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COAL FEEDER DEVELOPMENT PROGRAM

Phase III Report

September 1979

Work Performed Under Contract No. AC01-76ET10260

**Lockheed Missiles & Space Company, Inc.
Sunnyvale, California**

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U. S. DEPARTMENT OF ENERGY

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COAL FEEDER DEVELOPMENT PROGRAM

PHASE III REPORT

**LOCKHEED MISSILES & SPACE COMPANY, INC.
SUNNYVALE, CALIFORNIA 94088**

SEPTEMBER 1979

**PREPARED FOR THE UNITED STATES
DEPARTMENT OF ENERGY
OFFICE OF ASSISTANT SECRETARY FOR FOSSIL ENERGY
OFFICE OF PROGRAM DIRECTOR FOR PROJECT MANAGEMENT
UNDER THE DIRECTION OF R. R. FLEISCHBEIN, P. E. - PROGRAM MANAGER**

**WORK PERFORMED UNDER
CONTRACT NO. EX-76-C-01-1792**

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ABSTRACT

As a result of the work carried out in Phases I and II, the Kinetic Extruder was selected for further development during Phase III.

Design studies were performed to identify components of the Kinetic Extruder which had an important impact on performance. These components were optimized and subjected to testing in the coal feeder test loop.

The improved components were incorporated in the design of the Kinetic Extruder, Model No. 3, which was successfully tested up to 400 psig.

The Kinetic Extruder Model No. 4 was designed to incorporate low differential pressure seals and provision for fluidic turndown control. This machine is ready for testing and engineering evaluation.

CONCLUSIONS AND RECOMMENDATIONS

During the Phase III efforts, emphasis was placed on establishing the design principles for each component of the feeder. The Model No. 3 rotor, which incorporates these principles to ensure smooth feeding of the coal to the rotor, careful attention to the details of the spinup zone, proper sprue profile selection, and sizing of the control orifice, has performed exactly to the design specifications.

To enhance seal life, special journal bearings, acting as a pressure breakdown structure, were selected and incorporated in the design of the Model No. 4 rotor. This rotor also includes provision for fluidic turndown over a 3:1 range.

The capability to design commercial-size feeders with confidence has been established. However, before plant operators will commit to install the novel coal feeder equipment in their plants, data pertaining to durability and maintainability are required.

To further the commercial development of the feeder, the following is recommended:

- (1) Design and build several 1- to 10-ton/h feeders to be placed in pilot plants or small demonstration plants. This action will permit assessing the feeder under actual field conditions and give Process Developers and A and E firms first-hand knowledge of the feeder capability.
- (2) In conjunction with (1) above, develop automated control systems to operate the feeder. These systems have to be tied into the overall plant control scheme because of safety considerations.

- (3) As an alternative to (1) and (2) above, test the feeder in a components test loop to assess durability and maintainability under simulated plant conditions.

Initial studies carried out with different feed stock and surveillance of feed stock properties at a coal-fired power station indicate that controls presently used for control of pulverizers will be compatible with the Kinetic Extruder. No additional restraints need to be imposed.

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Section 1
OBJECTIVE AND SCOPE OF WORK

The reliable feeding of large quantities of dry pulverized coal into pressurized reactors poses a challenging problem. Presently, some installations are using lockhoppers. However, at the high operating pressures and for large throughputs, which will require large valves, these systems are beyond the state-of-the-art, or at best inefficient. Based on the available evidence, the reliability of these systems will also impact plant operation. Slurry systems using either process-derived oil or water are in use or being contemplated. The slurries must be dried before further processing. This step, which has not been demonstrated for large-size applications, is clearly detrimental to overall plant efficiency. At present, no system is commercially available to feed large quantities of dry pulverized coal into pressurized reactors at the large rates projected for future gasification plants. The objective of the present program is to generate sufficient analytical and test data to enable the confident design and fabrication of coal feeders that are compatible with demonstration plant requirements and commercial applications. The present program was performed in the following three phases:

- o Phase I - Selection of Concepts. This phase, of 6 months duration, was designed to review potential candidates and equipment, synthesize designs, assess fundamental problem areas, and define laboratory evaluation techniques. The work accomplished has been reported in Coal Feeder Development Program, Phase I Report, Report FE 1792-8, dated 24 December 1975.
- o Phase II - Laboratory Scale Feeder Development. During this phase of the program, laboratory-size feeders were built and tested in a closed-loop test facility. The data resulting from laboratory testing permit confident design of pilot-plant-size equipment. The work accomplished has been reported in Coal Feeder Development Program, Phase II Report, FE 1792-34, dated July 1977.

- o Phase III - Feeder Evaluation. During this phase of the program, feeders compatible with the existing test loop were designed, built, installed, tested, and evaluated. The data resulting from this effort are sufficient to permit the confident design of commercial-size feeders.

This report describes the effort carried out during Phase III.

Section 2

DESCRIPTION OF THE KINETIC EXTRUDER DRY COAL PUMP

The Kinetic Extruder takes dry pulverized coal from an atmospheric hopper across the pressure barrier and delivers the coal to the high-pressure hopper. The heart of the Kinetic Extruder is a high-speed rotor containing converging radial channels or "sprues." The coal path inside the rotor is illustrated by the cutaway drawing in Fig. 2-1. In operation, pulverized coal is fluidized and fed through the stationary coal feed tube to the center of the rotor. Upon entry into the spinning rotor, centrifugal force causes the pulverized coal to flow into the sprue channels and to form a compacted, moving plug in each of the sprues. The moving plug forms the pressure seal. Excess gases are removed from the rotor through vent channels in the walls of the feed tube by means of a small suction system. The pressure in the center of the rotor is kept slightly subatmospheric.

The coal flowrate through the sprues is controlled by end-hoppers termed "control orifices," located at the outlets of each sprue. The discharge coefficient of the control orifice can be changed by controlling the pressure gradient across the control orifice. This control is accomplished by injecting control gas through the hollow shaft into a gas distribution manifold inside the rotor, and finally into the control orifices. A throttling range of 3 to 1 has been demonstrated with a 30-psi orifice pressure differential. The coal is ejected from the rotor into the pressurized housing and collected by gravity in the receiver. The receiver forms part of the feeder housing.

The pressurized coal is discharged from the receiver through an isolation valve into a high-pressure hopper. The pressure in the receiver is maintained by injecting gas into the receiver to compensate for the high-pressure gas carried from the receiver with the coal. The gas requirement is mainly

Fig. 2-1 Lockheed Kinetic Extruder Rotor

Fig. 2-2 Kinetic Extruder Closeup View

dependent on the coal bulk density in the hopper outlet line. For a dense phase fluidized flow in this line, approximately 1 acfm of gas is required for each ton/h of coal handled.

In addition to the rotor unit itself, the following main subsystems are required to operate the feeder:

- o Drive power system
- o Coolant and lubrication system
- o Rotor eye suction system
- o Pressurization gas system
- o Control gas system
- o Data acquisition and master control system

A view of a typical overall system configuration is shown by Fig. 2-2.

Section 3

SUMMARY OF PROGRESS IN PHASE III

Based on the results of Phase II, two systems were selected for the Phase III part of the development program. These are the Kinetic Extruder and the Ejector. Work on the Ejector started in July 1978, as scheduled, but was discontinued in August of 1978 at the request of DOE, and the development effort concentrated on the Kinetic Extruder.

During the early part of the program, operating parameters and concepts for commercial-size feeders were established. Simultaneously, special fixtures were designed and built to enable evaluation of the several components essential to efficient operation of the feeder. These included the coal feed pipe, the spinup zone, the sprue entrance configuration, the sprue area profiles, and the sizing of the control orifices. Special tests were performed to establish scaling laws for the power required to transfer the coal through the feeder (internal work) and also for the power required to overcome the aerodynamic drag (external work). In addition, the analytical model representing the sprue flow, developed under Phase II, was refined and validated. Using this information, the Kinetic Extruder, Model No. 3, was fabricated and successfully tested at delivery pressures of 415 psia. This pressure level represents the limit of our test capability and does not represent the limit of the feeder.

To this point all rotors were tested with a vertical, cantilever shaft drive system. Design studies indicated that a horizontal, straddle-mount drive system was to be preferred for commercial-size equipment. To verify that the Kinetic Extruder is insensitive to the direction of its axis of rotation, a housing adaptable to the Model No. 3 rotor and featuring a straddle-mounted horizontal axis was fabricated. Tests with the Model No. 3 rotor confirmed that the performance of the feeder was not affected by orientation of the axis.

In parallel with the Model No. 3 fabrication, another experimental rotor was tested. This rotor was to be throttleable by means of a system of gas injection ports. Tests with this configuration were extremely successful and verified the throttling concept. A 3:1 turndown ratio was demonstrated in these tests.

The general evolution of the Kinetic Extruder designs is summarized in Table 3-1. As shown, a total of seven rotors have been fabricated and tested through the entire program. Photographs of the last four rotors are included in Figs. 3-1 through 3-4. Only minor changes in the design of the internals of the rotor have been made since the segmented sprue design used in the sprue evaluation rotor was conceived. All of the rotors built, except for Model No. 1, have been 28 in. in diameter, have contained 12 sprue passages, and have pumped a nominal 1 ton/h of coal.

Because of the excellent test results obtained with the Model No. 3 feeder, an effort was started to design and fabricate the Model No. 4 feeder. This machine, compatible with the existing test loop, incorporates all features of a commercial feeder: the straddle-mounted horizontal axis is supported by water-lubricated journal bearings, low differential pressure seals are used with automatic pressure control, and provision is made for fluidic turndown control. Fabrication is complete and testing with the Model No. 4 feeder can be carried out.

After the planned tests are successfully completed, all data required to design a commercial feeder with confidence will have been generated. The durability and maintainability of the equipment remain to be proven, and some planning and testing will be required to fully integrate the feeder into the master control loop of an automated coal conversion plant.

Table 3-1 EVOLUTION OF KINETIC EXTRUDER DESIGNS

Item No.	Description	Axis and Mounting	Bearing Type	Seals	Performance
1	<u>Model No. 1</u> First Test Rotor (Phase II)	Vertical, Cantilever	Rolling Element	Mechanical Face Full ΔP	Small pressure capability
2	<u>Model No. 2</u> Introduced Stationary Hub in Rotor Eye (Phase II)	Vertical, Cantilever	Rolling Element	Mechanical Face Full ΔP	Pumped to 150 psig, demonstrated basic feasibility of Kinetic Extruder
3	"Sprue Evaluation Fixture," first use of replaceable sprues and control orifices	Vertical, Cantilever	Rolling Element	Mechanical Face Full ΔP	Pumped to 300 psig
4	Equipped for Throttling via Fluidic Control	Vertical, Cantilever	Rolling Element	Mechanical Face Full ΔP	Demonstrated 3:1 turndown ratio
5	<u>Model No. 3</u> All-Up Non-throttleable Design	Vertical, Cantilever	Rolling Element	Mechanical Face Full ΔP	Pumped to 400 psig, met all design goals
6	Model No. 3 Rotor in Horizontal Mounting Enclosure	Horizontal, Straddle	Rolling Element	Mechanical Face Full ΔP	Demonstrated feasibility of horizontal configuration
7	<u>Model No. 4</u> All-up Throttleable Design in Horizontal Configuration	Horizontal, Straddle	Journal	Mechanical Face, Low ΔP	Fabricated but not yet tested

Fig. 3-1 Sprue Evaluation Test Fixture Partial Assembly

Fig. 3-2 Rotor for Kinetic Extruder, Throttleable Unit

Fig. 3-3 Rotor for Kinetic Extruder Model No. 3

Fig. 3-4 Rotor for Kinetic Extruder Model No. 4

Section 4

DETAILED DESCRIPTION OF TECHNICAL PROGRESS

4.1 TEST LOOP CONFIGURATION

The facility used for testing the coal feeder is shown schematically in Fig. 4-1 and is further illustrated by the photographs in Figs. 4-2 and 4-3. In this configuration, the rotor was suspended from the lid of the receiver vessel on a cantilivered shaft, as shown in Fig. 4-4. The feed hopper, inlet coal feed pipe, rotor and drive shaft, and receiver hopper were all stacked on a common vertical axis. The configuration shown was used for most of the test work.

In the latter part of the program, testing was done with a horizontal rotor shaft configuration. The test layout for these tests is given in Fig. 4-5. As can be seen, the rotor is now mounted in a new housing which is mounted on top of the receiver tank. The horizontal shaft rotor is driven from one side of the enclosure, and coal is fed in through the opposite side. The supply tank is offset, requiring a short horizontal run of the coal feed line.

Operation of the Kinetic Extruder was essentially the same in both test-loop configurations. All of the tests of the unit at high-delivery pressure are conducted on a batch basis. The pump cannot be started with a significant pressure difference across the sprue channel. Coal flow must be first initiated while the receiver hopper is at atmospheric pressure, and once the sprue plug is established, the hopper may be pressurized to the test level. To shut down, the procedure is reversed. Each of vessels could hold approximately 1/2 ton of coal. However, allowing for ullage and for time for pressurization and depressurization, steady-state run times were normally about 10 minutes in duration. At low-delivery pressures (< 50 psig), coal could be satisfactorily recycled through the recycle line from the receiver vessel back to the feed hopper while the feeder was operating. In this way, continuous run times of several hours could be achieved.

