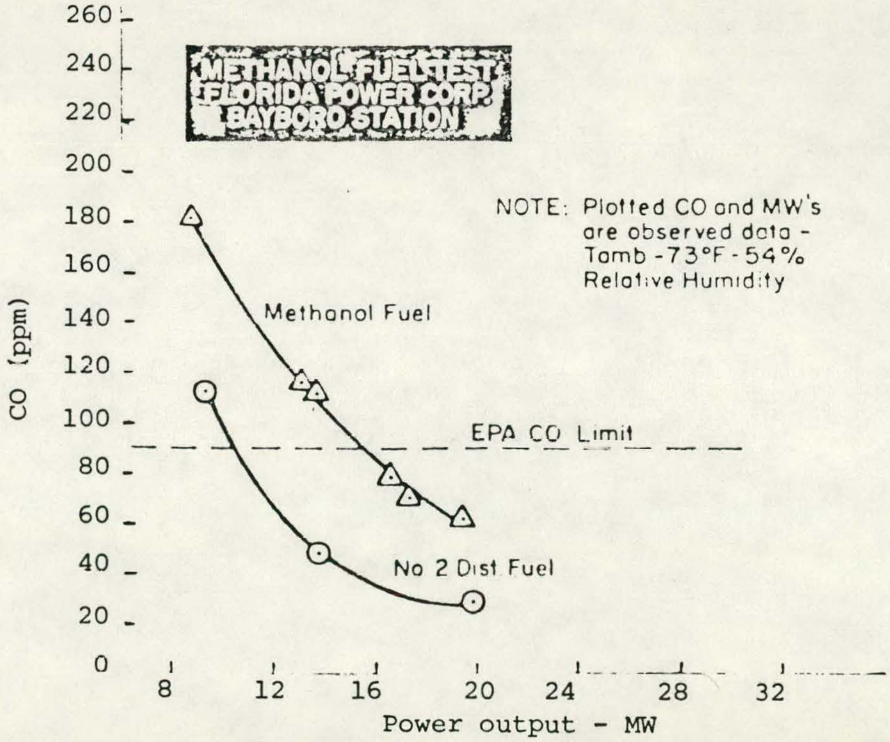


Fig. 5.3.2-2 - CO Emissions - Gas Turbine Test with Methanol Fuel

6.0 SUBSTITUTION OF COAL FOR PETROLEUM AND GAS IN PIPELINE OPERATIONS

The President's energy goals for the 1977-85 period, as stated in his message to Congress in April 1977, place heavy emphasis on both energy conservation and increased production and use of coal. Since coal is the Nation's most abundant energy resource, the President is thereby relying on coal to replace oil and natural gas for many industrial fuel applications. One of the major goals is to increase coal production by more than two-thirds, from 665 million tons in 1976 to more than 1 billion tons by 1985. Electric utilities are being pressed not only to use coal for their new power plants but also to convert existing oil and gas-fired generators to coal. The same philosophy applies to other industries that consume power, including the pipeline industry. The issues being addressed in this study of energy consumption in pipeline transportation systems must therefore include not only efficiency improvements in pipeline drivers using conventional gas and petroleum fuels but also methods of burning coal in those power plants.

The conversion of coal into synthetic fuels presents difficult economic problems. At present it is estimated that synthetic liquid petroleum, using the best available processes, would have to sell for about $20-25/bbl, which is nearly double the world price of crude oil. It is also estimated that, based on current technology, it would cost about $35 billion to build the coal liquefaction plants needed to replace only 10% of today's total petroleum consumption. Production of synthetic gas from coal is likewise not yet economically practical. Processes now being tested in pilot plants would require a price at least twice as high as the free market price of natural gas produced and sold within the same state; and three to four times higher than the government-regulated price for natural gas sold interstate. To build a commercial gasification plant capable of processing more than 15,000 tons of coal a day, it is estimated that the cost would be as much as $1 billion [Mullany, 1977].

ERDA, shortly after its creation in January 1975, moved toward the implementation of large demonstration projects for converting coal to clean fuels. The conversion and use of coal was given high priority in ERDA's National Energy Plan, issued in April 1976. ERDA to date has defined several demonstration projects for translating advanced concepts into commercial use. One, a clean boiler fuel plant, is now under way, with the award of a $237 million contract to the Coalcon Co. in 1975. The others include a pipeline quality gas plant aimed at industrial and commercial heating, and a fuel gas plant for electric power utilities or industrial uses. Industry cost sharing is concentrated in the more advanced phases of coal conversion, although there is some industrial cofunding in earlier developmental stages [White, 1976].

ERDA's coal conversion and utilization effort is directed toward demonstrating second-generation technology on a near commercial scale in the early 1980's. A variety of processes is being developed to convert Eastern and Western coal to liquids and gases. Coal utilization programs are directed toward development of processes to permit increased use of coal by direct combustion, with the objective of developing and demonstrating on a commercial scale the direct combustion of high-sulfur coal without exceeding pollution standards. Fluidized bed combustors containing sulfur oxide sorbents will be used in the burning of coal. This direct combustion technology is considered to have near-term (1985) potential as an alternative to existing boiler systems that use scrubbers for emission control.

While the coal conversion and utilization efforts in ERDA's Fossil Energy program discussed above have potential benefits in the overall spectrum of power generation, there are alternative approaches which could have significant benefits more directly related to the pipeline industry. One of these is discussed in para. 4.3.6 of this report concerning pipeline application of fuel cells, in which the potential application of fuel cell powered DC motors to eliminate throttling losses in liquid pipelines, is described. Another concept which merits consideration is the direct utilization of pulverized coal in liquid pipeline drivers with the coal brought to the pumping station in the form of slurries. The technical and logistical aspects of this concept are discussed below.

6.1 Coal Dust as a Pipeline Driver Fuel

The question of substituting coal and coal-derived fuels for the petroleum fuels presently used in pipeline operations involves consideration of the practicality of coal-fired engines and slurry-fired engines and the logistics of transporting coal and coal slurries to the pipeline pumping stations.

6.1.1 Coal Dust as an Engine Fuel

Any consideration of burning coal in engines presupposes that the coal has undergone certain processing. Coal receives an initial processing as it comes from the mine, including cleaning and grading to size. Those impurities that are readily removable (e.g., slate, shale, clay, sandstone, and pyritic sulfur) are eliminated by physical treatment. Organic sulfur and some incombustible materials cannot be eliminated by physical treatment. While there are techniques for removing some of the ash and sulfur, these processes are sophisticated and expensive, making it doubtful as to whether they offer any real advantage over coal liquefaction or gasification. Processing of pulverized coal for use in furnaces is a well established technology. However, little is known concerning the optimum particle size for efficient combustion in internal combustion engines.

6.1.1.1 Reciprocating Engine Fuel

Early in the development of the reciprocating internal combustion engine, attempts were made to use coal as a fuel. Almost all work has involved the use of pulverized coal either in a dry state or slurried in an oil or aqueous carrier. In 1898 Rudolph Diesel, after experimenting with an internal combustion engine using powdered coal as fuel, developed the compression ignition engine that bears his name. The powdered coal engines were not then successful and were abandoned when the diesel engine, using oil as fuel, ran with an efficiency much higher than any previous engine.

Development projects on coal-burning diesel engines were independently conducted by five industrial companies in Germany during the period 1916 to 1944, after which all efforts in this field were terminated [Soehngen, 1976]. Under these programs, which were in progress for periods ranging from to to 24 years, approximately 19 coal dust engines in the power range from 10 to 600 HP and rated at speeds from 160 to 1600 rpm were built and tested. The major problems experienced can be summarized as follows:

(1) Fuel feeding and control system. The most critical components were in the pre-chamber system with its control valves and nozzles, the task of which was to hold a properly metered amount of coal dust or coal dust/air mixture and transfer this fuel into the engine cylinder with correct timing and duration of injection in uniformly dispersed form and with a minimum of injection energy. Two basic systems were developed: (a) a compressor injection system using compressed air from an external compressor for pressurizing the pre-chamber and injecting the fuel in the cylinder; and (b) a compressor-less injection system using pressure rise through partial combustion of the fuel within the pre-chamber to inject the fuel into the cylinder. The compressor injection system had several drawbacks including high energy consumption, high cost, and complexity, with resultant reduction in system reliability. The compressor-less self-injection system did not adequately control ignition timing or injection timing or fuel leakage from the pre-chamber to the engine cylinder during the charging period.

(2) Fuel and combustion characteristics. The widely varying chemical and physical properties of coal made it very difficult to match the operational requirements of the engine. Factors of particular importance are heating value (typically ranging from 7000 to 15,000 Btu/lb); ignition temperature (ranging from 250°C for lignite to 800°C for anthracite) and ash content. The combustion phenomena involve three major parameters including ignition time, main combustion time, and burnout time. These parameters are affected by physical, fluid mechanical properties, and particle size and structure. Despite more than 100 years of combustion research, engineers are not yet in agreement on the exact combustion mechanism of a complex fuel such as coal.

(3) Wear and erosion. Excessive wear of vital engine components due to the abrasive action of ashes and unburned coke particles were the most serious detriment to long duration operation. Major engine components affected were cylinder liners, piston rings, injection nozzles, valve seats, and bearings and other sliding surfaces.

Although the German programs were, to a degree, successful in solving some of the basic problems of the coal-dust diesel engine, fuel consumption and efficiency, reliability and duty life of essential components were definitely inferior to those of comparable oil diesel engines. It is significant that virtually all engines built under these programs were experimental engines, only two of them having been put into practical use for driving machinery in factories on a routine basis.

Experience with coal-burning engines in the U.S. has been very limited, but in general has corroborated the results on those developed in Germany. Another problem of some significance, although not discussed in the German reports, is air pollution which can arise from the high sulfur content of some coals and the particulate matter in the exhaust. Because of these problems, as well as the wide variability of chemical properties of coal mined throughout the country, most researchers have concluded that direct utilization of pulverized coal in piston engines, at least those used in automotive and other mobile applications, is not practical. The use of powdered coal in large stationary engines should present less difficulty than in the case of the automotive engine because the stationary engines run at slower speeds and their combustion systems are less sensitive to fuel quality; however, they would still face many of the same problems with respect to metering, wear, and exhaust quality.

A relatively simple and more practical approach to burning coal in a reciprocating engine is to fit the engine with a suitable gas producer to partially convert the coal to combustible gases so that the gases may be consumed in the engine. Such gas producers convert carbon to carbon monoxide, losing about half the fuel heating value in the process. Cooled, filtered gas is then burned in the engine at normal engine efficiency, although the overall efficiency from fuel to output shaft power is of course reduced about 50%. The only engine modification required for a spark ignition engine is to replace the carburetor with a gas-mixing valve. For a compression ignition engine, a small pilot charge of diesel fuel is required to ignite the gas. In Europe, during World War II, about 500,000 trucks and autos were operated with coal or charcoal-fired gas producers because of the shortage of gasoline. Although bulky and inefficient, these vehicles proved to be fairly reliable and demonstrated that coal (or charcoal derived from wood) could be used as an emergency fuel when conventional liquid fuels were not available.

6.1.1.2 Gas Turbine Fuel

Some of the problems involved in burning pulverized coal is piston engines are also common to the open cycle gas turbine. In particular, the turbine is susceptible to hot corrosion and erosion resulting from high sulfur and ash content of most coals. The gas turbine does have one significant advantage over the diesel or Otto cycle engine in that it can burn a wide range of fuels of varying ignition quality. The absence of rubbing internal parts (pistons reciprocating in cylinders) in the compression and expansion processes is also an advantage from the wear standpoint.

Modifications necessary to a gas turbine to burn solid coal efficiently have recently been investigated by Solar Divsion of International Harvester Co. The work was done under a subcontract with Combustion Power Co., which had a contract with ERDA to identify the hot corrosion and erosion problems that would be expected in the hot end of a gas turbine burning coal in a fluidized bed combustor. The process included a fluidized bed combustor which operated on Illinois No. 6 coal, and three stages of filtration to remove the sulfur and separate the ash from the hot exhaust gases before they enter the turbine. The investigation indicated fouling, which can be expected from the fly ash at temperatures above about 470°C, to be the principal deterrent to the use of coal-fired gas turbines, although erosion may become significant in the lower temperature turbine stages and during spall of deposited ash. It was concluded that future work to improve the potential of operating gas turbines on coal must identify the principal contributor to the fouling mechanism, i.e, temperature, surface chemistry, particle energy; and then investigate strategies for mitigating the ash deposits and their resultant effects on the substrate.

Another contract, recently awarded by ERDA to Curtiss-Wright Power Systems, is aimed at demonstrating the feasibility of a gas turbine to burn high sulfur coal economically in utility service. The contract covers the design, construction, and operation of a pilot plant comprising a gas turbine with a fluidized bed combustor. This 300 MW pilot plant will be the first step in a practical application of this principle to power generation.

It is evident that the problems of burning coal in gas turbines are of far less magnitude than those in piston engines. If the erosion and corrosion problems can be successfully overcome, the open cycle gas turbine engine with the addition of a Rankine bottoming cycle could well prove to be a workable candidate for a prime mover for pumps used in liquid pipelines. As was discussed earlier in Section 4.1.1.2, an organic bottoming engine on a typical second generation gas turbine, representative of those presently installed on gas pipelines, could achieve an overall efficiency of more than 40% for the combined cycle plant. A retrofit of the pumping station could be accomplished either by fitting the combined cycle plant with an electric generator to supply power to the existing electric motor-driven pump, or by eliminating the electric motor and driving the pump directly from the gas turbine and Rankine cycle turbine (e.g., as illustrated in Fig. 4.1.1.2-1 above. In either case, a significant saving in operating cost over the cost of using utility power should result. There are of course other major cost factors to consider, including capital cost and maintenance cost, but the concept appears to be worthy of further study.

Another concept which merits consideration is the use of an indirect-fired coal-burning combined cycle pump station, such as a closed Brayton cycle power plant with a bottoming cycle, or some variation thereof. Since the turbine in a closed cycle power plant is exposed only to the working fluid (air in the case of a closed cycle gas turbine), it is not subject to the corrosion and erosion problems of an open cycle plant. The problems of burning coal are therefore confined to the air heater.

Development of closed Brayton cycle systems started in 1939 and the first power plant of this type was placed in service in 1940. This was an oil burning plant manufactured by Escher-Wyss in Switzerland. The first coal burning closed

Brayton system was an Escher-Wyss/GHH 2300 KW plant which started operation in Ravensburg, Germany, in 1956. It can be regarded as the starting point for the practical use of closed cycle machines after about 20 years of laboratory testing and development by Escher-Wyss and then licensees. This plant had approximately 120,000 hours running time by June 1976, with reportedly only minor problems and repairs. The combustion system could be changed in only one or two days to operate on coal, oil or gas. Altogether there have been some fifteen closed cycle gas turbine plants built in Europe, Great Britain, Russia and Japan, as indicated in Table 6.1.1.2-1. A number of these plants, including several which used coal as fuel, have accumulated over 100,000 hours of operation. Turbine inlet temperature have ranged from 650 to 750°C and plant efficiencies have been in the general range of 25 to 32%. Although those closed cycle plants have demonstrated ecnomic viability, utilities have been reluctant to install them on a broad basis, primarily, it would seem, because of the hitherto-wide availability of clean fuels whose combustion gases can be passed directly through a turbine. Such factors, without strong additional incentive or special requirements, have retarded nine rapid acceptance and application of closed cycle gas turbine systems. [Harmon, '76].

With the current emphasis on use of coal as fuel and with the advent in recent years of high-temperature materials, the closed Brayton cycle power plant may be the preferred approach to direct use of coal. One promising approach for achieving higher closed-turbine efficiency would be the use of new high temperature ceramics, such as silicon nitrate or silicon carbide, in the air heater. The addition of a Rankine bottoming system using waste heat from the air heater would result in a further increase in overall plant efficiency. Preliminary analysis indicates that, with a turbine inlet temperature of 1000°C (which should be feasible with ceramic materials in the air heater) and an organic bottoming system, an overall efficiency of over 40% could be achieved in a plant of 2000 KW or larger.

Units	Manufacturer	Application	Continuous Output	Heat Supply	Plant Efficiency	Commission-ing Date	Running Time	Fuel	Turbine Inlet Temp.	Comp. Inlet Pressure	Remarks
-	-	-	MWe	MW	%	-	Hours	-	°C	Bars	-
Escher-Wyss Zurich, Switzerland	Escher-Wyss	Power	2	-	32.6	1940	6000	Oil	700	-	First Test Plant Retired
Ravensburg, Germany	E-W + GHH	Power & Heat	2.3	2.3-4.1	25	1956	120,000 to 6-76	Coal or Oil	660	7.2	
Coburg, Germany	GHH	Power & Heat	6.6	8-16	28	1961	100,000 to 6-75	Coal	680	7.3	
Oberhausen I Germany	GHH	Power & Heat	13.75	18.5-28	29.5	1960	100,000 to 6-76	Coal/Coke Oven Gas	710	8	
Oberhausen II Germany	GHH + EVO	Power & Heat	50	53	31.3	1975	3,000 to 5-76	Coke Oven Gas	750	10.5	First Use of Helium as Working Fluid in Elect. Power Plant.
Haus Aden, Germany	GHH	Power & Heat Comp. Drive	6.4	7.8	29.5	1963	100,000 to 6-75	Mine Gas+Coal	680	9.3	
Gelsenkirchen Germany	GHH	Power & Heat	17.25	20-29	30	1967	75,000 to 6-76	Bl.Furn.Gas & Oil	711	10.2	
St. Denis Paris, France	-	Power	12.5	-	-	1951	5,000 since'56	Oil	660	-	First Big Plant/ Double Pressurized Heater
Toyotomi, Japan	Fuji Elect. & E-W	Power	2.0	-	26	1957	90,000	Nat. Gas	660	7.2	
Nippon Kokan Japan	Fuji Elect.	Power	12	-	29	1961	85,000 to 12-70	Bl.Furn.Gas	680	6.7	
Rothes Great Britain	-	Power	2.0	-	-	1960	~1,000	Coal Slurry	660	-	Stopped Due to Mine Closure
Altnabreak Great Britain	-	Power	2.2	-	-	1959	~1,000	Peat	660	-	Stopped Due to Mine Closure
Kashira Russia	E-W	Power & Heat	12	9-12	28	1961	-	Brown Coal	680	?	Achieved Guarantees
Spittelau Austria	BBC/E-W	Power & Heat	30-22	29-58	31-24	1972	-	Oil or Gas	720	-	Achieved Guarantees Dismantled
Die Oxygene Phoenix	E-W & LaFleur	Cryogenic Gas Production	2	-	-	1966	5,000	Nat. Gas	680	-	Helium Fluid; Dismantled
Adv. Power Conv. Exp. Test Facilities Ft. Belvoir, VA	Corps of Engineers (Stratos Et Al)	Army Power Requirements	0.5 0.5	-	18.6	1959	-	Oil	650	8.1	Experimental (Fluid, N_2)
Adv. Power Conv. Skid Experiment San Ramon, Calif.	Corps of Engineers (Stratos Et Al)	Army Power Requirements	0.500	-	16.7	1966	-	Oil	650	8.1	Experimental (Fluid, N_2)

Fig. 6.1.1.2-1 - All previously installed closed-cycle industrial/utility power plants

6.1.2 Logistics of Coal Consumption in Pipelines

The pumping stations in long distance liquid pipelines are for the most part located in remote sites, and there is currently no economical means of transporting coal to these sites. Since shipment by rail directly to the pumping station would be out of the question in most cases, the alternative would be to haul the coal by truck from the nearest rail terminal to the pumping site, where it would have to be processed into pulverized coal. This would be prohibitively expensive.

6.2 Logistics of Methanol-Coal Slurry as Pipeline Fuel

A potential solution to the logistical obstacle to use of coal in pipeline drivers lies in the use of the coal-methanol slurry, discussed earlier in Section 5.0, where it was concluded that the methanol-coal slurry offers an extremely interesting concept for pipelining coal. When that technology is developed, it will then be practical to consider use of crude oil and products pipelines to transport slurry as well. Thus, pipelines which transport oils and oil field and/or refineries to or through coal mining regions could also be used to move coal to their own pumping stations. The compatibility of the methanol-coal slurry with these other liquids at the interfaces must of course be investigated.

The pumping requirements of methanol-coal slurry would differ from those of crudes and products and would have to be carefully analyzed. The rheological tests described in paragraph 5.2.1 show that, with high concentratraticns of coal in the methanol-coal slurry, the pipe wall friction would be high and therefore more power would be required for a given throughput than with the conventional liquids. The pumps in crude oil and products pipelines are predominantly of the centrifugal type. Although these pumps would be suitable for pumping slurry, they are not likely to be the preferred choice, so that some efficiency penalty would be suffered. The mainline pumps used on the existing coal-water slurry lines are high efficiency, piston-type pumps, as indicated in Section 6.2. Although centrifugal pumps are used for in-plant commercial slurry systems, they are low efficiency type (on the order of 65%) with a relatively wide throat impeller clearance.

Additional questions of significance revolve primarily around whether the direct combustion of methanol-coal slurry in the gas turbine prime mover is determined to be practical. As indicated earlier in para.5, there is little question that methanol by itself is an excellent fuel for turbines. If it were necessary to separate the methanol from the coal, additional facilities would have to be provided and the logistics would be more complicated.

A detailed logistical analysis is beyond the scope of the present study. However, the concept appears to possess merit, and further study is recommended.

7.0 FLOW INDUCER IMPROVEMENTS

7.1 Liquid Pumps

Pumps used in liquid pipelines fall generally into two categories: centrifugal and positive displacement pumps.

7.1.1 Centrifugal Pumps

Centrifugal pumps operate at relatively high speeds and are usually direct connected to the drivers, the majority of which are electric motors. Centrifugal pumps are typically described as velocity machines in that their performance depends on the rotating velocity of impeller tips. The operating parameters that vary with speed are output flow, head, and the required drive power. Flow rates vary directly with speed; head varies with the square of speed; and required drive power varies with the cube of speed. In contrast to positive displacement pumps, centrifugal pumps develop a limited head at constant speed over an operating range from zero to rated capacity; therefore, excessively high pressures cannot occur. Figure 7.1.1-1 shows typical characteristic curves for a centrifugal pump at constant rpm.

The efficiency of single stage centrifugal pumps depends on specific speed (hydraulic design), capacity, inlet head, internal running clearances, surface roughness, and stuffing box friction. The influence of specific speed and capacity are dominating in most cases. The best possible efficiencies of centrifugal pumps depend to a large extent on specific speed as shown in Fig. 7.1.1-2. At very low speeds, friction losses become excessive, resulting in a rapid drop in efficiency as specific speeds fall below 1500 gpm.

Figure 7.1.1.3 shows a family of efficiency curves for a typical centrifugal pump which would be used on a small pump products pipeline. The optimum efficiency of 87-88% is representative of current state of the art.

Centrifugal pumps used on pipelines transporting only one type of product are normally selected to provide a certain head at a design flow rate. The speed of the pump is constant for that flow rate. When the pumps are driven by constant speed motors, the throughput can be decreased by throttling. The simplest and most flexible method of varying flow is by use of a throttling valve in the output line. The throttling valve has several advantages. No pump modifications are required, and circuit changes are minor. With such a valve, flow can be varied precisely during operation to obtain required flow rates. However, throttling has several drawbacks, the most important of which is loss of pump efficiency. Because the pump is delivering full effort against a partial obstruction, total pump efficiency is low for the usable flow delivered and the driving motor may be overloaded. Accordingly, throttling is generally used only for applications requiring frequent flow variations, in which high power consumption is acceptable. Another way to achieve a lower throughput is to reduce the speed of the pump by using a variable speed drive unit, such as a diesel engine, fluid coupling, gear reduction, or variable speed motor.

When two pumps are operated in parallel, the combined delivery for a given head is equal to the sum of the deliveries at that head, as illustrated in Fig. 7.1.1.4. For satisfactory operation in parallel, the pump units must be working on that portion of the curve that drops off with increase in the individual capacities of the two units in order to assure stable flow distribution between the pumps.

When two pumps are operated in series, the combined head for any flow is equal to the sum of the individual heads at a given capacity, as shown in Fig. 7.1.1.5.

Figure 7.1.1-6 illustrates an example, using two pumps in series, in which it is necessary to operate the pipeline at a throughput lower than design. Operation at point a on the system curve requires a lower throughput, which can be achieved either by throttling by an amount represented by H or by operating at reduced speed along the lower of the two curves. The variable speed method of reducing throughput would normally be the more efficient method. The application of these principles to pipeline operation has been discussed in Section 4.3.6.

7.1.2 Positive Displacement Pumps

Positive displacement pumps can be categorized into two principal classes: rotary pumps and reciprocating pumps.

A rotary pump consists of an assembly of gears, valves, cams, screws, vanes or other moving parts which rotate in a fixed casing. Instead of "throwing" the liquid as in a centrifugal pump, the rotor components push the liquid toward the discharge port much as a piston of a reciprocating pump does. Unlike the reciprocating pump, the rotary pump discharges a smooth flow. They will handle almost any liquid that is free of hard and abrasive solid material. Neglecting slip, rotary pumps deliver almost constant capacity against variable discharge pressure. Typical capacity and horsepower characteristics of a rotary pump at a given viscosity are illustrated in Fig. 7.1.2.1.

Rotary pumps are manufactured with capacities ranging from less than 1 gpm to more than 5000 gpm. They can handle pressures ranging to more than 10,000 psi and viscosities ranging from less than 1 centistoke to more than one million SSU. Their broadest field of application is in handling fluids that have some lubricating value and sufficient viscosity to prevent excessive slip at required pressure.

Reciprocating pumps are positive displacement units that discharge a fixed quantity of liquid during piston or plunger movement through the length of the stroke. Disregarding leaks and bypass arrangements, the volume of liquid displaced during one stroke of the piston or plunger equals the product of the piston area and stroke length.

The advantages of reciprocating pumps are flexibility of operation, nearly constant efficiency for wide ranges in capacity and head, and ability to handle small volumes at high heads. Disadvantages include valve troubles, pulsating flow and head, higher cost, greater required floor space, and higher maintenance cost due to the complexity of moving parts.

Positive displacement pumps used in pipelines are predominantly of the reciprocating type. The inherently high efficiency of these pumps is almost independent of pressure and capacity and is only slightly lower for a small pump than for a large pump. They are most useful in applications requiring high pressure and relatively low capacity, where their high efficiency more than offsets the high initial cost.

7.2 Slurry Pumps

One of the most demanding applications for a pump is in slurry pipelines. Both of the major coal slurry lines currently in existence use reciprocating pumps. Pertinent data on these installations are shown in Table 7.2-1 [Thompson, et al.,1972].

Selection of pumps for slurry pipelines has been related primarily to two factors: required discharge pressure and abrasivity. For required discharge pressures under 650 psi, centrifugal pumps have been selected (for slurries other than coal) based on lower cost. For higher discharge pressures, only positive displacement pumps are technically feasible due to casing pressure limitations on centrifugal pumps. In systems where positive displacement pumps are utilized, the more abrasive slurries require the use of a plunger pump which has the capability to continuously flush the plunger of abrasive

Table 7.2-1

Reciprocating Pumps in Use in
Major Coal Slurry Lines

	Consolidation Coal System	Black Mesa System
Length (miles)	108	273
Diameter (inches)	10	18
Annual throughput (million tons/yr)	1.3	4.8
Type of pump	Double acting, duplex piston	Double acting, duplex piston
Pump Manufacturer	Wilson-Snyder	Wilson-Snyder
Pump drive (hp)	450	1500,1750, 1750
No. of pump stations	3	4
Total number of pumps	9	6, 4, 3
Flow per pump (U.S. gpm)	550	2100,1400, 2100
Maximum discharge pressure (psi)	1200	1080,1785, 1165
Concentration (% by weight)	50	45-50
Maximum particle size	14 mesh	14 mesh

material. The less abrasive materials allow the use of a piston pump. The fluid end sections of piston pumps and plunger pumps are shown in Figs. 7.2-1 and 7.2-2, respectively. In the case of the Consolidation Coal and Black Mesa slurry systems, positive displacement pumps were selected because of their higher pressures and operating efficiencies as compared with centrifugal pumps. Piston and plunger type pumps were both investigated, and piston pumps were chosen as they were considered to have acceptable life with the abrasiveness of the coal slurry to be pumped.

Table 7.2-2 compares pump capabilities in existing slurry lines, including other slurry materials (limestone, copper concentrate, magnetite concentrate, and others) as well as coal.

Considerable improvement in pump maintenance costs has been experienced on existing pipeline systems. This has largely been accomplished with experimental programs over a period of years. For example, the life of rubber valve inserts was increased from only 90 hours initially to 1100 hours with improved polyurethane inserts; the use of chrome-plated liners increased the life of piston inserts from 180 to 500 hours and doubled the life of liners; and piston rod packing life was increased from 100 to 6000 hours. Maintenance life on expendable parts for low and high abrasive slurries is shown in Table 7.2-3 below.

Slurry pipelines utilizing positive displacement pumps usually have at least two variable-speed operating pumps per station in order to vary throughput and to simplify restart of the pipeline after shutdown. A number of different speed control devices, as indicated in Table 7.2-4 below, are being used on existing systems.