Fig. 4-1 Facility Schematic - Vertical Shaft Rotor Tests

Fig. 4-2 View of Test Loop — Vertical Shaft Rotor Configuration

Fig. 4-3 Closeup View of Test Loop — Vertical Shaft Rotor Configuration

Fig. 4-4 Kinetic Extruder and Drive Assembly — Vertical Shaft Configuration

Fig. 4-5 Test Facility Layout for Horizontal Axis Kinetic Extruder Tests

Particle Attrition in the Test Loop There are three main reasons for particle attrition in the Kinetic Extruder test loop:

- (1) Attrition during pneumatic transfer of the coal from the lower tank (the transfer is generally made using a 50-psi pressure drop yielding coal velocities in the transfer line of up to 70 fps; the line contains four 90-deg sharp bends)
- (2) Attrition in the lower tank caused by the induced vortex (during feeding at high pressure a major portion of the coal in the lower tank becomes suspended in the strong vortex induced by the rotor),
- (3) Attrition during passage of the coal through the sprues and in decelerating from the rotor rim speed after leaving the sprue orifices.

Permeability Data The clearest manifestation of attrition in the test loop is a decline in permeability of the test coal with time. Figure 4-6 gives permeability data plotted against run time. Run time is defined as the total accumulated time the Kinetic Extruder was pumping coal. This time is discontinuous and may be accumulated over a number of days. The data show a permeability decline of about 20 percent per hour of run time.

The data shown in Fig. 4-6 also show a significantly lower permeability for the coal mixture taken from the test loop before testing when compared with coal taken directly from bags. This may be due to attrition in the transfer line. Coal is initially loaded into the lower tank and then transferred pneumatically to the upper vessel. Samples are obtained from the upper tank; thus, the $t = 0$ data in Fig. 4-6 corresponds to coal that has already been through the transfer line once.

*Subsequently, a grate was installed to kill the vortex in the bottom half of the lower tank. This mechanism was thus alleviated in later tests.

Fig. 4-6 Permeability of Coal Samples From Test Loop
(Lockheed Test at 1000-g Compaction)

Sieve Analyses. Sieve analyses of the test-loop coal samples show very little change with time. Figure 4-7 shows the percent passing the 325- and 400-mesh sieves as a function of time for the same samples that were tested for permeability. As can be seen, the -400 mesh fraction shows no systematic variation with time and the -325 mesh just a barely perceptible increase.

Scanning Electron Microscope Photos. Scanning Electron Microscope (SEM) photographs have been made of some of these coal samples (Figs. 4-8, 4-9, and 4-10). Figure 4-8 shows SEM photos of fresh coal. Figure 4-9 shows a test-loop sample before feeding through the Kinetic Extruder ($t = 0$ data in Figs. 4-6 and 4-7). Figure 4-10 is the sample after several trips through the Kinetic Extruder ($t = 2.2$ h data in Figs. 4-6 and 4-7). Two different views of the sample are included in each case.

As can be seen by comparing the two views of the same sample shown in Fig. 4-10, segregation takes place in loading the microscope slide. Thus, particle-size comparisons cannot be made. However, the individual character of the particles seems to be similar in all cases. There are, for example, no agglomerates of fines in the fresh coal and no evidence of rounding-off corners in the used coal.

Following the above tests, a vortex breakup grating was installed in the receiving chamber and new data were obtained. These new data are shown in Fig. 4-11. As shown, the permeability still decays with time, but it is significantly more gradual than was the case without the grate. As can be seen, with the normal amount of makeup coal additions from time to time, the permeability stabilizes in a band between $3 \times 10^{-12} \text{ft}^2$ and $4 \times 10^{-12} \text{ft}^2$. This degree of variation poses no operating problems.

Conclusions. Particle attrition in feeding coal through the Kinetic Extruder is not a significant problem. A gradual decrease in coal permeability is observed with multiple passes around the test loop. This is due to an increase

Fig. 4-7 Sieve Analysis of Coal Samples From Test Loop

Fig. 4-8 SEM Photographs of Fresh Coal From Bag

Fig. 4-9 SEM Photographs of Coal From Test Loop. Coal has been pneumatically transferred from lower to upper tank but has not been fed through the Kinetic Extruder

Fig. 4-10 SEM Photographs of Coal From Test Loop After 2.2 h Run Time
(several trips through Kinetic Extruder)

Fig. 4-11 Permeability of Test Loop Samples as a Function of Kinetic Extruder Run Time

in fines but the change is too subtle to show up in sieve analyses or SEM photos. Furthermore, it appears more likely that the recycle through the transfer line and the particle impact in the vortex are responsible for the attrition, rather than passage through the Kinetic Extruder.

4.2 COMPONENT EVALUATION

Evaluation of each component of the Kinetic Extruder was planned in order to understand its operation and the physical principles involved, and to determine the scaling and optimization laws. These studies, described in detail in this subsection, address the design of the:

- o Coal flow passages to ensure smooth and efficient operation
- o Rotor to reduce power consumption
- o Bearing and seal configuration for long-life expectancy and cost effectiveness

4.2.1 Sprue Entrance Configuration

To evaluate the permissible flow angles at the sprue walls, the flow characteristics of the pulverized coal in the centrifugal force field had to be determined.

The "g" fixture shown in Fig. 4-12 was designed and fabricated to facilitate this determination. The fixture is mounted at the rotating shaft of the Kinetic Extruder. Coal is fed into the device while rotating, and the device is stopped after the coal flow is terminated. The unit is disassembled, and the shape of the trapped coal and wear patterns recorded. Before each test the interior surfaces are coated with a special paint which easily erodes, thereby revealing the flow patterns. From these observations, the design of the sprue entrance zone was determined. The flow angles which can readily be determined with the "g" fixture are:

- α = internal angle of slip in moving coal
- β = angle of free surface of material
- γ = minimum angle of slide on metal wall

Fig. 4-12 "g" Test Fixture

Figure 4-13 illustrates the appearance of the angles α and β in a draining gravity flow bin.

Since, in gravity flow, the acceleration vectors are parallel, the demarcation of the flow regimes forming a constant angle with the acceleration vector are straight lines as shown in Fig. 4-13.

When coal is subject to a rotating force field, as is the case in the "g" test fixture or the rotor of the Kinetic Extruder, the acceleration vector passes through the center of rotation of the fixture as shown in Fig. 4-12, and lines having a constant angle with respect to the acceleration vector are no longer straight lines but become curved as shown in Fig. 4-14.

Using polar coordinates, the shape of the curves is given by

$$r = r_1 \exp\left(\frac{\theta_1 - \theta}{\tan \epsilon}\right)$$

where

r_1 = radius at outlet opening

θ_1 = angle at outlet opening

ϵ = specified constant angle between curve and acceleration vector

Curves having a constant angle ϵ varying between 9 and 30 deg have been calculated using the geometry of the "g" test fixture and are shown, using x-y coordinates, in Fig. 4-15. In the "g" fixture, the demarcation lines shown as straight lines in Fig. 4-13 should appear similar to the curved lines of Fig. 4-15.

Tests to Determine the Angle β . To determine the angle β , which is the angle that the free surface of the coal makes with the acceleration vector, sufficient coal was fed into the rotating "g" test fixture to ensure that it

Fig. 4-13 Illustration of Angles α and β in a Bin

Fig. 4-14 Shape of Curves Having a Constant Angle With the Acceleration Vector in a Rotating Field

Fig. 4-15 Family of Curves Having a Constant Angle, ϵ . With the radial vector. (r_1, θ_1) correspond to "g" fixture outlet location

was filled to capacity. To achieve this objective, about 2 gallons of pulverized coal were fed into the fixture. The rotor was then stopped and the fixture disassembled. Figure 4-16 is a photograph of the bottom portion of the fixture after a test.

As indicated in Fig. 4-16, the angle β is larger on the windward side surface than the leeward side. This difference is probably due to an erosive effect of the impinging coal particles.

Tests performed at different rotational speeds showed no g-force effect on the angle β . An angle of 50 deg can be used for design purposes.

Tests to Determine the Angle α . The angle α denotes the interface between the moving coal and the stationary or trapped coal. It can be observed only under moving conditions. To discern this angle, the surfaces of the "g" fixture were coated with an easily abradable paint. The moving coal particles would abrade the paint, leaving a clean metallic surface, while the stationary coal would leave the paint intact. The wear patterns of the paint would thus reveal the coal slip lines.

Two types of paint were successfully used: a water-base artist's poster paint and a water slurry made from chalk dust. The chalk dust gave the most striking patterns. During these tests, coal was fed through the "g" fixture from the upper tank at about a 1 ton/h rate. Runs were of approximately 10-min duration.

Figure 4-17 shows the wear patterns left on the chalked lower plate after a test run. Residual coal has been removed before taking the photograph. Measurements taken from the photograph indicate an angle α of 11 to 12 deg over most of the flow channel. There is an increase in α to approximately 20 deg very close to the outlet points. The demarcation line closely resembles the curves shown in Fig. 4-15 for a constant angle of about 11 deg.

Fig. 4-16 "g" Fixture Test Results Showing the Angle β

Fig. 4-17 "g" Fixture Test Results Illustrating Angle α . Wear pattern in chalked lower plate is shown with residual coal removed. 2140 rpm, 1.09 tons/h

Tests to Determine the Angle γ . The slide angle γ is defined as the maximum angle of slide along a solid surface. In the design of a hopper, it would be the largest permissible hopper angle for the hopper to run full and to flow across its entire cross section.

The importance of being able to use large wall slope angles was demonstrated during rotor optimization studies. As the permissible wall slope angle is increased, fewer sprues are needed, and a smaller-diameter rotor requiring less power to drive it can be designed.

An initial test series, using straight metal wall sections mounted at the outlets of the "g" fixture, indicated that the maximum permissible wall angle was approximately 20 deg. Tests were then conducted using constant wall angles. Surfaces of constant wall angle, similar to the curves presented in Fig. 4-15, were manufactured and mounted in on the "g" test fixture. The curves correspond to γ angles of 16 deg, 20 deg, 25 deg, and 30 deg.

The test procedure was the same as used previously. Figures 4-18 and 4-19 show results from a test run of about 8-min duration at a coal feed rate of 1.5 tons/h and at a 2480-rpm rotational speed. The wear patterns of the chalked fixture surfaces indicate that the $\gamma = 16$ -deg channel was definitely running full. The $\gamma = 20$ deg channel appears to have been close to running full, but the 20-deg channel outlet became plugged and the wear pattern was less well defined. The $\gamma = 25$ -deg and 30-deg channels clearly were not running full. Based on these tests, a maximum wall angle of 16 deg was selected as a design value for the Kinetic Extruder passages.

4.2.2 Sprue Design Evaluation

Design studies leading to the optimization of the sprues indicated the desirability of decoupling the gas-sealing function of the sprue from the coal-metering function. These considerations and difficulties encountered during Phase II in operating the Kinetic Extruder Model No. 2 led to the general sprue design shown in Fig. 4-20. Coal enters the sprue through the funnel and into the sprue body where most of the pressure drop is sustained. The coal plug velocity in the sprue is controlled by the control orifice.

Fig. 4-18 Views of Lower Section of Test Fixture, 2480 rpm, 1.5 tons/h.
 (A) residual coal pattern; (B) coal brushed off exposing wear
 pattern on chalked surface

Fig. 4-19 Views of Upper Cover Plate, 2480 rpm, 1.5 tons/h.
 (A) original condition; (B) coal brushed off exposing wear
 pattern in chalked surface

Fig. 4-20 Rotor for Sprue Evaluation Test

To test this novel design concept and to determine an empirical relationship between the coal plug permeability as measured in the laboratory apparatus and as observed in the coal feeder, a sprue evaluation test fixture was designed and built. This unit used the interchangeable sprue-funnel-control orifice design concept illustrated by Figs. 4-20 and 4-21. As shown in Fig. 4-20, the funnel and sprue are assembled by a screw joint and are inserted into the rotor. The assembly is held in place by the control orifice wedge. Figure 4-21 is a photograph of one set of components for such an assembly.

The rotor contains 12 such sprue assemblies. Sizes for the various sets of hardware fabricated for testing are given in Table 4-1. Larger-size sprues are used for fine low-permeability grinds, and the smaller diameters are optimized for coarser, highly permeable coal. The rotor was mounted in the test loop in the vertical configuration. A partial assembly (bottom cover plate removed) of the mounted rotor is shown in the Fig. 4-22. During operation, the inner hub containing the coal entrance opening is stationary, and the rotor revolves around it.