Table 7.2-2

Comparison of Pump Capabilities

Pump type Pump type	Maximum pressure (psi)	Maximum flow (gpm)	Mechanical efficiency (psi)	Maximum particle size
Plunger	3500-4000	920	85-90	8 mesh
Piston	2500-3000	2700	85-90	8 mesh
Centri-fugal	600-700	50,000	40-75	6 mesh

Table 7.2-3

Maintenance Life on Expendable Pump Parts

	Expendable Part Life (hours)*	
	Low abrasivity (piston)	High abrasivity (plunger)
Valves	1100	500
Piston rod	3000	-
Plunger sleeve	-	720
Piston liner	4000	-
Brass bushings	-	425
Packing	6000	425

*Approx. 1500 psi differential pressure

Table 7.2-4

Speed Control Devices for Electric

Drive Slurry Pumps

Type Speed Control	Where Used
Fluid drive	Consolidation Coal[1], Savage River,[2] Black Mesa[1]
Eddy current	Calaveras,[3] Waipipi[4]
Wound rotor motor with liquid rheostat	Bougainville[5]
Synchrodrive	West Irian[5]
Pole changing squirrel cage	Trinidad[3]

1-Coal slurry lines
2-Iron concentrate slurry line
3-Limestone slurry lines
4-Magnetite concentrate slurry line
5-Copper concentrate slurry line

Fluid couplings have proved to be satisfactory in high horsepower ranges. They are rugged, reliable, and require little maintenance. Eddy current couplings have performed satisfactorily in the lower horsepower ranges (less than 1000 hp) and provide more precise control than fluid couplings, particularly at lower pump speeds. Variable speed motors have also proved satisfactory for slurry pumping service, although they are more sensitive to variations in load than fluid and eddy current couplings. The pole changing squirrel cage motor is a variable speed device capable of operating at two speeds and operates at high efficiency at either speed. By having several pumps in parallel with different speed ratings, combinations can be used to provide several speed capabilities. The "Synchrodrive" unit is a device which combines gear reduction and speed control elements and which can be obtained in any size required for a slurry pump. This results in a saving of space and cost, and maintenance costs should be low because of reduced complexities of the system.

Centrifugal pumps are used extensively for in-plant commercial slurry systems, typically those used for in-plant transportation of slurries in the mining, cement, and other industries. Their application is generally restricted to short distances because of their limited head capability, lower allowable casing pressures, and lower efficiencies. On long distance slurry pipeline systems they sometimes serve as booster pumps, providing suction pressure required for mainline reciprocating pumps. Centrifugal pumps are also used to pump slurry through safety loops, allowing system operators to monitor the slurry for quality before committing it to the pipeline.

The efficiency of a centrifugal slurry pump is low because of the necessarily robust nature of the impeller design and the relatively wide throat impeller clearance. Efficiencies of 65% are common, compared to 85 to 90% on the positive displacement pumps used in slurry systems. The centrifugal slurry pump is a flexible piece of equipment in that, if sufficient drive horsepower has been installed, the head capacity can be increased or decreased simply by changing the speed of the pump. Belt-driven units are most common and the speed change is generally achieved by changing the drive sheave. In pumps with metal impellors, the diameter of the impeller can be increased or decreased to match the system characteristics.

New coal slurry pipelines in the planning stages may require throughputs as high as 20,000 U.S. gallons per minute to be transported hundreds of miles. Assuming that the pumps on the new generation slurry pipelines will be adaptations of existing positive displacement pumps, this could mean that pump capabilities of 4000 to 5000 gpm (as compared to the 1785 gpm maximum capacity of present pumps) will be desirable.

Another type of positive displacement pump which may offer potential for future slurry pipelines is a high flow, high pressure axial flow pump such as used in the NASA Saturn space program for pumping liquid hydrogen. A pump of this type is available with a volume flow rate of 18,000 gpm at 1000 psi discharge pressure. It could therefore replace as many as eight of the present piston pumps in a long distance coal slurry pipeline, and would probably require less maintenance. However, the axial flow pump is approximately 5% less efficient than the piston pumps now used, and it is doubtful if the overall cost would be less than with piston pumps of the latest design.

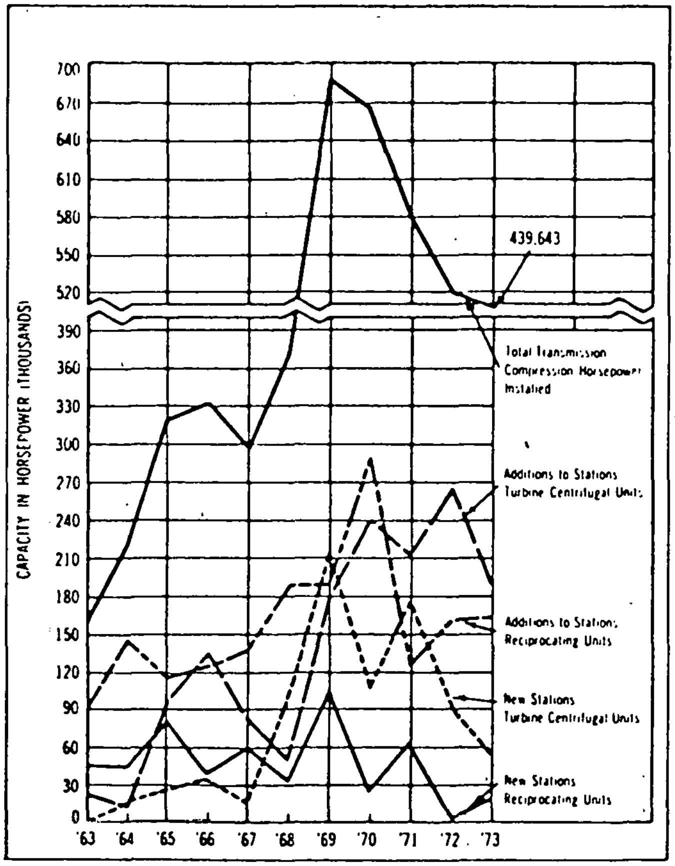
In general, it is concluded that the state of the art in both centrifugal and positive displacement pumps is well advanced and there are no technological breakthroughs which could be exploited in an ERDA-supported program.

7.3 Compressors

Until about 1947, the compression requirements of natural gas pipelines were satisfied entirely by reciprocating compressors. Since that time, the acceptance of the centrifugal compressor has steadily increased until it constitutes a substantial proportion of the total transmission compression horsepower installed. Figure 7.3-1 (FPC, 1974; Gas Turbine International, 1974] shows graphically the total capacity in installed compression horsepower for both gas turbine-driven centrifugal units and reciprocating units for the years 1963 through 1973. Over 50% of the total transmission line compression horsepower installed during this period was of the turbine centrifugal type. However, during 1973, the trend reversed. Since the Arab oil embargo, not only have new units tended toward reciprocators because of their higher efficiency, but on those lines whose sources are decreasing, the first units to be taken out of service were the turbines.

The centrifugal compressor is classified as a dynamic machine because all compression is achieved by continuous dynamic action of the blades and channels. Inertial forces are transmitted by a rotating impeller which, by centrifugal motion, adds kinetic energy to the gas by acceleration. The gas flows from the impeller into the diffuser where the gas decelerates and the kinetic energy is transformed into potential energy, i.e., pressure energy. A single centrifugal stage provides a relatively low pressure ratio (currently in the range of about 1.15 to 1.4). When larger ratios are desired, additional stages are added in series.

The natural gas centrifugal compressor was originally designed for natural gas boosting service. Pressure ratios up to 5:1 or higher (depending on gas properties) over a wide range of flows can be achieved. Higher pressure ratios can be obtained by operating two or more compressors in series with intercooling. Maximum discharge pressures of up to 4000 psig can be obtained using a high-pressure case.



Source: FPC Statistics

Fig. 7.3-1 - Transmission Line Compression Installed (reported 1963-1973)

The centrifugal compressor has broad pressure-volume characteristics. These are accomplished by using backward swept impeller blades, vaneless diffusers, and a variety of inlet guide vanes. Compressors using this type of staging have relatively low pressure ratios per stage, so a multistage machine is necessary for all but the lowest pressure ratios.

One of the most important parameters for classifying compressor impellers hydrodynamically is specific speed. It is calculated by the formula

$$\text{Specific speed} = \frac{\text{rpm}\sqrt{Q}}{H^{3/4}}$$

where

Q = inlet volume flow (ICFM), and

H = head (feet).

The specific speed is the speed that would be required of a geometrically similar machine to produce unit head with unit flow.

The peak efficiency of centrifugal impellers occurs in the specific speed range of 650 to 800. The selection of design speed among available gas turbine drivers is limited, and the pipeline industry has been reluctant to use gears. Therefore, in some cases it may not be possible to obtain the best specific speed range with the driver speeds available. However, efficiencies of over 80% within the specific speed range of 400 to 1350 can be expected with most single-stage impeller wheels. Figure 7.3-2 shows how the maximum flow capability of a compressor changes depending on the head [Walker, 1970].

Development of higher efficiency centrifugal compressors is continually pursued by some of the major industrial firms. For example, one of the leading gas turbine manufacturers has achieved an optimum efficiency of 86% in a single-stage compressor of 1.3 pressure ratio with a 24-inch-diameter impeller.

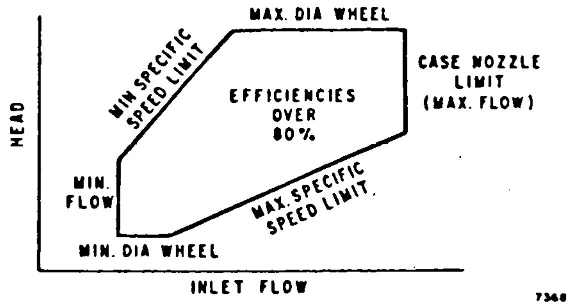


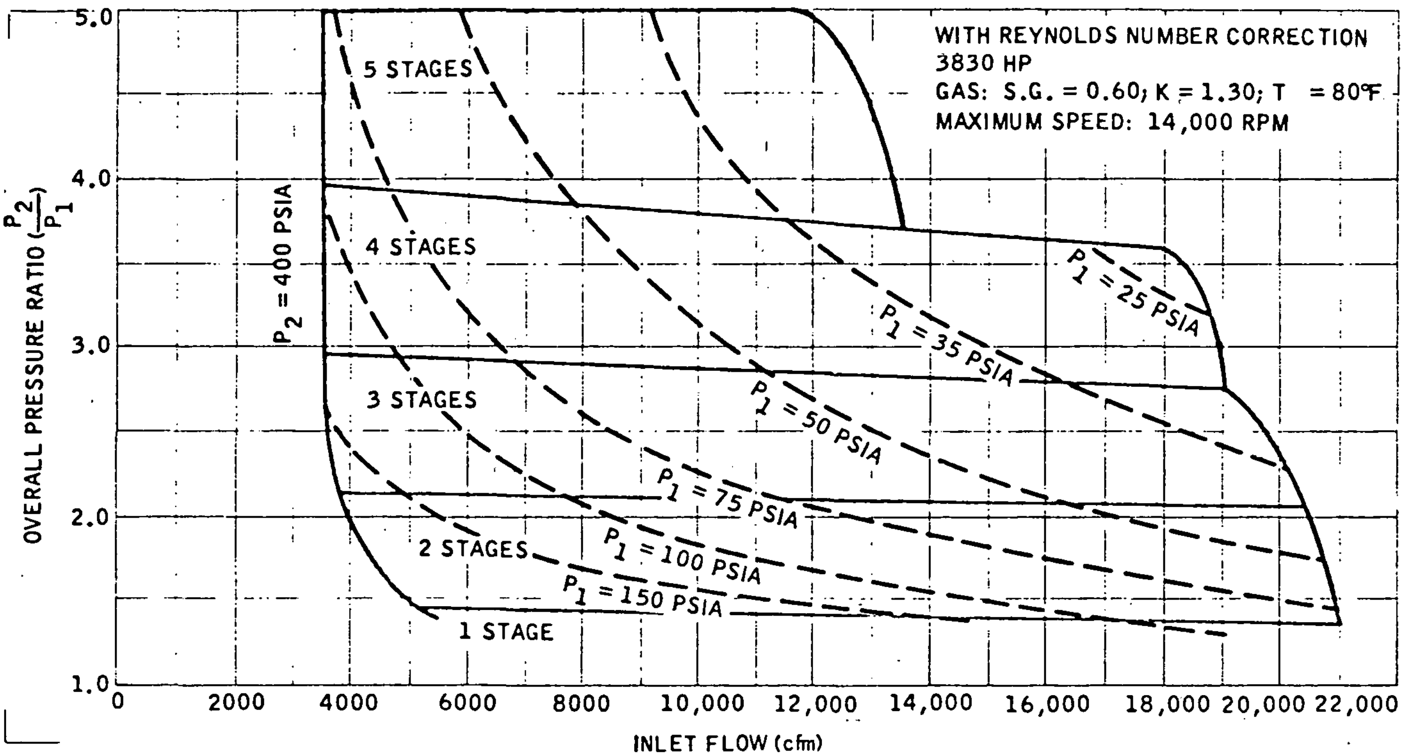
Fig. 7.3-2 - Centrifugal compressor case characteristics

However, the improvements in single-stage compressors cannot be directly translated into multistage machines, which require some compromises because of the broad range of flows and pressures involved.

In a typical case, a given compressor configuration is designed to handle a wide range of applications. For example, Fig. 7.3-3 shows a performance map of a Solar C505 centrifugal gas compressor. Using the curves, it is possible to determine whether a specific compression job is within the capability of this basic machine, and how many stages would be required for the specific site condition.

At the low heads required for pipeline service, the axial flow compressor could offer the possibility for a 6 to 8% improvement over current centrifugal equipment. About 24 axial flow compressors have been built and operated in closed-cycle gas turbine power plants throughout the world; the pressure levels, volume flows, and heads are comparable to gas pipeline requirements. Such a compressor, however, has two problems, one in design and the other in operation. The design problem is that of coping with the high bending loads imposed on the blading by the high specific mass flow. The operational problem stems from the narrower operating range from design point to surge at the tip speeds dictated by available turbine speeds and inlet volume flows [Robinson, 1972]. The axial flow compressor is also less rugged, more complex, and more sensitive to damage from ingestion of foreign objects than a centrifugal type. With the efficiency improvements that are being made in centrifugal compressors, there appears to be little likelihood that the axial compressor will be built in any substantial numbers for pipeline service.

The reciprocating compressor is classified as a positive displacement type machine in which a quantity of gas is drawn into a cylinder, where its volume is reduced and its pressure increased by movement of a reciprocating piston. Piston com-



Source: Solar publication T28B/675.

Fig. 7.3-3 - Solar C505 compressor performance pressors are designed for pressures as low as 1 psi above atmospheric or, with single stage compression, up to approximately 100 psi. When higher pressures are desired, the compression is divided into stages. The reciprocating compressor has the advantage of higher efficiency than the centrifugal compressor at the higher pressure ratios, although in many applications the centrifugal can claim efficiency superiority at pressure ratios below 1.35 as illustrated in Fig. 7.3-4 [Walker, 1970].

Many of the gas engine compressors used in pipeline applications are of the integral type, in which the power cylinders and compressor cylinders are arranged in a V-angle configuration on the same block and crankshaft. Normally such a machine has one compressor cylinder for each two power cylinders. Other gas engine compressors are furnished as matched engine-compressor sets, with the engine driving a separate compressor unit. Unloader valves used in the compressor cylinders are of various types, including both poppet and plate valves.