Table 4-1. AVAILABLE SPRUE BODIES, FUNNELS, AND CONTROL ORIFICES

Type	Number Available	Outlet Diameter (in.)	Inlet Diameter (in.)	Area Ratio A_{23}
Sprue Body	12	0.164	0.283	3
Sprue Body	12	0.231	0.401	3
Sprue Body	12	0.327	0.567	3
Sprue Body	12	0.164	0.201	1.5
Sprue Body	12	0.189	0.231	1.5
Sprue Body	12	0.231	0.283	1.5
Sprue Body	12	0.283	0.347	1.5
Sprue Body	12	0	—	—
Funnel	12	0.283	1.625	33.6
Funnel	12	0.401	1.625	16.4
Funnel	12	0.567	1.625	8.2
Control Orifice	12	0.082	0.6	—
Control Orifice	6	0.093	0.6	—

Fig. 4-21 Photograph of Elements Comprising a Sprue Evaluation Assembly

Fig. 4-22 Sprue Evaluation Test Fixture, Partial Assembly

The sprue evaluation fixture performed quite well in testing. The control orifices were found to function exactly as designed in forming and maintaining a stable moving sprue plug. Delivery pressures up to 300 psig were obtained. Total accumulated test time for the rotor was approximately 14 h. The $d_3 = 0.283$ in., $A_{23} = 1.5$ set of sprues was found to be optimum for the 70-percent, 200-mesh test coal in use.

Figures 4-23 and 4-24 show a comparison of some of the test data with theoretical predicted limiting curves. Figure 4-23 gives the comparison of the data with theory for a permeability, k , of 4×10^{-12} ft², and Fig. 4-24 is for $k = 5 \times 10^{-12}$ ft². It was not clear in the tests whether the maximum pressure limit encountered during operation was due to an insufficiency of rotational body force or increasing gas back leakage. If it was the former, then Fig. 4-22, with $k = 4 \times 10^{-12}$ ft², shows the better agreement between theory and experiment. If it was the latter, then $k = 5 \times 10^{-12}$ ft² would be the preferred value. In any case, the effective value of the moving coal plug in the sprues was shown to be somewhere in the 4 to 5×10^{-12} ft² range. Bench permeability tests on coal samples taken during the test period gave values in the 3 to 4×10^{-12} ft² range at similar compacting forces. This indicates that the permeability of the moving coal is higher than that of static plugs, as might be expected. However, the correction factor which needs to be applied appears to be fairly small. This was confirmed in later, more careful tests with the Model No. 3 rotor. The analytical model of the sprue flow, and comparisons of predictions with the more detailed data from the Model No. 3 rotor tests, are discussed in later sections of this report.

4.2.3 Rotor Drag Determination

The total power required to operate the Kinetic Extruder consists of two parts:

- (1) "Internal work," i.e., the work done on the coal while inside a sprue in accelerating it up to the rotor rim speed and in overcoming friction and pressure forces
- (2) "External work," i.e., work done to overcome the aerodynamic windage on the exterior surfaces of the rotor

Fig. 4-23 Comparison of Sprue Evaluation Test Data With Theory
for $k = 4 \times 10^{-12} \text{ ft}^2$

Fig. 4-24 Comparison of Sprue Evaluation Test Data With Theory
for $k = 5 \times 10^{-12}$ ft²

The internal work is directly proportional to coal throughput, while the windage is a more or less fixed parasitic loss and only increases with throughput insofar as wheel size increases. In small-scale (e.g., 1 to 5 tons/h) high-pressure feeder units, the aerodynamic drag loss is, by far, the dominant term. In large units, where the windage is distributed over a large number of tons/h, it no longer dominates.

The external work can be calculated from the semi-empirical relationship:

$$V_E = C_m \frac{1}{2} \rho \omega^3 r^5 \left(1 + \frac{5}{2} \frac{h}{r} \right) \quad (4.1)$$

where

- V_E = external power requirements
- r = rotor radius
- h = rotor rim thickness
- ω = rotor speed
- ρ = gas density external to the rotor
- C_m = empirical torque coefficient

The empirical torque coefficient, C_m , requires experimental measurement since the data available in the literature could not be applied with any degree of confidence to a high-speed rotor in coal dust environment.

Information was also desired to determine whether the gas density ρ had to be modified to account for the fact that coal particles are entrained with the gas, and verification was needed that the correction term, $(1 + 5/2 h/r)$, could be used to predict the effect of rotor thickness on torque requirements. To obtain the required data, a series of tests was conducted using simple rotors fabricated from sheet metal and also with the Kinetic Extruder rotors as they became available.

The disks were mounted in the test fixture in place of the Kinetic Extruder rotors, as shown in Fig. 4-25. Figure 4-26 shows a photograph of one drag test rotor mounted on the drive shaft before its installation in the test loop. Figure 4-27 illustrates the type of the disk configurations tested.

Fig. 4-25 Thick Disk Test Configuration

Fig. 4-26 Thick Disk Drag Test Rotor

Fig. 4-27 Disk Configurations Tested

Shaft torque was measured by means of a torque meter affixed to the hydraulic drive motor. The meter was calibrated by means of dead weights. Experimentally, the net aerodynamic torque was determined by subtracting gearbox-related drag and, when required, the small internal coal work term from the gross shaft torque data. Gearbox losses were determined from no-flow tests at atmospheric pressure, normally made just before the test run. The theoretical expression $(mr^2\omega)$ was used for the internal work correction.

Figure 4-28 shows typical torque-speed curves obtained in a rotor drag test without coal flow. Curve 1 shows the shaft torque measured while the rotor is spinning in a 200-psig atmosphere. Curve 2 gives the torque at 0 psig which is predominantly due to parasitic drag in the oil-filled gearbox.

The torque difference between Curves 1 and 2 is caused by the aerodynamic drag, and this difference is used to determine the torque coefficient.

Figure 4-29 shows the torque coefficient calculated from the measured torque data for the various disk configurations as a function of disk Reynolds number

$$Re = \left(\frac{\rho \omega r^2}{\mu} \right)$$

The theoretical curve for a thin, hydraulically smooth disk is also shown for comparison.

Several conclusions can be drawn from the data. First, the thickness correction term

$$\left(1 + \frac{5}{2} \frac{h}{r} \right)$$

is reasonably successful in correlating the data for the three disk thicknesses tested. Second, the single disk data are higher than the theoretical curve and show no decreasing trend with Reynolds number. As discussed in Appendix B, this behavior would indicate that the disks are effectively hydraulically rough. Finally, the paired thin disks show considerably less drag than the single disk of the same overall thickness (7.6 in.). This finding

Fig. 4-28. Typical Torque Data for Rotor

Fig. 4-29 Torque Data for Thick Disk Test Series

indicates that Kinetic Extruders with multiple tiers of sprues should be left unshrouded.

Figures 4-30 and 4-31 give similar torque coefficient data for the sprue evaluation rotor and the Model No. 3 rotor. The dashed line in Fig. 4-30 represents data obtained for the sprue evaluation rotor at 250 psig and no coal flow (sealed sprue passages). The plotted data points are from coal pumping tests at various delivery pressures. These data are somewhat scattered but indicate that the torque coefficient is not dependent on whether or not coal is being pumped through the device. In other tests it was verified that no change occurred in C_m as the quantity of coal in the receiver tank was varied from nearly empty to nearly full. Finally, it may be noted that the sprue evaluation rotor data shown in Fig. 4-30 are consistent with the disk test series data shown in Fig. 4-29.

Figure 4-31 provides a comparison of drag test data obtained with the Model No. 3 rotor and the C_m data obtained from the disk tests. All of the Model No. 3 data were obtained while coal was pumped at pressures ranging from 200 to 400 psig. Clearly, C_m exhibits no pressure dependence. The apparent average C_m value for the Model No. 3 rotor, about 0.0085, is somewhat higher than the value of approximately 0.007 found in the tests with the other rotors. At this time no explanation for this change is apparent. A value of $C_m = 0.0085$ has been adopted for use in design studies.

Polished Disk Drag Tests. A limited series of drag tests was conducted with a highly polished disk (20 to 30 μ -in. rms) in an attempt to reduce the C_m values toward the hydraulically smooth value. This was generally unsuccessful. After a number of tests it became obvious that fine coal dust adhering to the rotor surfaces was creating roughness. Upon disassembly after a test, the rotor invariably was quite dirty. Frequently, when the rotor was first started at a slow speed, it would exhibit a high C_m value. After running at high speed and high pressure, the C_m value would reduce. The cause appeared to be that initially adhering of considerable coal to the rotor

Fig. 4-30 Torque Coefficient With and Without Coal Throughput for Sprue Evaluation Rotor (Rim Thickness = 0.625 in.)

Fig. 4-31 Torque Coefficient for Kinetic Extruder Model No. 3 (Rim Thickness = 1.125 in.) Compared With Disk Test Series Data

made it rough. Running at high speed and pressure evidently removed some of this roughness causing coal dust. However, the torque coefficients were never significantly lower than those presented in Fig. 4-29.

Shrouded Rotor Tests. Early in the Phase III program, drag tests with the Model No. 2 rotor enclosed in a shroud were conducted. The shroud configuration is shown in Fig. 4-32. The purpose of the shroud was to reduce drag by two mechanisms:

- (1) The fact that the rotor drag in a tight housing is alleged to be lower than that of a free rotor
- (2) Reduction of the dust loading in the vicinity of the rotor

As shown in Fig. 4-32, the shroud was equipped with ports that could be opened or plugged to allow varying amounts of gas to circulate through the shroud.

Figure 4-33 summarizes the drag test results obtained using the shrouded rotor. As stated, the Model No. 2 rotor was used for the tests. The curves labeled "Free Rotor" correspond to the unshrouded Model No. 2 rotor; the curve labeled "Shrouded Rotor - No Flow" corresponds to the shroud configuration with all of the ports plugged; finally, the curve labeled "Shrouded Rotor + flow" indicates that some of the ports were open allowing gas circulation. As shown in Fig. 4-33, the data at least show the expected trend in that

Free rotor drag > shrouded rotor + flow drag > no flow drag

However, the improvement using the shroud was quite modest. In view of the small benefits and additional assembly work when the test rotors were mounted, the lower portion of the shroud was not used in the remainder of the program.

During tests with the shroud in place, the rotor induces a considerable gas flow through the shroud. This flow was measured by means of pressure taps across the ports where gas entered the shroud. The flowrate is computed from:

Fig. 4-32 Shroud Configuration

Fig. 4-33 Effect of Shrouding on Torque Coefficient

$$Q = C_D A \sqrt{\frac{2 g \Delta P}{\rho}}$$

where

- Q = volumetric flowrate
- C_D = orifice discharge coefficient, assumed to be 0.6
- A = port area
- ρ = gas density
- ΔP = (tank pressure-port pressure tap pressure)

The flowrate data are given in Fig. 4-34.

4.2.4 Control Orifice Evaluation

Two basic control orifice configurations have been under consideration for the Kinetic Extruder. These are the conical control orifice shapes and the simple orifice plate as illustrated in Fig. 4-35. Both have been used in the 1 ton/h unit, and each has its potential advantages. The advantage of the conical shape is that for a sufficiently small cone angle, δ , a uniform mass flow across the hopper section is assured. Undesirable stationary pockets of coal, which cannot be avoided with the orifice plate design, are eliminated. However, since coal is rubbing on coal with the orifice plate design, this type is less subject to wear. It also would be more easily fabricated of wear-resistant materials.

To minimize driving power requirements, it is desirable to make the control orifice length, i.e., the distance between the sprue outlet and the orifice outlet, as short as possible. Since the orifice plate should be located far enough from the sprue outlet so that the region of stationary coal does not reach into the sprue proper, and since as shown in subsection 4.2.1, the wall slide angle exceeds the internal slide angle, the conical configurations can be made shorter than the orifice plate configuration.

Fig. 4-34 Induced Gas Pumping Rate Through Shroud (Two Sides)

$\delta = 13^\circ, 22^\circ$

(a) Conical Control Orifice

(b) Flat Orifice Plate

Fig. 4-35 Control Orifice Configurations (1 ton/h Unit)

Tests of the control orifices designs in the throttleable rotor showed that the conical types were more responsive to throttling. Two cone angles were tested $\delta = 13$ deg and $\delta = 22$ deg. Similar data were obtained for the flat plate orifice configuration. These data are presented in Fig. 4-36. As shown in that illustration, the flat-plate orifices require twice the control orifice ΔP as the conical orifices to produce the same increase in throughput.

The 22-deg conical orifices performed indistinguishably from the 13-deg models from the standpoint of throttleability. However, some coal deposits left in the 22-deg orifices indicated a funnel flow type of behavior. This behavior caused no problems in the tests but may be undesirable over long periods.

Based on the above considerations, the conical control orifice is the favored design at this time, with a cone angle between 13 and 22 deg. The exact cone angle that can be used should be further verified in long-duration tests by using simultaneously two or more designs in the same rotor.

4.2.5 Feed Pipe Tests

The diameter of the feed pipe, i.e., the channel through which coal is fed into the eye of the Kinetic Extruder rotor, is an important parameter. This dimension should be minimized to the extent possible to allow the use of a smaller shaft and shaft seal.