Many of the reciprocating engine compressors installed in natural gas transmission lines are operated in parallel with centrifugal compressors. In most cases there are several of the reciprocating engines with a large number of unloaders on the compressor cylinders. The reciprocating engine compressor units are usually of lesser capacity than the centrifugal compressor. Station control is accomplished by the unloaders until the horsepower is reduced to permit shutdown of a unit. Since a centrifugal compressor impeller has an operating flow range of approximately 70 to 130% of the design flow, it is not desirable to use a large centrifugal to handle the flow swings of a station when in parallel with reciprocating engines. It is much more desirable to base load the centrifugal to a certain horsepower level and allow the reciprocating units to handle the station swings [Walker, 1970].

In general it is concluded that, as far as efficiency improvements are concerned, there does not appear to be any significant rationale for advancing the state of the art in the compressors themselves by ERDA support. Rather, as discussed elsewhere in this report, the principal gains in efficiency will come from the prime movers through new design concepts and cycle improvements.

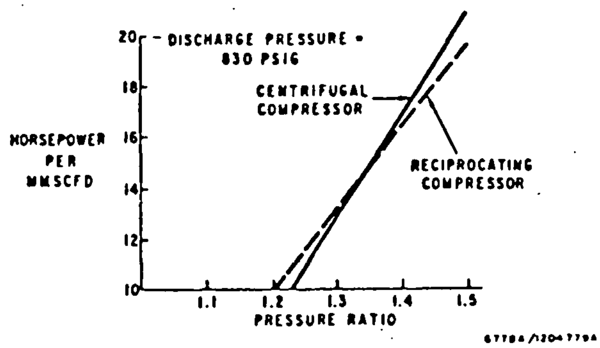


Fig. 7.3-4 - Horsepower required for centrifugal and reciprocating compressors

8.0 REDUCTION OF RESISTANCE TO FLUID FLOW

To examine the potential savings in both energetics and economics that might accrue from significant reductions in resistance to fluid flow, some simulations were run using the pipeline economic model (PEM), which was developed under Task 1 of this project. The model is described in reports R-3021 and R-3069 of this series (See Table 1.1-1 of this report).

For the simulation, the baseline petroleum products reference system, with a linefill consisting of 70% gasoline, 5% avgas, 5% kerosene, and 20% No. 2 fuel oil, was compared with the same system and linefill, but with viscosity reduced by factors of two and five. Selected output results are tabulated in Table 8.0-1.

It is seen that for the 20-year period from 1976 to 1996, a saving of $22.375 million in total energy cost could be derived from reducing the viscosity by one-half and a saving of $48.205 million from reducing the viscosity to one-fifth that of the baseline case. Savings in present value of the energy used are $6.516 million and $14.016 million for the one-half viscosity case and the one-fifth viscosity case, respectively. These savings figures are obtained by comparing the "Energy Costs" and "Present Value of Energy Used" for the three viscosity categories under the heading "Energetics" in Table 8.0-1. They are attributable entirely to the increased throughput which could be obtained by viscosity reductions alone and do not take into account any additional capital expenditures which might be involved in achieving the viscosity reductions.

The three primary methods of reducing fluid resistance in pipelines are:

(1) Heating the fluid to reduce viscosity
(2) Using additives in the fluid to reduce viscosity

Table 8.0-1

Viscosity Reduction Study*

	Baseline Case		Viscosity reduced to ½		Viscosity reduced to 1/5	
	Total (20 yr)	Average	Total	Average	Total	Average
Activity						
Throughput (MM bbl/mi)	1,704,590.797	85,229.539	1,704,590.797	85,229.539	1,704,590.797	85,229.539
Revenues	1,612,855.578	80,642.775	1,590,763.516	79,538.176	1,565,276.891	78,263.844
Economics						
Operating income (ICC Rules)	430,947.094	21,547.354	431,157.617	21,557.881	431,413.906	21,570.695
Net income (book profit)	375,137.840	17,863.707	375,348.352	17,873.731	375,604.641	17,885.935
Present value of book profits	136,608.590	6,830.429	136,787.037	6,839.352	137,004.424	6,850.221
Net cash generated	302,389.715	15,119.486	302,600.223	15,130.011	302,856.520	15,142.826
Present value of net cash generated	115,793.697	5,786.185	115,902.142	5,795.107	116,119.530	5,805.977
Rate of return on total capital (%)	8.470	8.470	8.472	8.472	8.474	8.474
Energetics						
Energy used (MM kw-hr)	5,577,754.250	278,887.711	5,022,350.187	251,117.508	4,381,353.437	219,067.672
Energy costs (MM $)	225,170,787	11,258,539	202,796,303	10,139,815	176,965,961	8,848,298
Present value of energy used (@10%)	65,065.001	-	58,549.497	-	51,044.112	-

*Dollars in thousands

(3) Using internal coatings in the pipe to reduce friction.

8.1 Pipeline Heating to Reduce Viscosity

The use of heating in petroleum pipelines has been confined primarily to relatively short length lines (100 mi or less) carrying liquids such as heavy crudes and heavy fuel oils which are too viscous to be pumped at ordinary ambient temperatures. Practice in the past has been to heat the liquid to a temperature much above its pour point (often from 140°F to 210°F) at the initial pumping station, then pump it through an insulated pipeline, kept hot by running a parallel, smaller line carrying steam. On such lines the pumping stations are usually spaced closely together and the oil is heated at each station.

Polyurethane foam coatings have been used for thermally insulating pipelines for more than 10 years, but extensive use of these coatings has occurred only since about 1970 [Hale, 1973]. More than 600 mi of insulated pipeline of all sizes, including length of pipeline up to 100 mi, have been installed since that time. The value of polyurethane foam for thermal insulation lies in the fact that it has the lowest value of thermal conductivity (0.13 Btu/sq.ft/hr/°F/in-per ASTM D2326) of all commercial insulation materials.

Another method of heating developed more recently uses a concept known as "skin effect current tracing" (SECT). This system, as illustrated in Figs. 8.1-1 and 8.1-2, uses the principal of skin effect, whereby a conductor is placed inside a heat tube and grounded to the far end of the heat tube [Hutson, 1976]. The heat tube is welded to the exterior of the carrier pipe, which is common carbon steel pipe. An AC potential is applied between the heat tube and conductor, causing an alternating current to flow down the conductor and return on the inside of the heat tube. The depth of penetration (skin depth) of the current is related inversely to

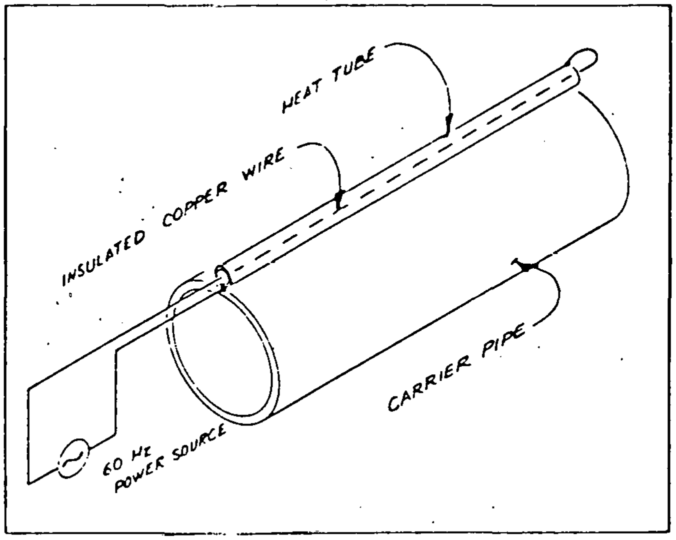


Figure 8.1-1 Basic skin effect tracing circuit.

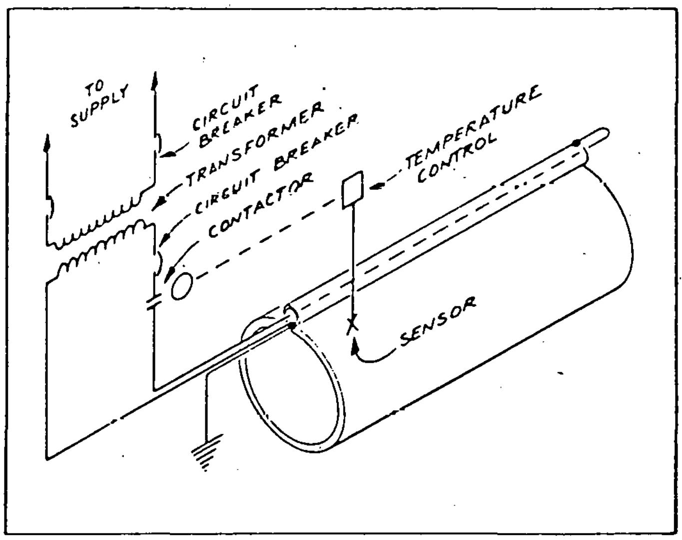


Figure 8.1-2 Details of SECT power circuit.

the square root of the frequency, relative magnetic permeability of the heat tube, and conductivity of the heat tube. The welded contact between the heat tube and the carrier pipe provides a heat flow path, and the temperature of the heat tube is normally not more than 15 to 20 degrees higher than that of the fluid. At the SECT control center, a transformer (Fig. 8.1-2) is provided for stepdown of the high voltage power supply, and temperature controls are used to maintain the required temperature settings. For most applications, one heat tube is used for fluid pipes up to 12 inches in diameter, two tubes for intermediate sizes and three or more tubes on pipes larger than 30 inches in diameter.

The SECT system was developed in Japan and is licensed to the Ric-Wil Co. of Canada and its subsidiary, Pipe Heating Systems, Inc., of Brecksville, Ohio. It has been used for heating pipelines carrying viscous crude, heavy fuel oil, molten sulphur, chemicals, etc. Approximately 45 systems have been installed in the United States in lines ranging up to about 10 mi long.

The effect of temperature on liquid viscosity may be correlated within the accuracy of most experimental data (1 to 2%) with the de Guzman-Andrade equation [Perry and Chilton, 1973].

$$\mu = Ae^{B/T} \quad .$$

This requires knowledge of two or more values of μ for evaluation of the constants A and B. When only one value of μ is known, the temperature dependence may be obtained within approximately 20% with the generalized chart shown in Fig. 8.1-3 [Perry and Chilton, 1973]. The chart is based mainly on data for organic liquids and is representative of liquids such as crude oil and petroleum products. It is evident that heavy liquids experience a much greater change in viscosity with changes in temperature than lighter liquids. This can be illustrated by the follow-

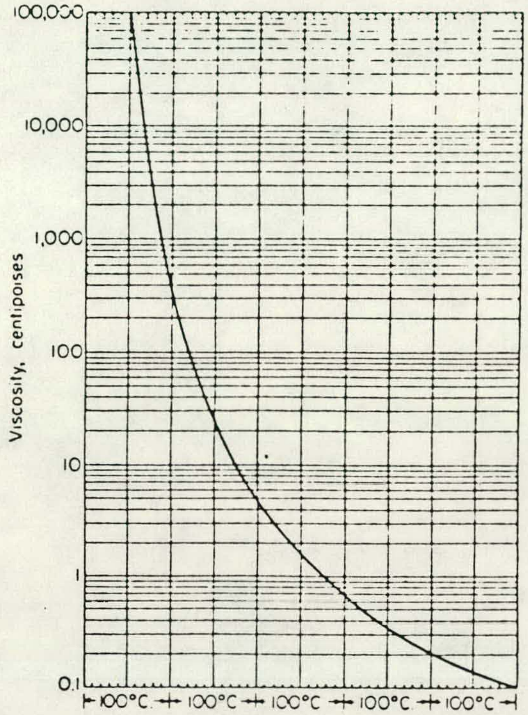


Figure 8.1-3 Approximate temperature variation of liquid viscosity.

ing example, which compares the viscosities of two typical crude oils at two different temperatures [Baumeister and Marks, 1958].

	Temperature (°F)	Viscosity (Centipoises)
California crude, light	60	48
	150	9
California crude, heavy	60	3500
	150	70

To assess the economics of heating pipelines to reduce friction losses, discussions were held with pipeline heating system contractor representatives. They indicated that such systems range in cost from about \$1 to \$5 per diameter inch per linear foot of pipeline, depending on the location of the pipeline, the temperature/viscosity characteristics of the fluid pumped, and a variety of other factors. (Refer to Table 8.0-1).

Using the most optimistic estimate of \$1/diameter inch/linear foot for the heating installation, the total increase in capital cost of reference the pipeline would be:

24" diameter x 5280 ft/mi x 686 mi x \$1/in-ft
= \$87 million.

Since the present value of the energy saving, at the postulated 20-year average cost of 0.037 \$/kwhr which was used in the comparison of Table 8.0-1, line 11 was only \$14 million, it is seen that the cost of energy would have to increase by more than a factor of six to even pay for the installation. It seems obvious from this comparison that the use of heating in a long distance petroleum pipeline is not economically practical and will not be in the foreseeable future.

It is concluded that pipeline heating is a well established technology for assisting in the pumping of viscous liquids for relatively short distances, but would not be economically attractive for long distance pipelines until energy costs were much greater at present and also probable until insulation costs were less.

8.2 Internal Coatings to Reduce Viscosity

Results of various tests and applications over the past 30 years have proven that internal coating of pipelines is an effective method of increasing pipeline flow efficiency. In addition to increased throughput, internal coatings provide other advantages including: protection against corrosion; reduced cost of scrubbers, strainers, pigs, and other types of pipeline cleaning equipment; prevention of contamination from corrosive products; reduced maintenance and labor costs; protection of pipe interiors against accumulation of foreign materials; and reduction of leakage from pipelines.

Epoxy-type internal pipe coatings are currently being used in pipe for transmitting dehydrated natural gas, wet gas, crude oil, sour crude oil, salt water, fresh water, and petroleum products. Thousands of miles of internally coated pipelines are in service, with pipe sizes ranging from 2 inches to 42 inches.

Two principal methods of application are employed for internal pipe coatings - "in situ" (or "in-place"), and spraying [Kut, '67]. In-situ coating is applied to lines already laid and avoids the w welding problem. Historically, the first coating of this type was carried out in 1947-48 in sour crude and sour gas gathering lines. The first in-situ salt water lines were coated in 1953, and potable water the following year. In-situ coating is a highly specialized procedure, and only a few companies operate in this field; however, as a result of experience in North America, Europe, and other areas, ample background is now available, enabling the in-situ coating applicator to use a proven coating, specifically selected for the particular service requirements.

Spray application of internal pipe coatings to individual joints of pipe is the second procedure. Suitable spray and cleaning equipment is available today for internally coating small- and large-diameter pipes, with appropriately formulated epoxy-type coatings, following sound surface preparation such as abrasive blasting or acid cleaning.

The largest spray application of epoxy coatings is for internal coating of natural gas lines. The requirement here is not primarily to give a protective film, but to provide a smooth coating to improve throughput. A thin film, on the order of 1.5-2 mils, is applied, following surface preparation by a rotary wire brush. Complete maintenance of film integrity is not essential for this usage, and the welds will inevitably not be fully coated.

Internal coating of gas transmission pipelines started in 1953. At the inception of this development, many questions were raised as to whether the coatings used would have the necessary properties, and whether the cost of internal coating would be offset by the resultant benefits. These questions have been largely resolved by close cooperation between coating manufacturers, gas transmission companies, and engineers.

Of the many tests made to evaluate the advantages claimed for internally coated pipe, the most elaborate have been those concerned with increased throughput. Extensive work has been carried out to measure the smoothness of pipeline internal surfaces, both uncoated and coated.

In 1955 the Institute of Gas Technology (IGT) conducted a project titled "NB-14 - Internal Coating of Pipe ", under the sponsorship of the pipeline research committee of the American Gas Association. Its purpose was to evaluate the feasibility of internally coating pipe for gas pipeline service. As part of this program, some 10 different generic resins were evaluated, involving 25 coatings, including vinyls, alkyds, polyvinyls, furanes, coal tar epoxies, phenolics, and neoprene coatings. Results showed the superiority of the epoxy-based type.

A related project, No. NB-13, was established by the pipeline research committee at IGT to investigate pipeline flow efficiencies. The results of this study showed that, for the ranges of flow and Reynolds number encountered in most large-diameter pipelines, the flow efficiency is dependent on pipe roughness and independent of

Reynolds number. As part of its work, the pipeline research committee developed a new flow formula in which the effective roughness of the pipe is used as a factor. The following table compares flow efficiency with effective surface roughness of 36-inch pipe based on this new flow formula.

Type of 36" pipe	Effective surface roughness (in.)	Pipeline flow Efficiency (%)
Internally coated	0.00028	103.8
Very smooth commercial	0.00045	100
Average commercial	0.007	96.5
Stored	0.0013	91.6

During this same time period, Transcontinental Gas Pipeline (Transco) conducted an experimental test program on 1199 miles of internally coated pipeline ranging from 20 to 36 inches in diameter [Crowe, 1959]. Various percentages of pipeline were coated on different sections. Epoxy resin type coating was used, since engineering studies had indicated it to be the most desirable for the purpose; the results of NB-14 confirmed this thinking.

After the first internally coated pipe was installed in 1955, the pipeline flow test data failed to reveal conclusively that internal coating was economically justified. For the next two years, the flow tests showed some indication of increased efficiency and were encouraging enough to Transco management to continue the program to the extent of designing for and considering the effects of internal coating on all main line additional facilities for expanding their system. Costs for cleaning the pipe and applying an epoxy coating varied considerably due to methods of manufacture, rate of production, and other factors. However, a general rule of thumb at that time for estimating costs of materials, application, and handling was to allow one cent per diameter inch per linear foot, with the smaller sizes of pipe running slightly higher.

Another project conducted during the late 1950's involved a series of tests on an 11.9-mile section of 24-inch OD x .250-inch WT pipe which had been in continuous service in the Tennessee Gas Pipeline Co. system for 10 years. The tests were part of a joint project under the auspices of eight gas pipeline companies and two companies specializing in pipe coatings [Klohn, 1959]. The project was organized in three phases:

Phase 1 - The section was tested after 10 years of operation.

Phase 2 - The section was cleaned with two separately run wirebrush pigs, and tested after cleaning.

Phase 3 - The section was cleaned with two wire brush pigs, followed by a 3000-gal. plug of Ketone as a detergent. The line was then coated with an epoxy and tested.

The results of testing at various flow rates in each phase showed that pipeline deliverability was increased by approximately 5 to 10%, depending on the rate of flow. This increase was broken down to a nearly constant 4% increase directly attributable to the cleaning operation of Phase 2 and an additional 1 to 6% directly attributable to the work done in Phase 3.

The tests satisfied the overall objective of the project, which was to answer the question of whether the condition of the internal pipe surface has much of an effect upon the deliverability in a large-diameter pipe in commercial service. This question is answered by examining Fig. 8.3-1. The total increase in the transmission factor, which is directly proportional to deliverability, varied from approximately 5% at a Reynolds number of 7 million to about 10% at a Reynolds number of 7 million to about 10% at a Reynolds number of 18 million. The tests were repeated on the internally coated pipe in December 1959 (a year after the original tests) and, as indicated in Fig. 8.3-2, the results showed good correlation with the previous test results. From this it was concluded there had been no apparent deterioration of the interior coating.

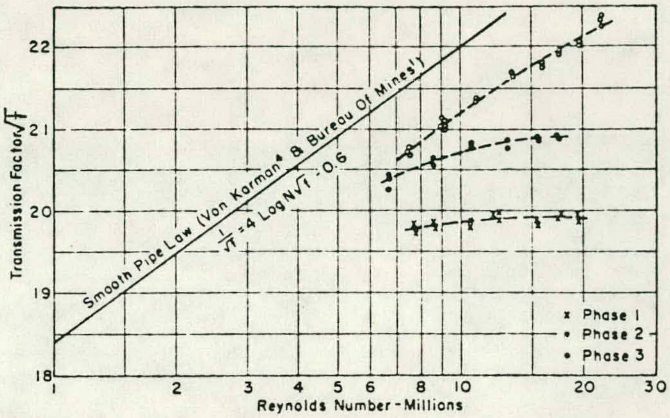


Figure 8.2-1 Performance on original tests, November and December, 1958.

Phase 1 - After 10 years operation
Phase 2 - After cleaning with wire brush pigs
Phase 3 - After internal coating

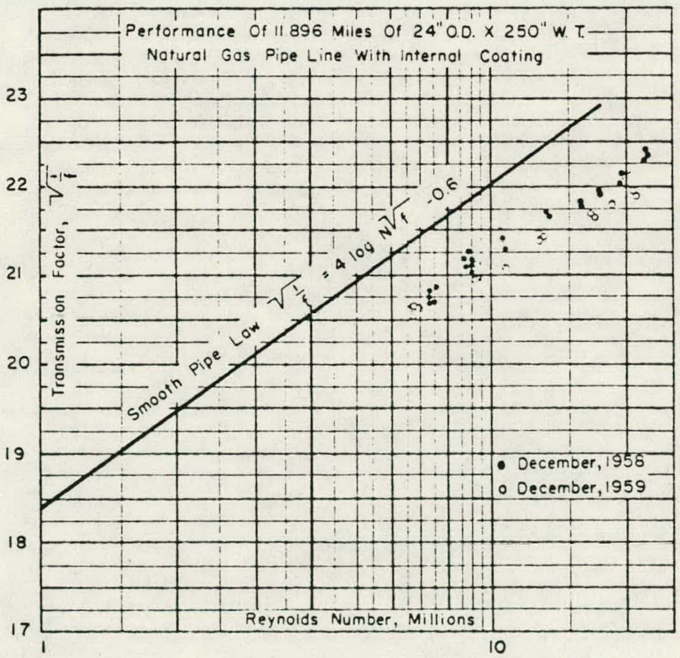


Figure 8.2-2 Performance of the same pipe one year after the tests described in Figure 8.2-1.

A significant recent application was recorded in 1973 when Signal Oil and Gas Co. completed the in-place internal coating of an offshore gathering system. Included in the system were 3 1/2 miles of two 12-inch and one 10-inch lines connecting platforms, and a 21.3-mile, 12-inch main line. All lines were internally coated with a polyamid-activated epoxy. Cleaning and coating operations on the gathering lines were carried out simultaneously from the platforms, and, on the main line, from a platform and an onshore site. Despite the fact that underwater tie-ins were involved, the use of divers was not required.

Another potential application for epoxy coatings is for protection against corrosion and metal loss within the interior surface of slurry pipelines. A recent paper [Jacques, 1977] evaluated alternative methods for corrosion control against various system parameters including capital cost, operating cost, and user suitability, environmental effects, effects on transport system, and flexibility. The alternative methods considered were heavy wall pipe, thin wall pipe with inhibitor, thin wall pipe with lining, thin wall pipe with oxygen removal, and thin wall pipe combination. The two methods involving internal coating, i.e., thin wall pipe with lining and thin wall pipe combination, ranked the most favorably on a cost comparison basis. The evaluation of thin wall pipe with lining was generally favorable otherwise except for concern expressed in assuring a continuous coating free of pinhole leaks. The cost figures for thin wall pipe with lining were based on applying 20 mils dry film thickness of epoxy lining. There is reason to believe that ample protection for a 25 to 30 year operating span could be provided with only 9 to 12 mils thickness of epoxy lining, which would make the cost comparisons even more favorable.

A current rule of thumb for estimating the cost of internally coating a new pipeline, as indicated by contact with one of the leading coating companies [Seefeld, 1977] is to use 15 cents per diameter inch per lineal foot for 6 mils thickness of epoxy coating, applicable to water, gas and petroleum products. For slurry, the figure would be 25 cents per diameter inch, based on 20 mils thickness of coating. These figures would apply for relatively short lengths of pipeline (up to 10 miles) and would be somewhat less for long pipelines. For in-situ coating of existing pipelines, there is no reliable formula for estimating costs, because of the highly variable nature of the cleaning and processing required.

8.3 Additives to Reduce Viscosity

The use of soluble additives to reduce the pressure loss associated with flow through pipes and tubes has been studied intensively since 1964. Although the data are confusing and sometimes contradictory, the indications are that a significant drag reduction can be obtained.

The theoretical basis for drag reduction by additives generally derives from modifications to the boundary layer of the flowing fluid. In the case of solid additives, particles with a high length-to-diameter ratio are apparently more effective than particles with other shapes. Several investigators attribute this to the alignment of the lenticular particles parallel to the direction of flow and the resultant modification to the laminar and transitional boundary layer adjacent to the channel wall. In the case of soluble additives, a similar phenomenon on a molecular level is postulated, since experimentally it is found that long-chain organic molecules have the most pronounced effect.

The most convincing explanation for the phenomenon of drag reduction is that propounded by P. S. Virk [Virk, '75]. His model is based on flow experiements with water as the solvent and with solutions of polymers at concentrations up to 300 ppm. The results of these experiments agree qualitatively with carefully conducted experiments by R. J. Hansen _et al_. at the Naval Research Laboratory, also with water as the solvent [Little, _et al_.]. Experiments with organic solvents [Ramakrishmaro _et al_.] are also in qualitative agreement.

At a Reynolds number of 10,000, the minimum friction factor attainable through drag reduction, according to Virk's equation, is 36% of the Newtonian friction factor; this is equivalent to a 64% reduction in pressure drop and pumping energy.

Virk's model postulated an elastic sublayer between the usual laminar boundary layer and the turbulent core; the hypothesis is that all drag reduction is related to the thickness and properties of this intermediate flow region. Historically, the analysis of friction losses in flow systems starts with the velocity distribution across the channel. For flow in pipes and tubes, a wide range of data is correlated in terms of universal velocity (U^+) and position (y^+) parameters as shown in the semilog plot of Fig. 7.1.2-1, taken from Brod [Brod, 1971].

For a Newtonian fluid in turbulent flow, two principal flow regions are observed as indicated by the curve A-B-N. In cases where drag reduction is observed, the Newtonian turbulent flow line is displaced parallel to the line B-N to form a velocity profile such as A-B-C-D, consisting of three flow zones: the turbulent core, the laminar boundary layer, and the intermediate "elastic" layer with the thickness represented by the horizontal "distance" between the points B-C.

The characteristics of drag reduction can be seen from this figure. The primary effect is that a higher mean velocity is observed for the same friction factor. This desirable effect apparently is a result of the significantly lower specific energy required to sustain turbulent flow of the system in the presence of the additive. The mean velocity increases above Newtonian velocity across the intermediate elastic sublayer, which extends from about $15 < y^+ < 60$. This effective slip is the explanation for the drag reduction phenomenon, according to Virk's model.

For practical application of drag reduction to design, the usual correlation of friction factor with Reynolds number is used, being derived by integration of the velocity profile of Fig. 8.2-1 over the cross section of the duct. The form of this correlation is:

$$f^{-\frac{1}{2}} = (4.0+\delta) \log_{10} (Re\ f^{\frac{1}{2}}) - 0.4 - \delta \log_{10} (\sqrt{2} d W^{o})$$

where

δ, W^{o} = parameters of the drag reduction system, experimentally determined

Re = Reynolds number of the solvent

d = pipe diameter.

Pipeline tests of drag-reducing additives have shown viscosity reductions of 48%, equivalent to a reduction in turbulent flow friction factor of 15%, which would also be the reduction in pump energy.

In report R-3022 (see Table 1.1-1 above), Sections 4.4.2.2 and 4.5.2.2, the annual energy costs of U.S. oil pipelines were seen to be 127 \$ million each for crude and products lines, or a total of 254 \$ million. The upside potential cost saving is thus 38 \$ million/year, or 2.5 \$ million for each point of efficiency gain. Clearly, a well-conducted R, D and D program should be highly cost effective, and it is accordingly recommended that such a program be undertaken.