Tests were conducted to determine the coal throughput as a function of pressure differential for the feed pipe. Three configurations were tested as illustrated in Fig. 4-37. Results are shown in Fig. 4-38. Since the Kinetic Extruder rotor was not mounted for these tests, the results reflect the feed pipe characteristics only.

Based on the results, a conservative flowrate of 2 tons/h/in.² was adopted for sizing feed pipes for various scale units. For a 50 ton/h unit, this corresponds to a feed pipe ID of 5.6 in.

Fig. 4-36 Throttleable Rotor Test Data

Fig. 4-37 Coal Feed Pipe Configurations Tested

Fig. 4-38 Feed Pipe Flowrate as a Function of Pressure Differential

4.2.6 Spinup Zone Evaluation

The power required to overcome the aerodynamic drag increases rapidly with rotor diameter. This diameter is influenced by the space required for the spinup zone, which is the space between the feed pipe outlet and the sprue inlet. Here, the coal is accelerated to the rotational velocity of the rotor and is swung out into the sprues. A series of tests with the Model No. 3 rotor were performed to determine the adequacy of the present design, to establish a benchmark for scaleup purposes, and to assess the potential for reducing the size of this zone. In the Model No. 3 rotor, the spinup zone lies between a 6- and 8-in. diameter area.

Using a series of control orifices of increasing diameters and thereby producing higher throughputs than the nominal 1-ton/h rate, the ultimate capacity of the present spinup zone was determined. A secondary output of the test series was the verification of the control orifice scaling law.

Three sets of flat-plate control orifices were tested. Orifice diameters used were 0.125, 0.136, and 0.153 in. in comparison to the normal 0.093-in. size which yielded a 1-ton/h throughput. Table 4-2 summarizes the data obtained in the test series. A maximum throughput of 2.50 tons/h was obtained using the 0.136-in. orifices at 3200-rpm rotor speed. The coal plugs were not maintainable in the sprue at higher throughputs. When the 0.153-in. orifices were used, the sprue plug could not be established without slowing the wheel to about half speed. Once the plug was formed, the rotor could be accelerated to a certain extent without ill effect. However, when a speed of about 2720 rpm was reached, the sprue plug seal was lost once again. At this speed, the throughput would have been 2.65 tons/h.

According to these tests, the maximum capability of the present feed pipe, spinup, and venting system configuration is 2.5 tons/h. Based on previous calibrations of the feed pipe and venting systems, it appears that the spinup region represents the choke point. As a final check to confirm this finding,

Table 4-2. SPINUP ZONE EVALUATION DATA

Control Orifice Diameter (in.)	Rotor Speed (rpm)	Throughput (ton/h)	C_D (lb/s/in. ^{5/2})	Comments
0.125	3200	2.03	0.268	Normal operation.
0.136	3200	2.50	0.267	Normal operation.
0.153	1780	1.73	0.247	Sprue plug could not be established at higher speed.
0.153	2720	2.65(a)	-	Sprue plug lost upon acceleration to this speed.
0.093	3530	1.00	0.248	Earlier data.
0.153	3300	3.20	-	Small impeller installed. Normal operation.

(a) Calculated value.

a small sheet metal impeller was installed in the spinup zone. It was found that this significantly increased the obtainable throughput, thus verifying that the spinup zone formed the choke point.

Using the data shown in Table 4-2, it is also possible to confirm the control orifice scaling law for flat control orifice plates

$$\dot{m} = C_D d^{5/2} \left(\frac{r \omega^2}{g} \right)^{1/2}$$

The calculated values of the discharge coefficient, C_D , are included in Table 4-2. As can be seen, the average value for these flat-bottomed orifices is $C_D = 0.26 \text{ lb/s/in.}^{5/2}$, and is essentially independent of orifice size and rotor speed.

4.2.7 Rotor Vent Line Calibration

To enable estimation of rotor eye vent line flow rates in the Model No. 3 tests from the measured pressure drop across the lines, the flowrate versus pressure differential relationship was established for nitrogen gas. A rotometer was used to measure gas flow.

The test data were correlated using the following equation:

$$Q_V = 0.71 (P_1 - P_2)^{1/2} \left(\frac{P_1}{14.7} \right)^{1/2}$$

where

Q_V = vent line flow (scfm)

P_1 = suction line inlet pressure (psia)

P_2 = suction line outlet pressure (psia)

The results is shown in Fig. 4-39. The vent gas flowrate calculated using the above equation during coal pumping tests will tend to be larger than the actual flow, because 2-phase flow effects are not accounted for. During coal flow experiments, some coal dust is entrained with the vent line gas flow, and this reduces the flowrate for the measured pressure drop. Thus, the actual flowrate will always be lower than the calculated value based on observed line pressure readings.

The data shown are used to estimate the rotor leakage rate and to size the rotor suction system capacity requirement for commercial installations.

Fig. 4-39 Calibration of Rotor Vent Suction Lines

4.2.8 Sprue Flow Detector Evaluation

Two types of instrumentation have been developed for monitoring the coal flow from individual Kinetic Extruder sprues. The first type detects the impact of the coal streams by means of a piezoelectric transducer mounted near the rotor rim. The second uses a light source and photodetector with the coal streams interrupting the light beam. Figures 4-40 and 4-41 are photographs of these two devices. The signals from both detectors are displayed on an oscilloscope with the sweep exactly synchronized with the rotor revolutions by means of a magnetic pickup and marker on the rotor. Individual sprue flow are readily identifiable on the display.

Both flow-monitoring devices operate satisfactorily. Of the two, the piezoelectric detector has to date been more reliable; however, this device has inherent wear problems at the coal impact point. Thus, the optical monitoring system is probably the better answer for a commercial-type feeder.

Typical signatures from the piezoelectric device are shown in Figs. 4-42a and 4-42b. In each figure, the upper position indicating trace is a display of a magnetic pickup signal and defines a fixed reference point on the rotor adjacent to sprue 1. The lower trace shows the transducer signal as the coal streams from each of the 12 sprues strike the detector.

Figure 4-42a shows the normal situation with all sprues flowing. As Fig. 4-42b shows, sprue 4 is plugged.

This display is used to take corrective action to restart the flow should a sprue become plugged. This action is taken without any interruption of the test run. The sprue unplugging simply consists of a small gas nozzle pointed at the rim of the rotor. Should a sprue become plugged, this gas jet is turned on and invariably restarts the flow almost instantaneously. The device has been extensively tested and is now being automated.

Fig. 4-40 Piezoelectric Sprue Flow Monitor

Fig. 4-41 Optical Sprue Flow Monitor

POSITION INDICATOR

NORMAL FLOW

(a)

POSITION INDICATOR

SPRUE 4
PLUGGED

(b)

Fig. 4-42 Piezoelectric Sprue Flow Monitor Signatures

The optical detector uses a light beam and a photodetector to observe the passing of coal jet issuing from the sprues. This type of detector is subject to degradation of signal strength due to dust deposition under optical surfaces. Buffering by the use of a purge gas jet is beneficial as shown in Fig. 4-43, which illustrates the signal recorded by the detector system as the purge gas flow is actuated and increased to its optimum value. The signal received under these conditions is much sharper than the piezoelectric gage output shown in Fig. 4-42.

4.2.9 Pressurization Transient Tests

Startup of the Kinetic Extruder, i.e., the establishment of the moving coal plugs, must take place at zero backpressure. To bring online a Kinetic Extruder unit, such as that illustrated by Fig. 4-44, the coal flow is started by opening valve A while the rotor housing is at ambient pressure and isolated from the high-pressure reactor by the closed valve B. Once the coal plugs are established, the unit is pressurized to the reactor level, and valve B is opened. To shut the feeder down, the procedure is reversed. Valve B is closed, the housing is depressurized, and then the coal flow through the rotor is shut off by closing valve A.

Since the unit is feeding coal during pressurization, the rotor enclosure vessel must be sized so that the coal accumulation built up during this transient may be accommodated. The size of the hopper portion of the unit thus depends on the pressurization rates that can be achieved in a stable manner.

Theoretically, only a finite pressurization rate can be tolerated by the dynamics of the coal plugs. The tolerable rate is quite high from a practical standpoint. As long as the time scale for pressurization is much greater than the time constant of the sprue flow, t_s , the sprue pressure distribution will not differ significantly from the steady-state profile. The sprue flow time constant is given by

No Gas-Purge
Flow

Low-Purge
Gas Flow

Optimum-Purge
Gas Flow

Fig. 4-43 Optical Sprue Flow Detector Signatures

Fig. 4-44 Lockheed Kinetic Extruder

$$t_s = \frac{L_s}{U_s}$$

where

L_s = sprue length

U_s = average coal velocity

A typical value would be $t_s \approx 1/2$ s. Thus, a pressurization time of the order of, say, $10 t_s$, or 5 s, should be comfortably tolerated.

As an experimental check, some pressurization tests were conducted using the Model No. 3 rotor mounted in the horizontal axis housing. A typical pressure transient is shown in Fig. 4-45, which was traced from the strip chart record of one test. Using the present gas supply line sizes, the highest receiver vessel pressurization rate that could be obtained was: 0 to 200 psig in 20 s (600 psi/min). No sprue flow interruptions or other untoward effects were observed during the pressure transients. This would be expected considering the sprue flow time constant. The tests also showed that typically only 10 to 20 s elapsed between opening the coal feedline valve and the establishment of a pressure sealing coal plug.

To conclude, based on startup transient considerations, the housing hopper section of the feeder need be able to accommodate only a 1/2- to 1-min coal delivery.

4.2.10 Wear Evaluation

As the coal passes through the passages of the rotor, the cross-sectional area decreases, causing an increase in coal velocity. The maximum velocity encountered is in the outlet of the control orifice where a velocity of about 30 ft/s is reached. Wear has been observed in the area upstream of the outlet orifice during previous testing.

Fig. 4-45 Pressure Transient in Tests of Kinetic Extruder Model No. 3

One-piece aluminum orifice/orifice holders were used in the sprue evaluation rotor. These parts showed definite signs of wear in the conical section after a few hours' run time. The controllable rotor, and subsequently designed rotors, are equipped with replaceable orifices. The following conical control orifices materials were screened: six hardened steel, three Ni-resist, and three TZM alloy orifices.

After about 20 h of testing, the six hardened steel, three Ni-resist, and three TZM alloy orifices were removed and sent to the Metallurgical Laboratory for examination. Microscopic inspection shows minor evidence of wear about 0.6 in. from the entrance and also around the approach to the cylindrical throat section. The cylindrical throat section is about 1/8 in. long, and no wear is apparent in the area. In fact, machine tool marks are still visible. Of the materials tested, Ni-resist appears best from an overall performance viewpoint.

Kennametal Grade 701 orifice inserts and Boron carbide inserts have been procured. LMSC plans to use these wear-resistant inserts in future tests.

For commercial units, the sprue entrance funnels and the sprue bodies will also be fabricated of wear-resisting materials. Stellite 6 appears to be a promising candidate at this time.

4.2.11 Rotor Mounting and Seal/Bearing Configurations

To determine the bearing and seal sizes and induced loads, the type of closure used must be specified. Candidate concepts are shown in Fig. 4-46 for the alternative Kinetic Extruder design. For each closure type, the rotor may be cantilever-mounted, or a straddle mount can be used. Five types of basic concepts were evaluated. For screening the designs and the selection of promising candidates for further study, sketches were prepared for each type of rotor support. These are shown in Figs. 4-47 through 4-51. The preliminary sketches were used as a starting point for the selection of bearing and seal designs. The seal and bearing design study was carried out by Mechanical

DISHED HEADS

FORGINGS

DISHED HEADS

CASTINGS -
WEBBED

DISHED HEADS

SPHERICAL, SPLIT CASE

DISHED HEADS

Fig. 4-46 Potential Closures for the Kinetic Extruder

Fig. 4-47 Schematic of Concept (a) - Cantilever With Thrust Bearing on Top

Fig. 4-48 Schematic of Concept (b) - Straddle Mount With Thrust Bearing on Bottom

Fig. 4-49 Schematic of Concept (c) - Straddle Mount With Thrust Bearing on Top

Fig. 4-50 Schematic of Concept (d) - Straddle Mount, Vertical Rotor

Fig. 4-51 Schematic of Concept (c) Semistraddle/Cantilever With Internal Drive Shaft

Technology, Inc., of Latham, New York, under subcontract to Lockheed. The complete design study report is included in Appendix A.

Based on the results of this study, a straddle-mount, horizontal shaft, journal bearing configuration was selected. To obtain long seal service life, the journal bearings are specially designed to act as pressure breakdown structures and relieve the high-pressure drop across the outboard seal which is normally associated with this type of machine. This is illustrated in Fig. 4-52.

In addition, water was selected as the lubricant because of good cooling ability. An antifreeze such as Prestone is added to protect the equipment during shutdown in cold weather.

Because of continuing effort to test 1-ton/h units closely resembling commercial-size feeders, we have designed the Model No. 4 rotor system to the above specifications. This feeder is mounted in an horizontal housing as shown in Fig. 4-53, is equipped with water-lubricated journal bearings, and also is provided with the capability for fluidic turndown control of the coal feed rate.