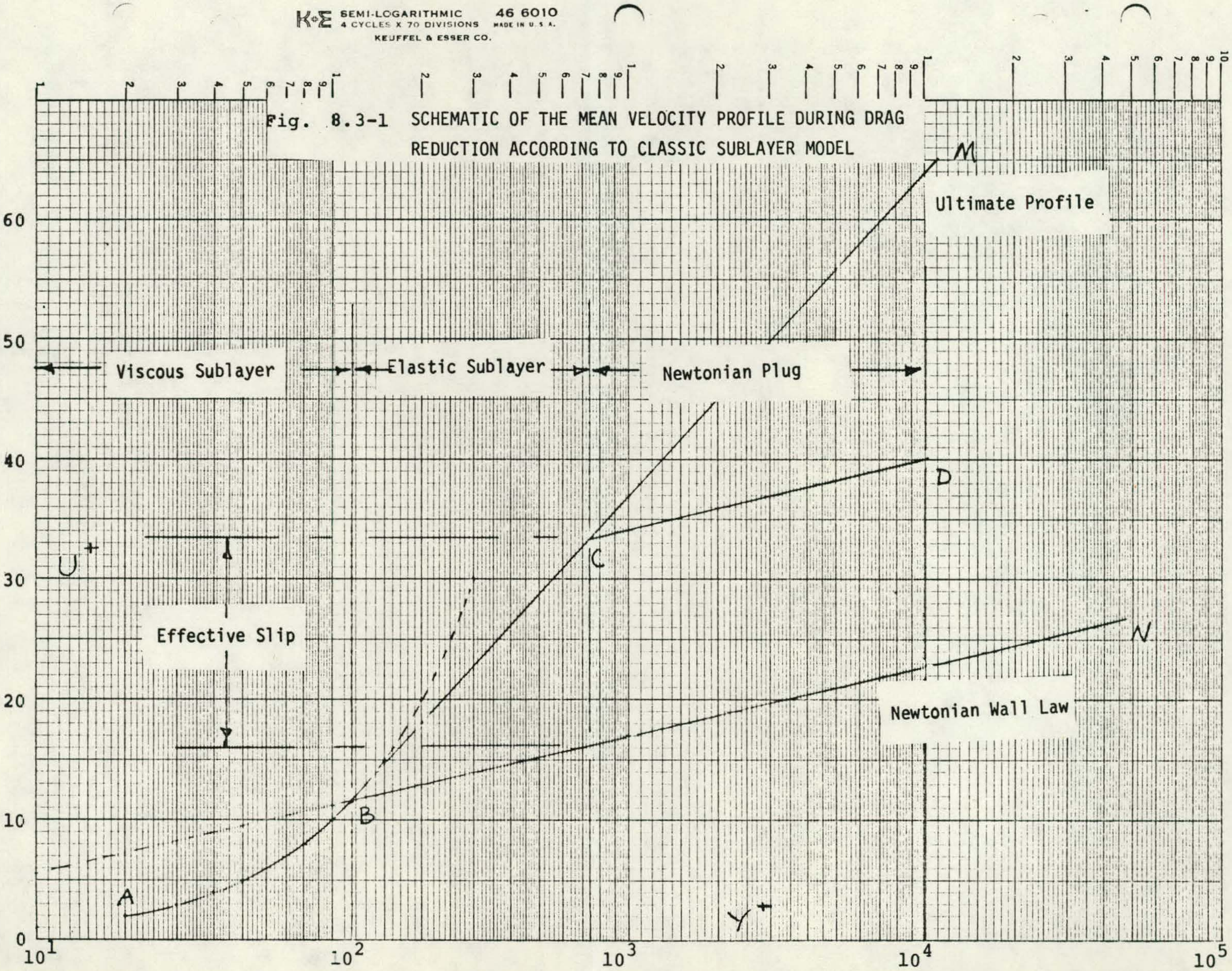


Fig. 8.3-1 SCHEMATIC OF THE MEAN VELOCITY PROFILE DURING DRAG REDUCTION ACCORDING TO CLASSIC SUBLAYER MODEL

9.0 OTHER IMPROVEMENTS

9.1 Cybernetics of Pipeline Systems

Cybernetics is a term coined by Norbert Wiener as a name for the science of information and control in complex systems. The science is now a well developed body of knowledge which is applied to practical systems through high speed, large memory computers, which assume many or all of the functions previously performed by human operators. Automation is a term which is often applied to this process.

The three major functions within a cybernetic system are:

1) The collection of information relative to the state of the system.

2) The processing of that information, along with whatever operator input may be required, through an algorithm which calculates the necessary control action.

3) The effectuation of the necessary control.

Functions 1) and 2) involve information and control only, and they can be performed by equipment which performs no other function and therefore operates independently of the pipeline system. This equipment is generally electronic, e.g., microwave communicators and high-speed computers. However, function 3 must be performed by equipment which is part of the pipeline system proper, i.e., motors, engines, pumps, valves, etc., and therefore performs noncybernetic functions as well.

Earlier, Section 4.3.6.1 presented a detailed discussion of duty cycles in product pipelines. There it was seen that such pipelines operate at steady state only when the entire linefill is a single product. However, this situation seldom obtains, and therein lies an opportunity for energy conservation. The existence of that opportunity was identified in Section 4.2.2, and the recommended approach to lifting of equipment limitations and solution of the associated problems was presented in Section 4.3.6.2. However, those discussions only treated function (3), the effectuation of necessary control, and showed that the ojbective is infinitely variable pump speed control, and recommended the fuel cell-DC motor approach. The cybernetic functions (1) and (2), the communication and control functions, still remain to be addressed. As noted earlier, these functions are generally performed by electronic communications equipment and high-speed computers. The following discussion of these applications is taken from Carter [1974].

Pipeline scheduling involves collecting and processing information from all shippers desiring movements as to the grade and quantity of the material to be moved, its origin and destination, and the approximate timing to assure that the various movements will arrive at the proper destinations on a timely basis.

After the schedulers have completed their work, they normally present a monthly schedule of movements to another group of people to handle the day-to-day, hour-by-hour, minute-by-minute activities necessary to carry out the schedule. These people are usually called dispatchers. Their functions and responsibilities are listed below.

1. The dispatcher must be assured that the material he receives is of proper quality and grade.
2. The dispatcher must accurately measure the volume received for each shipper and credit it to his account.
3. The dispatcher must keep accurate accounts of the location of the head-end and tail-end of each batch of fluid moving through the line.

4. A dispatcher must deliver the proper amount of the right grade and quality product at the right destination for each shipper.

5. A dispatcher must be assured at all times that his inputs equal his outputs, taking into consideration the fluctuation of working tankage.

6. All movements through a pipeline system must be accounted for and the proper charge assessed to them.

This dispatching function is the 24-hour/day, 7-day/week, nerve center of all pipeline systems. Dispatchers have at their fingertips the means for checking amounts of inputs and outputs and the flow through the pipelines by reading meters and guaging tanks remotely. They must have constant check on pressures to keep from bursting the pipelines. The quality and grade of the petroleum moving through the lines is known at all times through remote reading gravitometers, BS&W (basic sediment and water) monitors and other similar devices. All these are data needed by the dispatchers to evaluate and assure them the line is running properly and safely. The dispatchers direct tank farm personnel in switching tanks off or on the system and are in constant communication with deliverymen and gaugers, giving instructions as to the time, quantity and quality of material to be delivered at locations throughout the system. Practically all the work done by a pipeline company is dependent upon effective dispatching.

The dispatching operation is being automated in various ways by each pipeline company by using the electronic innovations of our day. In the early 1970's, Pipe Line Company advanced beyond the dispatching operation just described to a centralized Control Center operation. This Control Center combines the tremendous calculating capacity and data handling ability of the computer with the logic of the scheduler and the minute-to-minute instructions and logic of the dispatcher.

The use of computers and automation in the area of pipeline operations and control has been growing almost exponentially since the mid-1960's. A 1973 Survey on Computer Usage by the API showed that 57% of the pipeline companies were using computers for some type of operational application and that 33% were using computers for data acquisition and control. This compares with only 34% of the companies using computers in operational areas in 1962. In 1962 there were no on-line computer applications in the pipeline industry, and only three companies were studying on-line data acquisition in the mid-sixties.

On-line computer installations, such as that illustrated schematically in Figure 9.1-1,for data acqusition, dispatch calculations, and control are commonplace. In the future, they will be extended to all types of data. In addition to the present on-line readings for tank gauges, meters, temperatures, pressures, gravity, orifice flow rate, and interface detection, there will be on-line readings in the future for BS&W sampling, viscosity, vapor pressure, flash point, and even friction loss coefficients for dynamic hydraulic calculations. The principal advantage of on-data acquisition, is, of course, that it is must faster than obtaining telemetered readings by manual display, or over the phone for copying, and it is also more accurate. By 1974, 18 companies were known to have on-line data acquisition applications representing some 4000 on-line readings.

Dispatching calculations performed at a central computer location offer a tremendous advantage, especially when coupled with on-line data acquisition, because the computer can make these calculations both rapidly and accurately. These are the

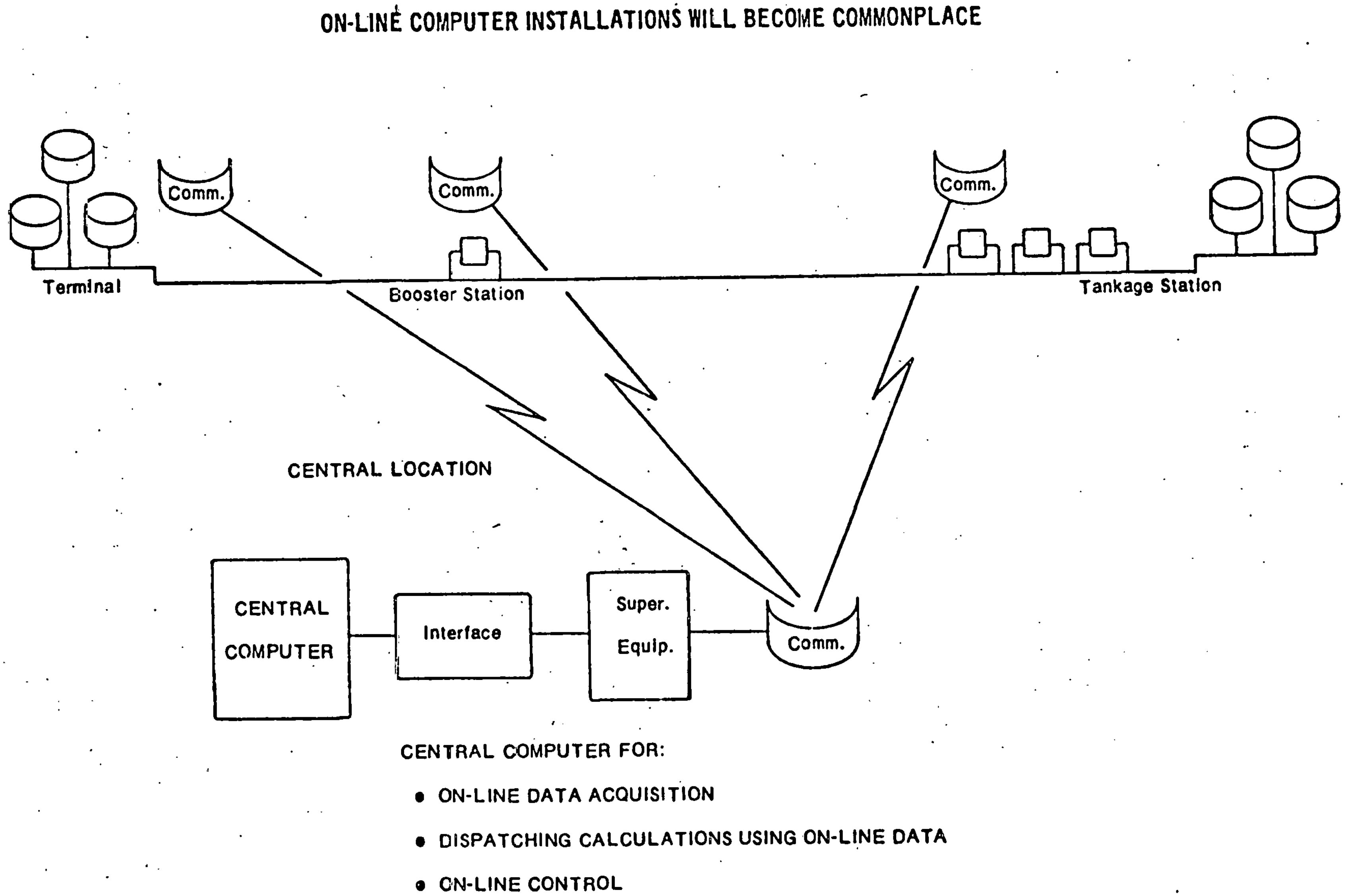
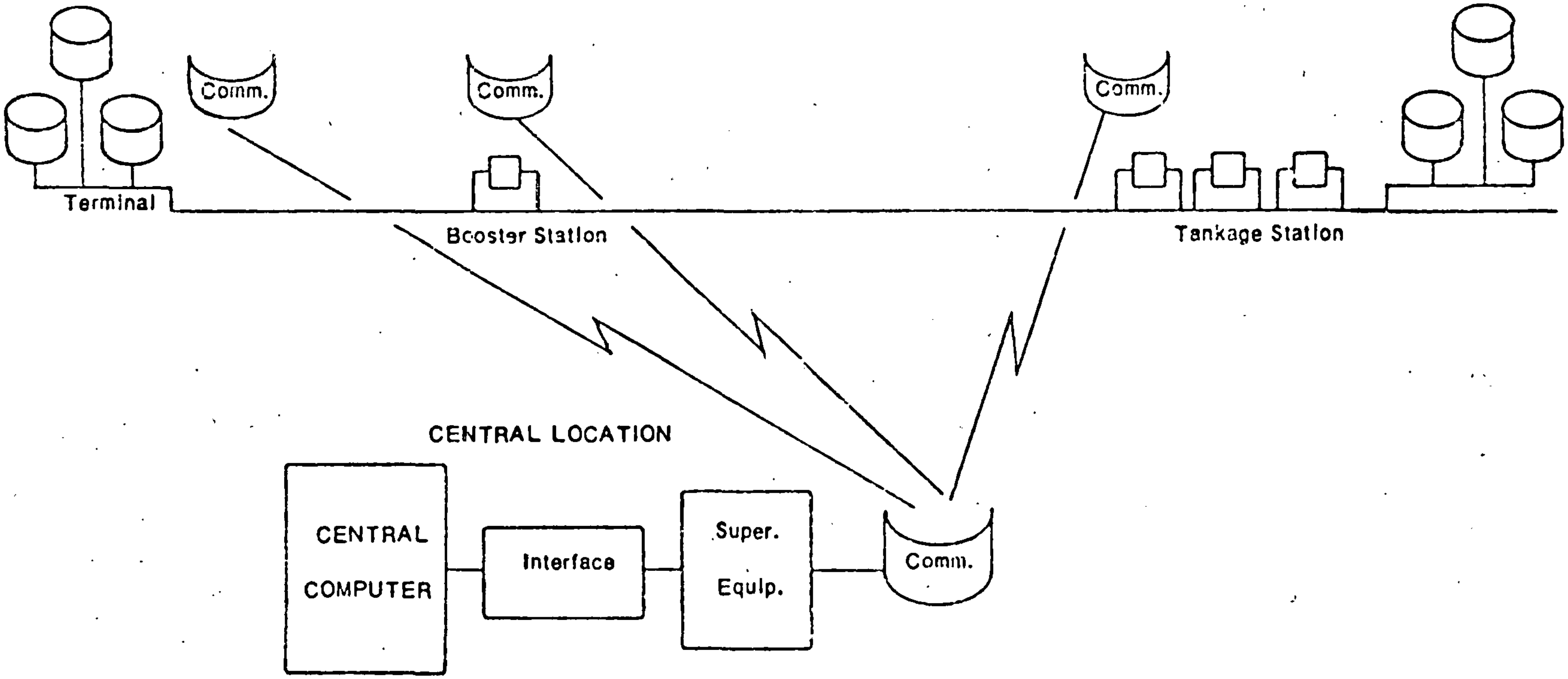


Figure 9.1-1. On-line computer control

types of dispatch calculations for keeping track of tankage and line inventory, calculations for receipt and delivery volume movement into and out of the line, line fill update (or batch tracking), batch arrival time, and line over and short calculations. In 1974 there were five companies that reported doing some type of dispatch calculations using on-line data.

There are definite advantages in having the remote unattended pump stations connected to the central computer for on-line control and interrogation of supervisory control status functions, as illustrated in Figure 9.1-2. Supervisory control equipment can be defined as equipment for controlling and supervising the status of some device, such as a motor or valve, over a communications circuit. The same equipment is used to telemeter pressures, temperatures, or tank gauges. The computer in this case has the ability to communicate with the remote control functions, and programs may be designed to perform certain control functions such as starting a pump unit at a specific time; or the program may or may not entail closed-loop operation. The computer can be tied to the remote supervisory control equipment to control pumping units, valves, and set-point controllers as well as to pick up supervisory status of alarms, pumping units running,etc. Another advantage of on-line control is that status information can be internally stored in the computer for hard copy output and records.

In 1964, ARCO Pipe Line installed what is believed to be the first on-line computer control system in the liquid pipeline industry. The operation of the system has been very successful and it has proved to be an economically justifiable endeavor. The system encompasses some 3200 miles of crude and products pipelines, 450 data acquistion points for tank gauge, meter, temperature and pressure readings, and performs



CENTRAL COMPUTER TIED TO REMOTE CONTROL EQUIPMENT FOR:

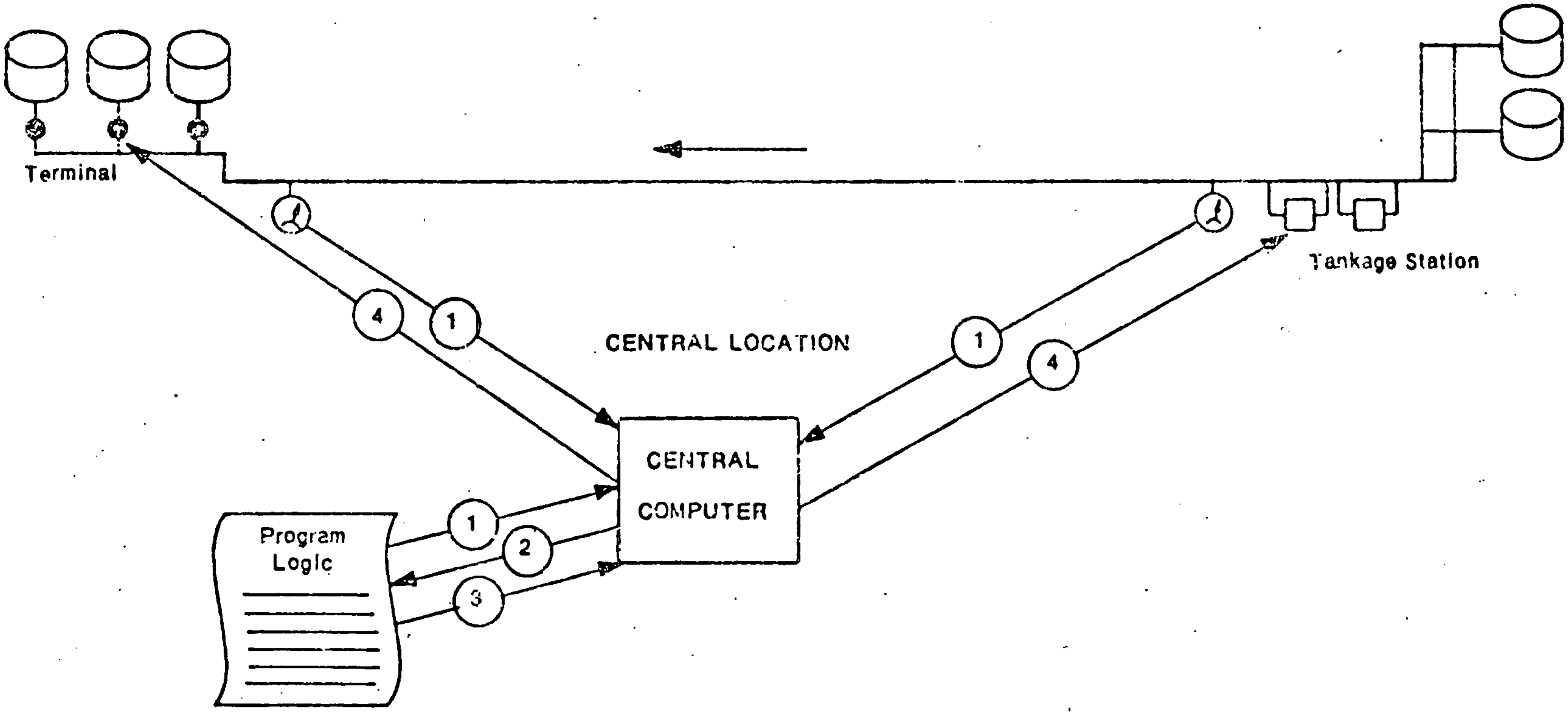
- START & STOP PUMPING UNITS
- OPEN & CLOSE VALVES
- SET POINT CONTROL
- ALARM INTERROGATION & RESET
- UNITS RUNNING STATUS

Figure 9.1-2. - Remote computer control of unattended pump stations some 370 independent supervisory control functions, including alarm and pumping units running status interrogation.

As technology and programming systems are developed, more and more sophisticated closed-loop programs are being implemented for controlling starting and stopping of pumping units, automatic switching of tankage and terminal manifold valves, pressure and flow set point control, and controlling upset conditions in general by closed-loop action. Some of these functions are illustrated schematically in Figure 9.1-3

Closed-loop control on the pipeline means that the computer, through its program logic, senses a condition and instigates some type of controlling action on its own without manual intervention. There can be varying degrees of closed-loop control. Pipeline segments may have only partial closed-loop control for a few functions, or the entire system may be under closed-loop control. The control may vary from a simple set point control of a suction pressure based on on-line calculation for arrival time of a different product, to the ultimate in which a schedule determines the closed-loop action for starting and stopping the pumping units, switching manifold valves, controlling optimum pumping units, and automically handling upset conditions. From a safety standpoint, closed-loop operations give faster reaction-control than manual operation in sensing upset conditions, e.g., pressure surges and other abnormal operating conditions, and in taking corrective action.

ARCO Pipe Line Company has a 185-mile section of 8-inch line from Western Oklahoma to near Shawnee, Oklahoma, which has nine injection stations on closed-loop control, as shown in Figure 9.1-4. This line handles a low vapor pressure crude and condensate and a high vapor presssure stream of natural gasoline



CLOSED-LOOP CONTROL:

1 SENSES CONDITION (IN FIELD OR BY COMPUTER PROGRAM)

2 PROGRAM MAKES CALCULATIONS, ANALYZATION

3 PROGRAM CALLS FOR COMPUTER ACTION

4 COMPUTER INSTIGATES ACTION OR CONTROL

Figure 9.1-3 Closed-loop computer control of pipeline system

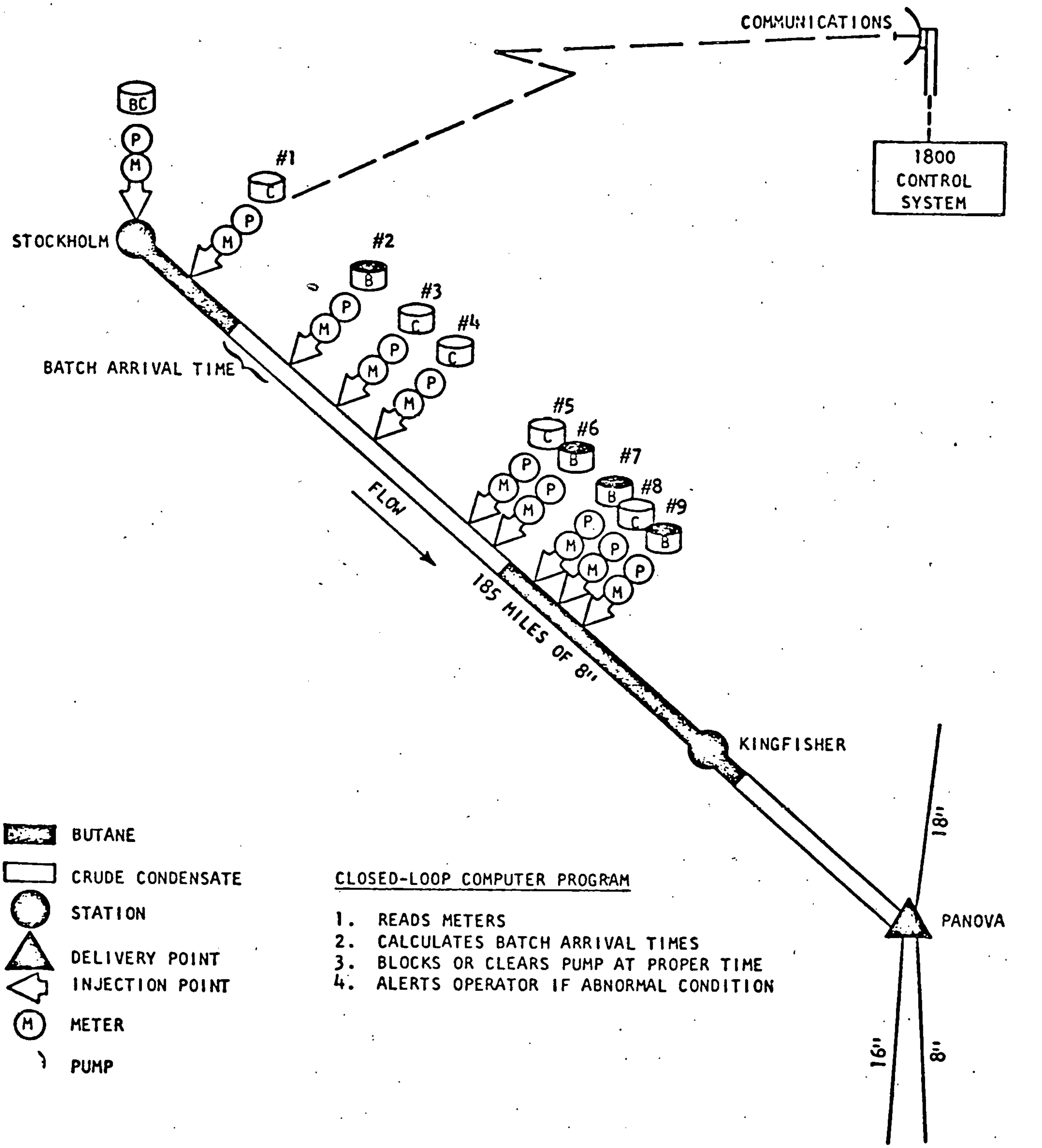
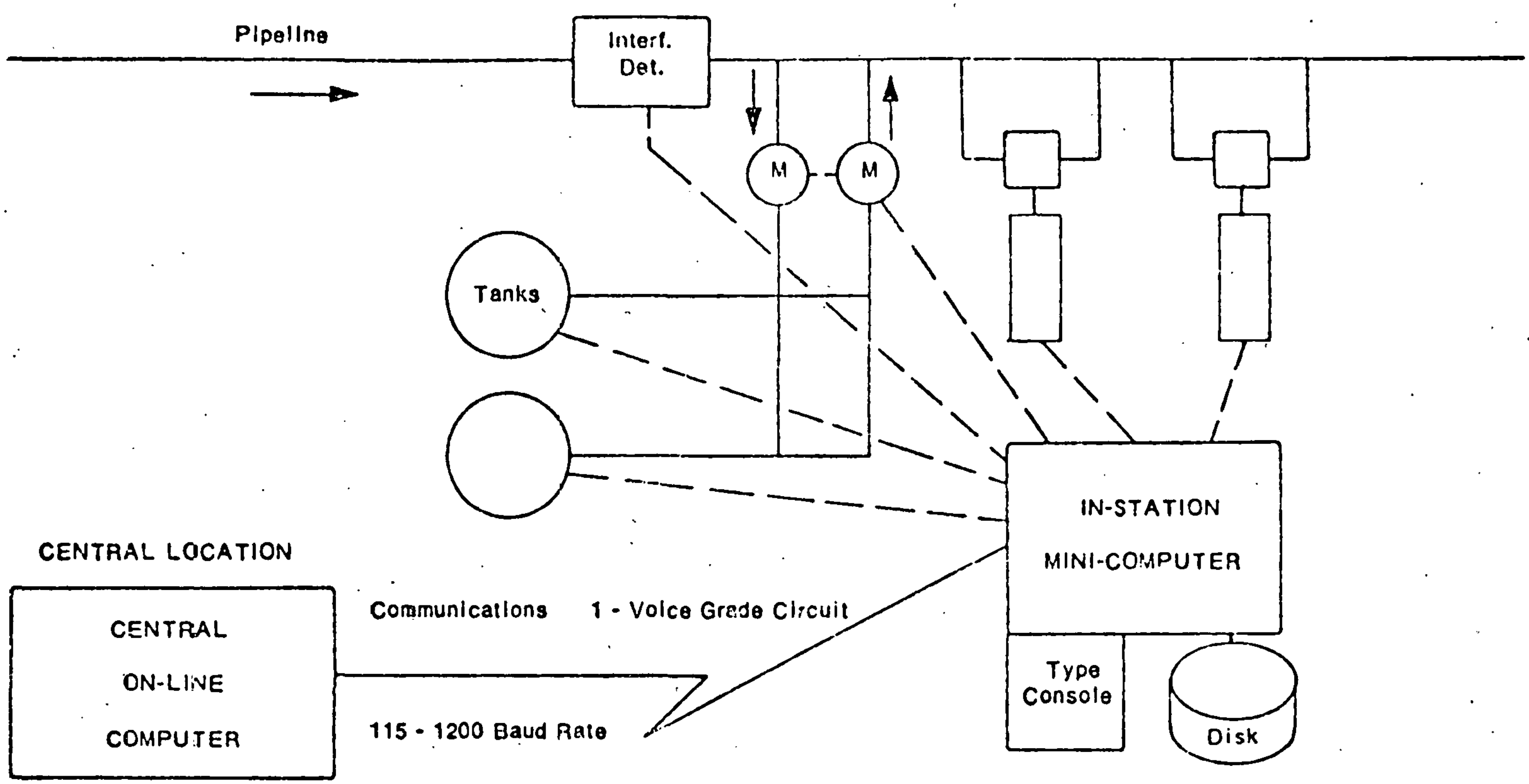


Figure 9.1-4 ARCO Western Oklahoma crude condensate line block-clear computer control program



- Switch Tankage Manifold Valves
- Start And Stop Pumping Units
- Data Acquisition:
  - Tank Gauge, Meter, Temperature, Pressure Readings
- Surveillance And Analyzation Of Station Operating Conditions

- Batch Interface Detection
- Vapor And Fire Detection And Control
- Programmed For "Safe" Operation:
  - If Communications Out To Central Location
  - For Upset Conditions

Figure 9.1-5 Mini-computer in-station control at tankage locations and butanes. The material from each of the nine points must be injected into the proper batch of compatible material as it passes the injection point. The computer control system reads the various meter readings, calculates the updated line fill and batch arrival times, determines if and when a pumping unit is to be started or stopped, and then initiates closed-loop action to block or clear the injecting pumping unit at the proper time. The program also alerts the operator to any abnormal conditions that are detected.