For the Model No. 4 rotor, two water supplies are used. One is a high-pressure supply (about 500 psi) used to supply the main bearings; the second is a low-pressure system (about 50 psi) and is used to supply the locating bearing situated between the stationary coal feed tube and the hollow rotor shaft. The high-pressure supply is regulated by a control valve to maintain a bearing injecting pressure of approximately 30-psi differential over the coal-delivery pressure. This limits the ΔP across the mechanical face type shaft seals to a maximum of 30 psi, thus ensuring long seal life.

B E F O R E

OUTBOARD SHAFT SEAL -
HIGH OPERATING PV VALUE
INBOARD SHAFT SEAL -
LOW OPERATING PV VALUE

A F T E R

OUTBOARD SEAL-BEARING JOURNAL
AND SHAFT AS PRESSURE BREAK-
DOWN STRUCTURE.
INBOARD SHAFT SEAL -
LOW OPERATING PV VALUE

Fig. 4-52 ΔP Across Seals

SECTION A - A
SCALE: 1/4"

Fig. 4-53 Assembly of Kinetic Extruder

4.2.12 Vertical Shaft Flow Simulator

A transparent flow visualization device, simulating the vertical shaft Kinetic Extruder, was fabricated and used in particle-flow studies in the Fluid Mechanics Laboratory. A schematic of the unit is shown in Fig. 4-54.

Experiments with the flow visualization apparatus were conducted to explore the dispersed particle flow which is induced by the rotating disk. For this purpose, the flow of coal powder within the high-pressure environment coal feeder vessel was simulated using glass microballoons with a density of 0.17 g/cm^3 and atmospheres of air and freon F-12 gas (CCl_2F_2). The use of the glass particles whose size ranged from 0.05 to 0.2 mm in air or F-12 allowed closer match of the ratio of particle density to gas density to that encountered in the N_2 or CO_2 atmospheres of the coal feeder at operating pressures. Thus, by varying the gas within the apparatus from air at 14.7 psia (1 atm) to F-12 at 14.7 and 36.8 psia (2.5 atm), the density of the gas could be varied by a factor of 10. In addition, the apparatus was equipped with a variable speed motor with a range of 0 to 1900 rpm. The variable speed feature permitted the surveying of particle-flow patterns as a function of disk speed.

The survey of the flow induced by the rotating disk has shown a wide range of flow regimes for the dispersed particle flow. These range from surface erosion without appreciable presence of the solid component in the gas, as depicted in Fig. 4-55, to intermediate states where particle clouds of increasing density and height occupy the vessel as shown in Figs. 4-56 and 4-57. The beginning and intermediate states are characterized by the formation of a central heap of particles. This is due to the action of the inward flowing boundary layer. Turbulent eddies near the surface of the heap transport particles into the main flow; they are then centrifuged toward the wall of the vessel where they spiral downward. It was also observed that at higher disk speeds particles flowed across the edge of the disk in an upward direction. The same phenomenon occurs in the present experimental coal feeder apparatus.

Fig. 4-54 Schematic of the Flow Simulator Chamber

Fig. 4-55 Surface of Particle Bed Eroded by Disk-Induced Flow of Freon F-12 at 14.7 psia, 760 rpm, $Re = 3.5 \times 10^5$

Fig. 4-56 Particle Flow in Freon F-12 at 14.7 psia, 1955 rpm, $Re = 8.9 \times 10^5$

Fig. 4-57 Dense Cloud of Particles in Freon F-12 at 36.8 psia,
1310 rpm, $Re = 1.5 \times 10^6$

For a disk diameter d_1 and solid particles of density ρ_s and size δ , the amount of dispersed particle flow is determined by the disk speed at the outer rim v , the gas density ρ_g , and the distance between the disk and the surface of the particle bed h . In this initial survey, h was held constant at $h/d = 0.55$. As expected, the density of dispersed particle flow was proportional to v and ρ_g , or to the Reynolds number of the swirling flow between the disk and the particle bed,

$$Re = \frac{\rho_g v h}{\mu}$$

where μ is the viscosity of the gas. Using this definition of the Reynolds number, an empirical correlation of various observed and photographed regimes of dispersed particle flow with the physical parameters of the flow as attempted. The density of the gas in the chamber was varied by establishing

gaseous atmospheres of air at 14.7 psia (1 atm), Freon F-12 at 14.7 psia, and Freon F-12 at 36.8 psia (2.5 atm). Preliminary results of this correlation are summarized below.

Observed Flow Regimes, $h/d = 0.55$	$Re = \frac{\rho_g v h}{\mu}$
No vertical transport of particles by the swirling flow	$Re < 5 \times 10^4 - 2 \times 10^5$
Light particle flow with erosion of the surface of the particle bed and beginning centrifuging of particles into the wall region (Fig. 4-55)	$Re = 10^5 - 5 \times 10^5$
Strong swirling flow of dispersed particles filling the chamber to $h/2$ (Fig. 4-56)	$Re = 5 \times 10^5 - 1.5 \times 10^6$
Very strong particle flow which fills the entire chamber (Fig. 4-57)	$Re > 10^6 - 1.5 \times 10^6$

A numerical data reduction and plotting routine was devised to correlate the observed flow regimes of dispersed particle flow with a number of nondimensional parameters. Some of these parameters, which were derived from dimensional analysis, are

$$\pi_1 = \frac{\rho v^2}{\rho_e g \delta}$$

$$\pi_2 = \frac{\rho v h}{\mu}$$

$$\pi_3 = \frac{\mu v}{\rho_e g h \delta}$$

$$\pi_4 = \frac{\rho v D}{\mu}$$

where

- ρ = gas density
- ρ_e = net density of solid particles, $\rho_e = \rho_s - \rho$
- ν = rim velocity of disk
- h = distance between disk and particle bed
- g = acceleration of gravity
- δ = particle diameter
- μ = gas viscosity
- D = diameter of flow chamber

The parameters π_2 and π_4 are the Reynolds numbers for the flow chamber and for the disk, respectively.

Other nondimensional parameters can be formed with the eight flow variables affecting the dispersed particle flow and have been included in the data analysis, but the initial results appear to indicate that the parameters listed above yield correlations from which empirical flow models can be derived. For example, one flow regime of light-to-moderate particle flow is characterized by the appearance of longitudinal streaks of particles along the wall of the vessel. The streaks are formed by particles spinning out of the flow because of centrifugal effects. This flow regime may be represented by the power relationship,

$$\pi_3 \pi_2^\alpha = \frac{\mu \nu}{\rho_e g h \delta} \left(\frac{\rho \nu h}{\mu} \right)^\alpha = K$$

where α and K are constants. The plot of the parameters π_3 versus π_2 for the data of the flow regime described above is shown in Fig. 4-58. The power law was derived from this plot by representing the data by a straight-line relationship; the dashed line is a least-squares fit to the data. The remaining flow regimes representing a variety of types and intensity of particle flow are analyzed similarly, and it appears that all flow regimes may be modeled by two or three empirical formulas similar to the one shown above.

Fig. 4-58 Data Correlation for the Flow Regime in Which the Dispersed Particle Flow is Characterized by Longitudinal Wall Streaks Due to Centrifugal Effects

It should be noted that the particle size δ and vessel diameter D were not varied in the series of flow observations. It is therefore not known whether δ and D actually scale as indicated by the parameters π_3 and π_4 , or by the other parameters included in the investigation. Further experiments would be required to determine the relationship of these flow quantities to the type and intensity of dispersed particle flow.

4.2.13 Horizontal Shaft Flow Simulator

A one-half-scale model of the Kinetic Extruder embodying the horizontal-shaft concept and shown schematically in Fig. 4-59 was fabricated and installed at the Fluid Mechanics Laboratory. Flow visualization studies carried out with this apparatus have yielded basic information regarding the motion of particles suspended in the disk-induced boundary layer flow. Data were also obtained pertaining to the density of the particle suspension and to the number of particles settled out in the bottom portion of the flow chamber.

The flow visualization apparatus represents a geometrical scale model and permits the study of the flow patterns that occur as a result of the rotating disk. Furthermore, the apparatus allows correct aerodynamic scaling of the motions of particles within the suspension and of the particle-settling characteristics in the less energetic regions of the flow. This is achieved by adjusting the ratio of the density of the particles to that of the ambient gas to values which occur in the actual operation of the Kinetic Extruder. For this purpose, the flow chamber is filled with a variety of gases at pressures ranging from 1 to 2.5 atm. For example, the gas density of $0.0012/\text{g}/\text{cm}^3$, with air or nitrogen at atmospheric pressure, increases to $0.0126/\text{g}/\text{cm}^3$, when the flow chamber atmosphere consists of Freon F-12 at 2.5 atm. The particles used to simulate the flow of coal dust are glass microballoons ranging in size from 75 to 200 μm with a bulk density of $0.17/\text{g}/\text{cm}^3$. Operating speeds of the rotor of 3600 to 3700 rpm are the same as for the full-scale Kinetic Extruder.

Fig. 4-59 Horizontal Axis Flow Simulator

The apparatus shown in Fig. 4-59 was used to conduct a series of flow visualization studies. The objectives were as follows:

- o To observe the overall flow pattern of the turbulent boundary layer between the Extruder disk and the housing walls
- o To observe the density of particle suspension maintained by the flow
- o To study the effectiveness of guide vanes installed between the upper housing and the lower section for the purpose of slowing down the flow within the tapered settling region

These characteristics were studied as a function of gas density. The disk speed was held constant at about 3600 rpm. For testing at the lower range of gas density, the apparatus was filled with dry nitrogen at atmospheric pressure. In this regime, a certain amount of the glass microballoons remained suspended in the flow, but settled out in the tapered bottom section of the apparatus at the same rate at which they were fed into the rotor chamber from a hopper above. When the feeding of powder stopped, complete settling of the particles could be observed while the disk remained in motion. Subsequently, the gas density was increased by a factor of 10.8 by operating the disk in an atmosphere of Freon F-12 gas at 2.5 times atmospheric pressure. Under these conditions, the flow suspended a substantially higher number of particles which did not completely settle out when the feeding of the powder was halted. The density of the residual particle flow (i.e., without feeding new powder into the simulator) was also observed to increase with the height of the particle bed in the lower settling region of the flow simulator.

The observations described above were documented with 16-mm color motion picture film.

An attempt was made to modify the flow of the particle suspension into the settling region by installing a strip of honeycomb between the disk and the tapered section. This modification had the effect of increasing the intensity of the flow in the settling region, and thus prevented the settling of

suspended particles. This observation led to the conclusion that the velocity and turbulence levels of the flow in the settling region might be regulated by proper installation of variable guide vanes or baffles. Such an arrangement appears desirable in order to control the density of particle flow during startup and shutdown operations.

A modification of the simulator was then made, and a series of experiments using the modified apparatus was completed demonstrating the feasibility of continuously controlling the flow intensity within the settling chamber of the apparatus. In this way, control can be gained over the settling of particles independent of the rotor speed.

The modification consists of a butterfly valve mounted immediately below the rotor disk that effectively divides the apparatus into a rotor chamber and a settling chamber. As shown in Fig. 4-60, the valve is mounted between the lower edge of the rotor and a section of honeycomb flow straighteners, and can be turned on its shaft from outside the vessel. In the open position, the valve is aligned with the disk-driven flow, allowing the flow in the settling chamber to be energized by the effect of the flow straighteners, while the turbulent fluctuations prevent the particles from settling out. With the valve closed, flow to the settling chamber is cut off, so that the amount of particles fed into the apparatus remains suspended in the rotor chamber. Partial opening of the valve induces some circulating flow within the settling chamber, but particles settle out at a rate that is initially proportional to the valve opening. Upon further opening of the valve, the upper flow induces gradually intensifying currents within the settling chamber, thereby progressively reducing the settling rate of particles.

These tests indicate that it is possible to isolate the rotor-induced turbulence by proper design of the rotor chamber bottom opening and to collect the particles in the settling chamber by gravity. From the settling chamber, one can withdraw the particles in a dense phase flow.

Fig. 4-60 Installation of Recirculation Control Valve in Flow Simulator

4.3 THROTTLEABLE ROTOR TESTS

Concepts for throttling the Kinetic Extruder by means of gas injection into the coal passages (fluidic control) were developed and verified in tests with a special rotor. This throttleable rotor was fabricated with a means for controlled gas injection either into the entrance of the control orifice or into the body of the sprues. Figure 4-61 gives a schematic of the unit, and Fig. 4-62 shows a photograph of the installation in the vertical shaft configuration of the test loop.

Details of the mode of gas injection for throttling is illustrated by Fig. 4-63. In the nonthrottling configuration, the gas-pressure difference across the control orifice hopper is zero, and the coal throughput is only a function of rotor speed. Specifically,

$$\dot{m} = C_D d^{5/2} \left(\frac{r\omega^2}{g} \right)^{1/2} \quad (4.2)$$

Note that in $\dot{m} \propto \omega$, so that some throughput variation can be accomplished by speed changes. However, large turndown ratios are impractical at high pressure.