The fastest-growing current recent development in the use of computers for pipeline automation is the use of the mini-computers. Figure 9.1-5 illustrates the application potential

ARCO has an IBM System/7 computer at their Humboldt, Kansas, Tank Farm pump station. This computer is designed to:

1. Effect station control
2. Acquire data
3. Maintain surveillance of operating conditions and protective controls
4. Transmit data to and receive data from the central IBM 1800 System computer at Independence.

Humboldt Station is a relatively simple tank pump station operation. The station has 4 main line pumping units, 3 booster pumps, 14 tanks with tank mixers, and 2 positive displacement meters with meter proving facilties. The station operates on a 24-inch pipeline system, handling 6 grades of crude oil.

The computer is programmed to perform a majority of the operating functions presently being done by the man. The most unique function the computer will do is the sensing of gravity of the incoming streams to appropriate tankage as required. This and other functions programmed into the computer will permit unattended operation at the pump station except for daylight maintenance, housekeeping and custody transfer functions.

The computer is interfaced to the existing pipeline station equipment and instrumentation, both analog and digital, in four major categories, according to type of operation :

1. It functions as a station controller by switching tankage manifold valves, based on schedules supplied by the central computer or by interface detection from the gravitometer of the incoming line. It also starts and stops main line pumping units, booster pumps and tank mixers on a schedule supplied from the central computer.

2. The computer acquires data from tank gauges and meter readings each hour or on demand, and pressure readings, temperatures, gravitometer, sump tank level, etc.

3. The computer operates as a surveillance device of the critical station operating conditions for safe control. It reads pressures, temperatures, sump tank levels, checks operating limits, and takes necessary action if any of these readings are out of predetermined limits. It checks pumping unit running conditions and valve positions every minute. It detects abnormal operating conditions such as excessive pump unit vibration, excessive seal leakage, high tank levels, etc., and takes whatever corrective action is necessary. If corrective action is necessary, the computer alerts the Control Center operator at Independence of the malfunction

condition and in some cases places a telephone call to an on-call station attendant in Humboldt.

4. The computer is tied directly to the central on-line computer in Independence by microwave communications and relays various instructions for pumping schedules, batch arrival times, pump unit start and stop schedules. It also transmits hourly data on volumes received and pumped, tank levels, pressures, temperatures, and any pumping unit or valve changes.

The basic pump station design is such that in the event of any failures, a safe condition is maintained.. This basic analogy is also carried through to the programs within the in-station mini-computer, and the operation of the pump station continues in a safe manner if the computer also fails.

Eventually computers will become the heart of complete management information systems whereby on-line data, operating information, and billing data of each shipper will be directly available to those shippers that are equipped to use it. All major operating records and statistical data will be kept on disk storage data files within the computer system for immediate retrieval. There will be CRT display units, or other terminal units, in the pipeline offices of the schedulers, oil movements managers, and financial managers, so that they can call for the latest up-to-date information required to perform their duties. There will be computer-to-computer tie-ins with computers and terminals at the shipping companies and other pipeline offices for direct access of operating information; for obtaining the latest scheduling information, inventory volumes, and shipping forecasts.

The foregoing discussion shows that the use of computers to control pipeline operation is a proved and well-accepted method of reducing oil operating costs. Additionally, a literature search revealed that more than 100 articles have been written on the design of, and operating experience with, automated and computerized gas pipeline systems.

In almost every case, reports indicate that decisions to automate were made after very careful study, and results have been very satisfactory. Some operating companies are retiring existing systems in favor of more sophisticated new systems. In general, computer systems offer system control, optimization, information display and reporting, and telemetry capabilities.

The literature review has also revealed that at least six companies offer automation systems equipment design and/or installation. In many cases the programming of the controllers is accomplished as a joint venture between the pipeline company and the controller supplier.

Some of the major suppliers of compression and pumping units used in pipelines have also developed the capability for computing physical parameters affecting machinery selection, and have rendered valuable assistance to the pipeline customer in optimizing system controls.

In view of the well-established position of computer optimized control in the pipeline industry and the continuing effort within the industry to introduce further improvements, there does not appear to be any reason for ERDA-supported R&D in this area.

9.2 Leakage Inhibitors

9.2.1 Leakage in Liquid Pipelines

Leakage from oil and gas pipelines accounts for a considerable proportion of total operation expenses. It also represents an energy loss, and reduction of leakage is therefore a matter of prime interest in this study.

The ICC does not publish summaries of financial and operating statistics of oil pipelines. However, the individual pipeline companies submit annual reports containing such data to the ICC. These reports contain operating expense accounts which include as one of the items, "Oil Losses and Shortages." Comparison of this figure with the Total Operations Expenses gives an order of magnitude indication of losses due to leadage.

Under Task 1 of this study, estimates of total energy consumption and of energy intensity were developed and presented in report R-3022 (see Table 1.1-1 above) of this series. Tables 9.2.1 and 9.2.1-2 are replications of Tables 4.4-1 and 5.4-1 of that report. A comparison of columns 3, 4, and 5 is interesting for several reasons. First, oil shortages and losses are seen to be significant, i.e., about 6%, in comparison with fuel and power costs. Second, they are also a significant fraction, that is, almost 20% of non-fuel expense. Third, extreme variations are seen from company to company, in some cases assuming large negative values. In fact, the second-largest absolute magnitude in the losses and shortages column is a negative number.

This latter observation indicates that further research would be required before any conclusions and/or recommendations regarding energy conservation can be developed from these figures. Clearly while the companies are most certainly

Table 9.2-1

MAJOR CRUDE OIL PIPELINE COMPANIES - U.S. INTERSTATE TRUNKLINES, 1976

COST INTENSITY ANALYSIS

Company	1 MM B-Mi	2 M B	3 Fuel & Power $	4 Oil Losses &Shortages $	5 Total Operating Expense $	6 Fuel & Power $/MM B-Mi	7 $/M B	8 Avg. Shipment Mi	9 % of Tot. Operating Expense Col.3	10 Col.4	11 Total Operating Expense $/MM, B-Mi	12 $/M B
Lakehead	293,629	391,540	18,507,533	832,231	23,206,770	63.03	47.27	750	79.8	3.6	79.03	59.27
Amoco	190,548	410,263	16,878,116	-	22,188,619	88.58	41.14	464	76.1	-	116.45	54.08
Shell	128,236	368,829	9,987,771	4,628,156	17,888,104	77.89	27.08	348	55.8	25.9	139.49	48.50
Mid-Valley	107,986	142,803	9,977,052	104,559	11,261,282	92.39	69.87	766	88.6	9.3	104.28	78.86
Texas Pipe Line Co.	94,083	335,957	6,668,842	397,358	9,143,089	70.88	19.85	280	72.9	4.3	97.18	27.22
Mobil	93,114	308,884	8,401,843	415,253	10,589,953	90.23	27.20	301	79.3	3.9	113.73	34.28
Arco	81,258	239,406	7,159,934	515,104	10,724,514	88.11	29.91	339	66.8	4.8	131.98	44.80
Marathon	63,480	256,586	5,646,913	(1,950,054)	4,974,345	88.96	22.01	247	113.5	(39.2)	78.36	19.39
Exxon	62,111	445,637	6,178,988	1,417,716	10,988,270	99.48	13.87	139	56.2	12.9	176.91	24.68
Ashland	52,542	76,148	3,594,068	665,964	5,007,641	68.40	47.20	690	71.8	13.3	95.31	65.76
West Texas Pipe Line	52,392	131,873	2,255,450	(90,901)	2,823,009	43.05	17.10	397	79.9	(3.2)	49.19	21.41
Southcap	44,234	69,378	2,393,579	-	2,649,767	54.11	34.50	638	90.3	-	59.90	38.19
Platte	35,357	51,307	1,821,852	-	2,165,823	51.53	35.51	689	84.1		61.26	42.21
Portland	23,322	140,242	3,082,364	872,086	4,173,679	132.19	21.98	166	73.9	20.9	178.96	29.76
Chicap	23,285	118,014	1,914,162	-	2,114,390	82.21	16.22	197	90.5	-	90.80	17.92
Texaco-Cities Service	22,715	109,398	1,915,315	(51,751)	2,787,382	84.34	17.51	208	68.7	(1.9)	122.71	25.48
Pure	20,939	93,228	1,825,319	12,325	2,861,435	87.17	19.58	225	63.8	4.3	136.66	30.69
Texas-NMex.	16,567	155,154	1,183,487	(30,687)	1,601,305	71.44	7.63	107	73.9	(1.9)	96.66	10.32
Owensboro-Ashland	16,033	54,348	538,092	(365,191)	373,886	33.56	9.90	295	143.9	(97.6)	23.32	6.88
Minnesota	13,330	51,304	2,553,828	(126,291)	2,706,174	191.58	49.78	260	94.4	(4.7)	203.01	52.75
Cities Service	12,788	104,546	1,115,491	(38.946)	1,629,096	87.23	10.67	122	68.5	(2.4)	127.39	15.58
Total/ Average	1,447,949	4,054,845	113,600,990	7,206,931	151,858,533	78.45	28.01	357	74.8	4.7	104.87	37.45

Source: ICC Annual Reports "P", Pipeline Companies, 1976

Table 9.2-2

MAJOR PETROLEUM PRODUCTS PIPELINE COMPANIES - U.S. TRUNKLINES, 1976

COST INTENSITY ANALYSIS

Company	1 MM B-Mi	2 M B	3 Fuel & Power $	4 Oil Losses &Shortages $	5 Total Operating Expense $	6 Fuel & Power $/MM B-Mi	7 Fuel & Power $/M B	8 Avg. Shipment Mi	9 % of Tot. Operating Expense Col.3	10 % of Tot. Operating Expense Col.4	11 Total Operating Expense $/MM, B-Mi	12 Total Operating Expense $/M B
Colonial	591,688	569,396	56,503,564	127,388	64,383,213	95.50	99.23	1039	87.8	0.2	108.81	113.07
Plantation	105,640	186,089	9,059,872	372,462	13,682,728	85.76	48.69	568	66.2	2.7	129.52	73.52
Texas Eastern	65,570	115,518	5,090,446	657,295	9,182,267	77.63	44.07	568	55.4	7.2	140.03	79.49
Williams	62,463	177,781	7,756,856	-	14,029,105	124.18	43.63	351	55.3	0	224.60	78.91
Mid-America	42,577	103,648	3,971,866	(478,014)	6,249,930	93.29	38.32	411	63.6	(7.6)	146.79	60.30
Explorer	33,805	59,029	1,730,074	623,459	3,312,077	51.18	29.31	573	52.2	18.8	97.98	56.11
Southern Pacific	26,080	206,846	4,648,535	67,154	8,566,931	178.24	22.47	126	54.3	0.8	328.49	41.02
Dixie	18,797	29,078	2,155,141	445,103	3,453,575	114.65	74.12	646	62.4	12.9	183.73	118.77
Hydrocarbon	18,474	27,364	3,670,318	-	5,262,716	198.67	134.13	675	69.7	0	284.87	192.32
Wolverine	13,009	83,276	2,615,420	193,035	3,855,490	201.05	31.41	156	67.8	5.0	296.37	46.30
Olympic	12,838	68,424	985,814	-	1,724,286	76.79	14.41	188	57.2	0	134.31	25.20
Santa Fe	9,683	20,044	265,876	-	7,954,109	27.46	13.26	483	3.34	0	821.45	396.83
Yellowstone	8,918	20,784	754,969	(26,027)	1,219,048	84.66	36.32	429	61.9	(2.1)	136.70	58.65
Laurel	8,457	42,706	582,661	699,672	2,146,303	68.90	13.64	198	27.1	32.6	253.79	50.26
Total/ Average	1,017,999	1,705,983	99,791,412	2,681,527	145,021,778	98.03	58.49	597	68.8	1.8	142.46	84.81

Source: ICC Annual Reports "P" Pipeline Companies, 1976.

losing some oil through leaks, the leakage can move only in the outward direction. The presence of large negative values leads then to the inference that the account is some sort of inventory-balancing artifice, rather than an actual tabulation of physical losses. Additionally, it is difficult to see how very large amounts of oil can be lost without some kind of environmental impact becoming evident. This physically evident consequence, together with the economic drive to minimize losses, should insure that the companies would move aggressively to inhibit and stop leaks.

In view of the foregoing, it is concluded that the information at hand does not warrant an ERDA program in this area. Moreover, it is unnecessary to develop further informat-ion, because the only method of active leak inhibition that has been identified is by internal coating of the lines, and an ERDA program in that area has already been recommended for other reasons in Sections 8.2 and 3.7 above. It is thus concluded that further research to understand the implications of column 4 in the Tables is not needed for present purposes.

9.2.2. Leakage in Gas Pipelines

Statistics of interstate natural gas pipeline companies for the year 1974 [FPC, 1974] show that transmission system losses of all A and B companies (those with annual revenues of $1 million or more) amounted to over 68 billion cu.ft. This quantity constitutes 2.4% of the total gas receipts for that year. At the current wellhead price, this loss would be worth over $100 million, and its true national value, as defined in Section 2.4 of report R-3024 of this series, is about $250 million.

Examination of the "unaccounted for" figures reported by the individual companies reveals that, unlike the oil line case, no negative values are reported. Also, in the case of gas, it is easy to see how significant leakage could occur without environmental impact or even detection. The reported figures may thus represent significant physically real, recoverable leakage.

In the past, a large part of this leakage was due to venting during compressor blowdown. With the recent strong emphasis upon conserving gas, the companies have generally taken measures to retain most of this gas.

In Section 8.2 above, it was recommended that further R & D be performed to realize the energy savings that are potentially possible through the use of internal coatings. This recommendation is reinforced by the possibilities for leakage reduction which have just been identified.

There is ample evidence to show that internal coatings are effective in reducing leakage in pipelines. In one case involving a low-pressure gas pipeline, in-place coating reduced leakage in a 2-mile section of an old line by over 93% [KUT, '67]. Before coating, the line was tested at 100 psi for a period of 24 hours. Leakage amounted to 292,000 cu.ft. daily. After coating, the same test recorded a daily leakage rate of only 19,000 c.ft. It was concluded that nearly complete leakage reduction might be achieved by using a larger application pressure and a greater number of coating runs.

One type of coating receiving growing acceptance for pipeline use is the epoxy coal tar coating, consisting of a blend of coal and epoxy resins with a curing agent. By blending the epoxy with coal tar, water resistance of the coating is improved and, by application of sufficient thickness of the epoxy coal tar coating, good leakage resistance is also achieved.

As noted earlier, further research would be required to clarify the true significance of the reported figures of unaccounted for gas. However, for present purposes, such research is unnecessary, for the same reasons as in the case of the oil lines. That is, the program of internal coating demonstration that was recommended for other reaons in Section 8.2 above will also inhibit leaks if it is conducted with that objective in mind. Since a much thicker coating is required for a leak inhibitor than for a viscosity reducer, in planning the demonstration program it may be desirable to conduct further research into the amount of leakage that exists.

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