In the throttling configuration, the control orifice gas pressure, P_c , is maintained at a value slightly different from the delivery pressure P_D . This induces a radial pressure gradient in the coal in the control orifice hopper. The pressure gradient at the outlet,

$$\left(\frac{-dp}{dr} \right)_o$$

adds to the body force and changes the throughput according to the equation

Fig. 4-61 Schematic of Rotor Assembly

Fig. 4-62 Throttleable Rotor Installation

Fig. 4-63 Throttleable Versus Nonthrottleable Configuration

$$\dot{m} = C_D d^{5/2} \left[\frac{r\omega^2}{g} + \frac{1}{\rho_s} \left(\frac{-dP}{dr} \right)_o \right]^{1/2} \quad (4.3)$$

If $(P_c - P_D) > 0$, the pressure gradient adds to the body force, and the throughput is increased; if $(P_c - P_D) < 0$, the throughput is less than the nominal unthrottled value. Calculations treating the control orifice as a high area ratio channel were performed. A typical set of calculations is shown in Fig. 4-64, which shows the outlet pressure gradient of the control orifice as a function of $P_c - P_D$. Based on these studies, it became apparent that a large outlet pressure gradient

$$\left(\frac{dP}{dr} \right)_o$$

could be generated with a relatively modest pressure difference $P_c - P_D$. This is due to the highly convergent shape of the control orifice.

In general, the throttling scheme worked exactly as predicted. The rotor was very responsive and changes in control gas pressure could be seen to be instantly reflected in the amplitude of the signals on the sprue flow monitoring instrumentation. Test data accumulated for the throttleable rotor are given in Fig. 4-65. This figure shows test points of the observed throttling ratio plotted against the control pressure difference $(P_c - P_D)$ for several conditions. Here

$$\text{throttling ratio} \equiv \frac{(\text{actual throughput})}{(\text{throughput for nonthrottleable orifices of the same diameter})}$$

The results show that about a 3:1 variation in throttling ratio was achieved with a control pressure change of only 30 psi. The data were also in very good agreement with theoretical predictions based on Eq. (4.3). Finally, it should be noted that the data confirm that the throttling ratio is a function only of the control pressure difference and is independent of the actual value of the delivery pressure.

Fig. 4-64 Theoretical Prediction of Control Orifice Outlet Pressure Gradients

Fig. 4-65 Throttleable Rotor Test Data

In addition to the tests with control orifice gas injection, the rotor was also operated successfully in the midsprue gas injection mode. Testing verified that the sprue gas pressure at the injection point was easily controllable by means of the injection gas flow. This finding demonstrates the feasibility of using gas injection to "fine tune" the sprue pressure distribution to adjust for short-term variations in feed-stock permeability. Such fine tuning could reduce power consumption.

4.4 KINETIC EXTRUDER MODEL NO. 3

4.4.1 Vertical Configuration Tests

The Kinetic Extruder Model No. 3 is a nonthrottleable rotor which was fabricated incorporating all of the information gained in the component evaluation tests. The Model No. 3 rotor diameter remained at 28 in., but the overall sprue length was made slightly longer than in the sprue evaluation fixture by reducing the spinup zone radius to 4 in. A similar funnel/sprue body/control orifice design was used in the Model No. 3 design as had been successfully explored with the sprue evaluation rotor. The Model No. 3 funnels were made from special castings having a 16-deg constant wall angle with respect to the acceleration vector, as discussed in subsection 4.2.2. Figure 4-66 shows a sprue assembly; Fig. 4-67, the rotor for the Model No. 3 feeder; and Fig. 4-68, the installation of Model No. 3 in the vertical shaft test configuration in which most of the tests were made.

Testing of the Kinetic Extruder No. 3 rotor went extremely well. A goal of 400 psia had been established and was met. A maximum delivery pressure of 415 psia was reached at a throughput of 1.04 ton/h. The runs in excess of 400 psia were of about 7 minutes in duration, and the feeder operated smoothly. The 400-psi level represents the maximum capability of the machine because of input power limitation.

Fig. 4-66 Typical Sprue Configuration

Fig. 4-67 Rotor for Kinetic Extruder Model No. 3

Fig. 4-68 Kinetic Extruder Model No. 3

The high-pressure test runs are discussed in detail in Appendix C from the standpoint of comparing the data with computer predictions. It is concluded that the predictions are in close agreement with the test data and the computer model can be considered sufficiently validated to be used for design of Kinetic Extruders to any particular requirements.

As discussed in subsection 4.2.3 and Appendix B, specific power requirements for the Kinetic Extruder are highly scale-dependent. A small 1 ton/h size unit cannot be pushed to extremely high pressures without severe power penalties. Figure 4-69 shows actual power consumption during tests conducted with the Kinetic Extruder Model No. 3. Generally, one tends to operate conservatively, i.e., with higher rotor speed and power than are necessary. Little performance margin was left during the 400-psi tests. A minimum-power curve passing through the 415-psi point using a $HP \propto p^{2.5}$ scaling law therefore constitutes a reasonable minimum operating power curve that can be used for extrapolation. In small, drag-dominated machines, the $p^{2.5}$ scaling law is appropriate. The aerodynamic drag-related power is proportional to $p\omega^3$, and the speed requirement for the body force to exceed the pressure gradient requires that $\omega \propto p^{1/2}$. Thus,

$$HP \propto \alpha p^{2.5}$$

4.4.2 Horizontal Shaft Configuration Tests

After completion of the Model No. 3 vertical configuration tests, the facility was converted to horizontal axis testing of the same Kinetic Extruder No. 3 rotor in a new housing. Figure 4-70 shows the horizontal shaft housing. The changeover required rework in the instrumentation, lubrication, and coal transfer systems and in the relocation of the drive system. However, the same basic rolling element bearing, full ΔP face seal design was used as had been used previously. It was planned to convert to the favored journal bearing concept. However, these designs were not complete, and it was judged desirable to uncover any possible "surprises" in horizontal operation at an early date. Figures 4-71, 4-72, and 4-73 are photographs of the horizontal

Fig. 4-69 Rotor Input Power as a Function of Delivery Pressure for Kinetic Extruder Model No. 3

Fig. 4-70 Horizontal Shaft Housing

Fig. 4-71 View of Coal Inlet Side of Kinetic Extruder

Fig. 4-72 View of Drive Side of Kinetic Extruder

Fig. 4-73 Kinetic Extruder Rotor - Model No. 3

axis feeder configuration. Figure 4-71 shows the Kinetic Extruder viewed from the coal inlet side, and Fig. 4-72 gives the view from the drive shaft side of the feeder. Figure 4-73 is a photograph taken with the housing lid removed, exposing the rotor.

A limited number of coal pumping tests were carried out with this horizontal axis unit at delivery pressures up to 300 psig. No differences in performance were observed with the rotor operated in the horizontal axis position when compared to previous operation in the vertical axis configuration. The added horizontal length in the coal feedline does not cause any problems in starting or maintaining coal feed to the rotor. Torque data indicate that rotor power consumption is unchanged. Total power consumption was slightly lower since gearbox losses have been eliminated.

A side benefit of the new machine is that it is much quieter since it uses a belt drive instead of a gearbox.

4.5 KINETIC EXTRUDER MODEL NO. 4

Because of the excellent test results obtained with the Model No. 3 feeder, an effort was started to design and fabricate the Model No. 4 feeder. This machine, compatible with the horizontal shaft housing, incorporates all features of a commercial feeder:

- o The straddle-mounted shaft is supported by water-lubricated journal bearings
- o Low-pressure differential seals are used with automatic pressure control
- o Provision is made for fluidic turndown control

The Model No. 4 feeder was fabricated, assembled, and installed in the test-loop late in Phase III. Figure 4-74 shows a sectional view of the rotor mounted in the housing. As shown, the feeder consists of four major subassemblies: the housing, the rotor, the drive shaft, bearing, and seal assembly, and the coal feed tube, bearing, and seal assembly.

A corresponding view of the drive shaft, bearing, and seal subassembly is shown in Fig. 4-75 and of the coal feed tube, bearing, and seal assembly in Fig. 4-76. The rotor assembly is shown in Fig. 4-77. Photographs of the subassemblies of the Model No. 4 hardware are included in Figs. 4-78 through 4-82.

The complete manufacturing drawing package for the Model No. 4 feeder is included in Appendix D.

The Model No. 4 feeder is ready for run-in tests, calibration runs, and engineering evaluation.

4.6 COAL PROPERTIES

4.6.1 Permeability and Density

The bulk feedstock properties needed for the design of the Kinetic Extruder are the bulk density, porosity, and permeability. Porosity is directly related to the bulk density by

$$\epsilon = (\rho_T - \rho_b) / \rho_T$$

where

ρ_T = true material density

ρ_b = bulk density

Therefore, only the bulk density and permeability at the state of compaction of the sprue material need to be determined.

A laboratory apparatus was assembled to measure permeability of a compacted column of coal. The schematic of the setup is shown in Fig. 4-83. To measure permeability, the material sample is placed in the sample tube and compacted in a laboratory centrifuge. The sample tube is then placed in the apparatus

Fig. 4-74 Kinetic Extruder - Model No. 4

Fig. 4-75 Drive Shaft Assembly

Fig. 4-76 Feed Tube Assembly

Fig. 4-77 Rotor Assembly

Fig. 4-78 Model No. 4 Rotor Body (Nozzle Blocks Not Installed)

Fig. 4-79 Control Nozzle, Sprue and Funnel Assembly

Fig. 4-80 Drive Shaft Bearing and Seal Assembly

Fig. 4-81 Coal Feed Pipe Side Bearing and Seal Assembly

Fig. 4-82 Installation of Rotor in Housing

shown in Fig. 4-83, and the flowrate of gas permeating through the column is measured at a given pressure drop. The pressure drops are kept small so that the permeability may be computed from an incompressible form of the Darcy equation. Specifically,

$$k = \frac{Q_s \mu L_s}{A_s \Delta P_s}$$

where

- Q_s = volume flow rate through sample
- L_s = material column length
- ΔP_s = pressure drop across column
- A_s = column cross-sectional area.

The sample tubes are 2.2 cm in diameter, and a typical compacted column length was 5 to 6 cm.

Figure 4-84 shows typical permeability data and bulk density data obtained by the above procedure as a function of compacting G-force in the centrifuge. The test coals are samples taken in conjunction with a series of Kinetic Extruder experimental runs. As discussed in Appendix C, at similar compaction stress levels the permeability of the flowing material in the Kinetic Extruder sprues appears to be only about 30 percent higher than that measured in the static laboratory tests. Thus, with only a small correction factor, the laboratory test data can be used as a basis for sizing the sprues.

To determine the to-be-expected range of variation of packed coal permeability under normal industrial operating practice, arrangements were made with Northern States Power Company of Minneapolis to supply 1-gal samples of as-fired pulverized coal at regular intervals from its Highbridge Power Plant. Ten samples, taken about twice monthly, have so far been received.

Fig. 4-83 Schematic of Apparatus to Measure Permeability of a Compacted Column of Coal

Fig. 4-84 Typical Permeability and Bulk Density Data as a Function of Compacting G-Force in the Centrifuge

Permeability and density test results for the samples are summarized in Table 4-3 and in Figs. 4-85 and 4-86. The key property, permeability, is similar in magnitude to the test coals we have been using and exhibits about a threefold overall variation among the 10 samples (only twofold if sample 1 is omitted). The spread in bulk density is approximately 10 percent. It may be noted that the three samples from the same pulverizer (No. 122) show almost no variation in properties.

We would conclude that the observed spread in permeability can be accommodated in the design of a Kinetic Extruder feeder unit. Based on the observed data, normal industrial controls are sufficient to ensure compatibility of the feed with the requirements of the feeder.

4.6.2 Flowability

Theoretical Background. The control orifice of the Kinetic Extruder is a miniature hopper or bin operating at high g-force. A considerable body of theory and engineering data exists relating to bin flow problems (Refs. 1 and 2), and this information may be applied to the Kinetic Extruder.

Basically, there are two types of flow disturbances that can occur at bin outlets:

- o Particle wedging
- o Cohesive doming

Particle wedging occurs when particles wedge into the outlet forming a bridge which will yield only if the strength of the individual particles is exceeded. Wedging is avoided if the outlet diameter is larger than a multiple of the particle size; most sources give three to five times as recommended multiples. This means that large feeders should be less subject to particle wedging than smaller machines.

Table 4-3. NORTHERN STATES POWER COMPANY SAMPLE PROPERTIES

Sample No.	Date Received	Description on Sample Container	Permeability ^(a) (10 ⁻¹² ft ²)	Bulk Density ^(a) (g/cm ³)	True Density (g/cm ³)	Porosity ^(a)
1	1/3/79	None	8.7	0.710	1.425	0.502
2	1/3/79	None	2.9	0.645	1.425	0.547
3	1/3/79	None	4.2	0.686	1.406	0.512
4	1/26/79	Highbridge, Can No. 4, 91 Pulverizer, 12/14/78	4.8	0.672	1.306	0.485
5	1/26/79	Highbridge, Sample No. 5, No. 122 Mill, 12/28/78	3.3	0.665	1.328	0.499
6	3/79	Highbridge Plant, 111 Pulverizer, 1/25/78	5.0	0.654		
7	3/79	Highbridge, Pulverizer No. 122	3.2	0.653		
8	4/79	Highbridge, Pulverizer No. 122, 3/5/79	3.0	0.648		
9	4/79	No. 12 Boiler, Pulverizer No. 123, Burner No. 6 Sampled 3/27/79	2.9	0.638		
10	5/79	Can No. 9, Unit 11, Pulverizer No. 112, Burner No. 6, 260 A, Load 80 MW, Highbridge Plant, St. Paul	5.2	0.669		

(a) The test approximates sprue compaction

NORTHERN STATES POWER CO. SAMPLES

Fig. 4-85 Bulk Density Test Data

Fig. 4-86 Permeability Test Data

Cohesive doming occurs when interparticle cohesive forces of the bulk material generate sufficient strength to form stable domes. Generally, the more fines a material contains, the more cohesive it becomes.

A theory developed by Jenike (Refs. 1 and 2) to analyze the cohesive-doming phenomenon includes several key ideas which we will briefly touch on here. First, the flow/no flow criterion is basically derived by equating the strength of the bulk material to the stress developed in the abutments of the dome of material over the bin outlet. This simple principle is somewhat complicated by the fact that the material strength is not constant, but instead increases with consolidating stress (i.e., compaction). Furthermore, the stress in the dome is also related to the consolidating stress, both stresses being proportional to the span of the dome.

Figure 4-87 illustrates the situation graphically. Curve A gives the unconfined yield strength of the bulk material as a function of consolidating stress. (Curve A is an empirically determined material property which Jenike calls the flow function.) Curve B represents the dome abutment stress which plots as a straight line. The slope of Curve B is a function of the material friction angles and the hopper angle, and is calculable theoretically. In the usual case, Curves A and B will intersect defining a critical stress (or strength) value, S_c . To the right of the intersection, the stress in a dome exceeds the material strength and, consequently, a dome cannot form and the material flows. To the left of the intersection, strength exceeds stress and we have no flow.

The theoretical relationship between the span, d , of the critical dome and S_c is given in Refs. 1 and 2:

$$d = \frac{S_c H(\theta)}{\rho_c g} \quad (4.4)$$

Fig. 4-87 Illustration of Flow/No Flow Criteria for Bin Outlets

where

ρ_c = bulk material density

g = gravity acceleration constant

$H(\theta)$ = a weak function of hopper angle θ , $H(\theta) = 2.0$ to 2.5 for conical hoppers

It is important to note that although this theory has been developed for gravity flow, Curves A and B, as well as the critical value S_c , are independent of the G-level.

Therefore, at high G-forces the governing relationship becomes

$$d = \frac{S_c H(\theta)}{\rho_c g G} \quad (4.5)$$

where the G-force is:

$$G \equiv \frac{r \omega^2}{g}$$

For a given outlet diameter, d , there exists a critical value of G , G_{crit} such that we have flow if $G > G_{crit}$ and no flow when $G < G_{crit}$.

Or, viewing it in a different way, for a given G-force there exists a critical diameter, d_{crit} , for which flow will occur only if $d > d_{crit}$. Note that $d_{crit} \propto \frac{1}{G}$. This means, for example, that the same material will flow through a hole 1000 x smaller at 1000 g than it will at 1 g. This explains why coal will flow through the narrow Kinetic Extruder passages without bridging.

The critical G-force can be calculated from Eq. (4.5) and is given by:

$$G_{\text{crit}} = \frac{1}{d} \frac{\rho_c g}{S_c H(\theta)} \quad (4.6)$$

Equation (4.6) is the relationship that was to be verified through experiments.

Centrifuge Experiments. These experiments are an elaboration of previously described tests in a laboratory centrifuge (see Appendix C, Ref. 3). Figure 4-88 illustrates the test devices. In a typical experiment, coal is placed in the upper portion of the test tube and the tube is centrifuged at a series of increasing G-force levels. After each G-force step, the tubes are examined and a note made of whether or not coal flow into the lower part of the tube has occurred. In this way, the critical G-force level can be established.

Experiments with 14 different sprue outlet configurations have been performed to date. Five outlet diameters - 0.020 in., 0.025 in., 0.040 in., 0.065 in., and 0.090 in. - were used with two different conical hopper angles, 8.5 deg and 20 deg, and a set of square cross-section models has been tested.

Figure 4-89 shows a set of results for the 20-deg cone angle. The test coal is the "fine coal" (nominal 70% passing No. 200 sieve) used in the Kinetic Extruder Model No. 2 runs and was, in fact, taken from the test loop vessel.

Three test outcomes are represented by the data shown in Fig. 4-89:

- (1) No coal passing the sprue outlet - these data are plotted as open circles
- (2) All possible coal flowing with the lower chamber filled up to the sprue outlet - plotted as solid circles
- (3) A small amount of coal flowing and then the outlet becoming blocked again - half-shaded data points.

Fig. 4-89 Centrifuge Test Results. Fine coal, 20-deg conical sprue, no preconsolidation

Data presented in Fig. 4-89 define a critical G-force limit line between the free-flowing and non-freeflowing regions. For the three largest outlet sizes, this limit line is reasonable consistent with Eq. (4.6); i.e., $\log G_{crit}$ versus $\log (1/d)$ lines has roughly a 1:1 slope. However, the 0.020- and 0.025-in. holes remain filled at much higher G-forces than expected by extrapolating the cohesive-doming limit line. The interpretation is that somewhere between the 0.040- and 0.025-in. diameters particle-wedging comes into play which is, as expected, much more effective in stopping the flow than cohesive doming.

It is important to note that, according to Fig. 4-89, the critical G-forces required to overcome cohesive doming are low in comparison to typical Kinetic Extruder sprue outlet G-forces which typically are about 5000 g's.

The next series of results is shown in Fig. 4-90. These experiments were run using the same sprues and coal as used previously, but the coal was compacted at 500 g. That is, after the coal was introduced into the test tubes, the sprue outlets were blocked from below and the tubes centrifuged at 500 g for approximately 2 min to deaerate and compact the coal. This procedure had the effect of making the results more reproducible and eliminating most of the "partial flow" data points. It also increased the G_{crit} limit line by about a factor of 2 over the data points shown in Fig. 4-89.

Figure 4-91 shows results from tests with 30-percent extra water (by weight) added to the coal. With this amount of water, this fine coal appeared reasonably dry after thorough mixing, but centrifuging caused some of the water to separate. According to Fig. 4-91, the higher moisture content increases the G_{crit} levels by a factor of 4 over the drier coal. However, typical Kinetic Extruder G-levels remain higher still by a wide margin.

Sets of tests were also conducted using 8.5-deg hopper angle sprues and rectangular 20-deg hopper shapes. There was no significant difference from the conical 20-deg hopper results.

Fig. 4-90 Centrifuge Test Results. Fine coal, 20-deg conical sprue, preconsolidated at 500 g

Fig. 4-91 Centrifuge Test Results. Fine coal, 20-deg conical sprue, preconsolidated at 500 g, 30% moisture

Test results illustrating the effect of the precompacting g-level on the flow-inducing G-force are presented in Fig. 4-92. As shown, the data for the two test coals shown plots are approximately linear. Any data to the right of the 45-deg line in Fig. 4-92 would correspond to free-flowing conditions at a constant G-level.

Finally, the test was standardized at a 0.093-in. outlet size, a 20-deg hopper angle, and 1000-g precompaction. Thus, a flow-inducing G-force of less than 1000 g would guarantee free flow. Likewise, less than 1000 g would imply that the critical diameter for flow at 1000 g is smaller than the 0.093-in. test orifice size, and if the coal is still nonflowing at 1000 g, this indicates that the critical diameter is greater than 0.093 in. The test should ideally be conducted at a G-level characteristic of the Kinetic Extruder (e.g., 5000 g); however, these levels are well beyond the reach of the available laboratory centrifuge. It should be noted that this test is somewhat conservative in another way since it presumes the existence of a dome and measures the g-force required to break it up.

Using the standardized test conditions, investigations were made of the effects of the following factors on flowability:

- (1) Variation between coal types
- (2) Added moisture
- (3) High temperature

Table 4-4 gives a listing of samples tested and their as-received moisture contents. Figure 4-93 shows a summary of the flowability test results as a function of moisture content. The open circles correspond to samples tested with as-received moisture; extra water has been added for the solid circle data points.

The results show a general decrease in flowability with moisture content, but there is also a significant variation between the samples of various types. Consequently, the as-received data form a rather broad band. Only one coal, Western Kentucky No. 9, failed to flow in the 0.093-in. orifice bench test at

Fig. 4-92 Flowability Test Results (0.093-in. Test Orifices)

Table 4-4. COAL SAMPLE MOISTURE ANALYSES

Sample No.	Coal Type and Source	Moisture (%)
N-1	Northern States Power Co. Samples	11.8
N-2	Northern States Power Co. Samples	17.8
N-3	Northern States Power Co. Samples	12.3
N-4	Northern States Power Co. Samples	18.4
N-5	Northern States Power Co. Samples	13.4
N-6	Northern States Power Co. Samples	18.3
N-7	Northern States Power Co. Samples	14.2
N-8	Northern States Power Co. Samples	14.9
N-9	Northern States Power Co. Samples	14.3
N-10	Northern States Power Co. Samples	15.4
S-1	Shell Samples, Auguste Victoria	1.5
S-2	Shell Samples, Auguste Victoria	2.6
S-3	Shell Samples, Griesborn Duhamel	4.0
S-4	Shell Samples, Ground Fluid Coke	0.8
S-5	Shell Samples, Ground Fluid Coke	0.6
K-1	Kennedy Van Saun Samples, Bell Ayre South	11.2
K-2	Kennedy Van Saun Samples, Lower Freeport	10.8
K-3	Kennedy Van Saun Samples, Western Kentucky 9	2.9
K-4	Kennedy Van Saun Samples, Ohio 9	2.1
L-2	Lockheed Test Coal, Test Loop (1/9/79)	3.8
L-3	Lockheed Test Coal, Test Loop (1/19/79)	3.4
L-7	Lockheed Test Coal, From Bag (5/29/79)	3.5
L-8	Lockheed Test Coal, Test Loop (5/29/79)	2.9
L-9	Lockheed Test Coal, From Bag (5/29/79)	3.3
L-10	Lockheed Test Coal, From Bag (2/14/79)	3.5

Fig. 4-93 Flowability Through 0.093-in. Orifice Test Results as a Function of Coal Sample Moisture Content

the as-received moisture content. Repeating the test using 0.125-in. orifices yielded a flow-inducing G-force of 170 g. This would indicate a bench test critical diameter of between 0.093 in. and 0.125 in. for this coal. However, the Western Kentucky coal was successfully pumped in tests with the Kinetic Extruder with only 0.084-in. control orifices. This finding confirms the conservative nature of the bench test.

The results shown in Fig. 4-93 also indicate that added water has a somewhat more detrimental effect than the original moisture in the coal. Increasing the moisture to the range of 25- to 40-percent moisture yields poor flowability through the 0.093-in. orifices at 1000 g. Finally, above 45 percent the water content is enough to form a slurry, and the mixture again flows very freely.

Feed Stock Temperature Effects. In commercial applications, the Kinetic Extruder may be supplied by a feed stock at an elevated temperature ($\sim 200^{\circ}\text{F}$) due to upstream drying operations. To establish that the flowability of the coal is not adversely affected by higher temperature, flowability tests were conducted using heated coal samples. The test procedure was similar to that used previously except that the coal samples were heated in air to temperatures up to 450°F before flow was induced in the centrifuge. The 1000-g precompaction step in the test procedure was done at room temperature.

Figure 4-94 shows the test results in terms of the G-force required to induce flow through the test orifice as a function of temperature. The solid data points signify that flow occurred at the indicated G-level, open circles correspond to lower G-level attempts where flow did not occur. The coal is free-flowing over the entire temperature range since the flow-inducing G-force is everywhere less than the 1000-g precompaction level.

Fig. 4-94 Effect of Temperature on Coal Flowability (0.093-in. Test Orifices, Lockheed Test Coal)

A slight decrease in flowability is evident in going from room temperature to 150°F. However, no further change occurs until the 450°F level is reached. At this temperature, the coal was smoking considerably and higher-temperature tests were not run.

These tests indicate that no problems should be encountered at temperatures below 350°F, which is the operating limit imposed on the coal feeder.

4.7 TEST EXPERIENCE WITH VARIOUS FEEDSTOCKS

A number of different coals and coal grinds have tested using one or another of the Kinetic Extruder units. A summary of the experience with the various feedstocks is given in this subsection.

o Utah Test Coal (70% - 200)

This test coal is the Utah bituminous coal which we obtain, ground to the 70-percent passing 200-mesh specification, in 50-lb bags from our local supplier. Taken directly from the bag, the coal generally had a moisture content of about 3 to 4 percent and tested at a permeability of $k = 5$ to 7×10^{-12} (ft²). After recycling several times through the test loop, the permeability generally fell to 3 to 4×10^{-12} (ft²).

Most of the testing was carried out using the coal, and no problems were encountered.

o Utah Test Coal (20% x 0)

To demonstrate the capacity of the Kinetic Extruder to handle fluid bed type of coal grinds, a supply of the Utah Coal ground to a nominal 20 x 0 specification was obtained and tested in the Kinetic Extruder. This coal had the following sieve analysis:

<u>Sieve No.</u>	<u>% Passing</u>
50	88
100	60
200	32
325	19
400	15

The standard permeability test yielded $k = 11 \times 10^{-12} \text{ ft}^2$ for a test-loop sample of the coal. This is about 3x as permeable as the 70 percent - 200 mesh grind normally used.

The increase in coal permeability may be compensated for by a corresponding decrease in sprue cross-sectional area. Accordingly, the smallest available set of sprues (0.189-in. outlets) were installed for tests. These have about one-half the area of those normally used for the fine coal. The tests confirmed that feeding the 20 x 0 grind through the Kinetic Extruder is quite feasible. No flow problems were associated with the larger particles, and operations were quite smooth.

Increasing gas back leakage, with the use of the available sprues, limited operations to a maximum delivery pressure of 175 psia. However, this is in no way a fundamental limitation. To achieve higher delivery pressure, either the coal velocity could be raised by further reducing the cross-sectional area of the sprue, or the capacity of the rotor eye suction system could be increased, or both.

o Test Coals Received from Kennedy Van Saun Corporation (70 percent - 200)

Four test coals, ground to a nominal 70-percent - 200 mesh specification, were received from Kennedy Van Saun Corporation in large enough quantities to use in the Kinetic Extruder test loop. The shipment included three bituminous coals Lower Freeport, Western Kentucky 9, and Ohio 9 - and one subbituminous, Bell Ayre South. Table 4-5 lists some of the standard data for the coal shipments.

Table 4-5. TEST COALS RECEIVED FROM KENNEDY VAN SAUN CORPORATION

Seams	Lower Freeport	Western Kentucky 9	Ohio 9	Bell Ayre South
Class Geographic Supplier	Bituminous Appalachian Pennsylvania P&L Co.	Bituminous Mid-Continent Pittsburgh Midway	Bituminous Appalachian Conoco Coal	Subbituminous Western AMAX

AS RECEIVED AT KENNEDY VAN SAUN CORPORATION -
PROXIMATE ANALYSIS

Moisture (%)	0.90	2.33	2.26	24.48
VM (%)	27.43	36.80	34.31	33.85
Ash (%)	10.17	10.20	23.49	5.78
FC (diff. (%))	61.50	50.67	39.94	35.89
S (Total) %	0.86	3.66	4.36	0.48
Btu	13,964	12,746	10,855	9,086
Hardgrove	95 - 100	55 - 58	55	40 - 50
Nominal Grind	70% - 200	70% - 200	70% - 200	70% - 200
Permeability (10^{-12} ft ²) (Lockheed Test)	1.4	5.9	5.4	6.0
Moisture (%) (as received at Lockheed)	10.8	2.9	2.1	11.2
Flowability Test g-level for flow (Lockheed Test with 0.093 in. orifices)	330	>1000	230	170

The Lockheed test permeability of three of the four coals was found to be similar to the Utah coals previously tested (i.e., 5 to 6 x 10^{-12} ft²). However, the Lower Freeport Bituminous coal was found to have a much lower permeability than the other coals (1.4×10^{-12} ft²).

Visually, the Freeport coal appeared much finer, and a sieve analysis showed 84 percent passing 200-mesh.

Bench flowability tests of the coals were also carried out in our laboratory centrifuge. As indicated by the results given in Table 4-5, using the present 0.093-in. test orificies, the Kentucky 9 coal samples were nonflowing in the laboratory centrifuge tests.

Before testing the Kennedy Van Saun coals in the Kinetic Extruder, a goal of 1-h running time and a minimum delivery pressure of 200 psig with each of the coals was established. The Model No. 3 Kinetic Extruder was used for the experiments.

Testing of the Ohio 9 and Bell Ayre South coals was completed without incident. Both of these coals behaved very similarly to the Utah coal used in all previous tests. The 1-h, 200-psig goals were met or exceeded using the normally used set of sprues.

Testing with Western Kentucky 9 Bituminous coal was also successful but more cumbersome. Frequent sprue plugging occurred and attrition of coal in the test loop was more rapid. Nevertheless, we were able to test for 1 h at delivery pressures up to 300 psi, meeting the performance goal.

The performance goals could not be met using the Freeport coal because of excessively low permeability of this out-of-specification coal. The maximum delivery pressure that could be reached was 50 psig. To reach high pressures, a sprue cross-sectional area approximately twice as large as that of our largest available set would have been required.

These tests indicate that coals ground to the proper specifications will exhibit only small variations in permeability and, as indicated in subsection 4.4, normally used industrial controls should ensure that day-to-day variations in feedstock will cause no operating problems in commercial plants.

4.8 OPTIMIZATION OF THE KINETIC EXTRUDER

The design of an optimum Kinetic Extruder rotor for a given delivery pressure, feed rate, and coal feed grind is a fairly complex task. Two computer codes are used to carry out the design process. The first code is used to select promising sprue shapes and to size the sprues in terms of total outlet area. This code solves the gas-percolation problem and was first described in our Phase II report (Ref. 3). The theoretical basis for the analysis, as well as favorable comparisons between code predictions and Model No. 3 test data, are the subject of a paper submitted to the AIChE Journal. This paper is included in the present report as Appendix C.

The second code uses the selected sprue shape as input and completes the geometric rotor design. The output data locate the sprue on the rotor and specify the number of sprues, the sprue outlet area and orifice area, the power requirement, and the sprue wall slope angles. The use of the codes for designing an optimum Kinetic Extruder is as follows.

Selection of the Sprue Design and of the Specific Coal Flowrate. The generic sprue design is specified by the sprue length and the area ratio along the sprue. For the double-tapered sprue* shown in Fig. 4-95, which incorporates a transition from a rectangular to circular cross section, one needs to specify the area ratio $A_{23} = a_2/a_3$, the overall sprue area ratio $AR = a_1/a_3$, the location of the slope break point $BP = L_1/L$, and the sprue length L . In addition, the properties of the coal and the desired delivery pressure must be known. The computer output data will show the pressure, pressure gradient, and the gas leakage through the sprue as a function of the coal specific flowrate. Of particular interest is the leakage Q and the pressure gradient at the sprue outlet

$$\left(\frac{dP}{dz} \right)_3$$

*The advantage of using double-tapered sprues has been discussed in subsection 3.2.8 of Ref. 3.

Fig. 4-95 Generic Sprue Configuration

Figure 4-96 shows the result of such a calculation for a double-tapered sprue specified by the design parameters:

$$A_{23} = 3.49$$

$$AR = 80$$

$$BP = 0.4$$

for sprue lengths of $L = 14, 18, 24,$ and 30 in.

For a given sprue length, there exists a specific coal flowrate at which no leakage out of the sprue will occur ($Q = 0$). This flowrate is selected as the design point for the particular sprue.

The selection of proper sprue shape at this time is done by trial and error. Bad choices can lead to undesirably high pressure gradients, high coal flowrates, excessive power consumption, etc. As experience is gained, better selection of candidate shapes is made.

Rotor Design and Selection of Optimum Rotor. To design a rotor based on the sprue designs described above, additional design principles must be established. Referring to Fig. 4-97, one must specify the distance R_1 , which is the distance from the axis of rotation one wishes to locate the sprue entrance. Similarly, one must establish a value for the fraction of available area at the radius R_1 , which can be used as the sprue entrance area, and the ratio of the entrance width to height. To keep the wall slope angles α_1 and α_2 close to each other, a ratio of approximately 0.5 is desirable. As shown in Fig. 4-97, the wall slope angle is the angle between the wall and the acceleration vector. Once these parameters are specified, the computer calculates:

- (1) The total sprue exit area required to yield the desired coal delivery rate
- (2) The total sprue inlet area required based on the selected area ratio AR

Fig. 4-96 Relationship Between Gas Flow Parameters and Coal Flowrate

Fig. 4-97 Generic Sprue Configuration Used for the Optimization Study

- (3) The sprue entrance height based on the selected ratio of height to width and the area utilization factor
- (4) The sprue inlet and outlet area
- (5) The number of sprues required
- (6) The entrance wall slope angles

At this point, the essential geometric wheel dimensions have been determined. Next, the computer calculates the rotational speed required to ensure that the body force exceeds by a specified safety factor the pressure gradient throughout the sprue. Using this information, one can calculate the size of the outlet orifice, shown as a_5 in Fig. 4-97. This completes the rotor design procedure, and the required power input to the rotor can now be computed. This input power must provide for the frictional or aerodynamic work required to overcome the boundary-layer drag on the exterior surfaces of the wheel, overcome inertial and frictional losses in the sprue, and provide the kinetic energy carried away by the coal leaving the rotor. The computer calculates these for each wheel design.

Representative plots of computer output data are shown in Figs. 4-98 through 4-102.

Figure 4-98 shows the sprue outlet diameter as a function of the sprue entrance radius R_1 for various sprue lengths. As shown, longer sprue lengths yield larger exit diameters, a desirable condition when plugging of the sprue is of concern.

Figure 4-99 shows the number of sprues required as a function of R_1 for different sprue lengths.

Figure 4-100 shows the outlet orifice diameter as a function of R_1 for different sprue lengths.

Figure 4-101 shows how the wall slope angle α varies as a function of R_1 for different sprue lengths.

Fig. 4-98 Sprue Outlet Diameter as a Function of R_1 and Sprue Length

Fig. 4-99 Number of Sprues in Rotor as a Function of R_1 and Sprue Length

Fig. 4-100 Sprue Orifice Diameter as a Function of R_1 and Sprue Length

Fig. 4-101 Sprue Wall Slope Angle as a Function of R_1 and Sprue Length

Fig. 4-102 Specific Power Requirement as a Function of R_1 and Sprue Length

The power requirement is shown in Fig. 4-102 as a function of R_1 and the sprue length. Using the information displayed in Fig. 4-101, one can superimpose lines of constant wall slope angles into Fig. 4-102.

An "optimum" design can now be selected. If we limit the wall slope angle to 15 deg, the minimum power requirement criteria would indicate a sprue length of about 22 in. with an R_1 of 13 in. The rotor would use 40 sprues with a sprue outlet diameter of about 0.9 in. and an orifice diameter of 0.24 in. If desired, larger values of R_1 and larger sprues can be used with a relatively small penalty in the power requirement in the sample case.

At this point, a check should be made to determine the adequacy of the spinup region; that is, we verify that selection of R_1 provides sufficient spinup capacity for the design feed rate. Similarly, the body force and pressure gradient distribution are examined to ensure that the body force is equal to or greater than the pressure gradient throughout the sprue. Our present criteria demand that this should be true even if the coal flow stops. This ensures that in a plugged sprue coal will not be ejected from the sprue, thereby causing blowback. If the rotor fails to meet any of the established selection criteria, a new generic design for the sprue must be conceived. The overall design process is illustrated in flow chart form in Fig. 4-103.

The method described above has been used to design a 50-ton/h rotor for a design delivery pressure of 600 psi. This effort is described in the following section.

Design Study for a 50-ton/h Feeder. A 50-ton/h Kinetic Extruder design study has been carried out to illustrate the optimization procedure and to identify key factors. This particular study was carried out early in Phase III before most of the component testing was completed. Therefore, some of the numerical data are obsolete. However, the design procedure remains generally valid. Process and coal specifications are assumed to be as follows:

- o Coal inlet pressure: 14.7 psia
- o Coal outlet pressure: 600 psi

Fig. 4-103 Kinetic Extruder Rotor Design Flow Chart

- o Coal throughput: 50 ton/h
- o Coal density in Extruder sprues: 40 lb/ft³
- o Coal porosity in Extruder sprues: 0.50
- o Coal permeability in Extruder sprues: 6×10^{-12} ft²

The above coal properties are representative of the nominal 70-percent passing 200-mesh coal tested in our 1-ton/h Kinetic Extruder Model No. 2 unit.

Figure 4-97 illustrates the generic sprue configuration that has been assumed for this study, and also defines the nomenclature. The sprue is double-tapered with a rectangular inlet tapering to a conical secondary section. The aerated outlet cone shown has the purpose of maintaining the design flowrate. The hopper wall slope angles, α_1 and α_2 , are particularly important design factors; these angles must be steep enough to permit uniform flow across the sprue cross section. Normally, we select entrance conditions such that $\alpha_1 = \alpha_2$.

Figure 4-104 shows results from the gas percolation flow computer calculation for a particular sprue shape ($AR = 80$, $A_{23} = 3.49$, $BP = 0.4$) and several sprue lengths. The parameters plotted are the sprue outlet dimensionless pressure gradient

$$\left(\frac{dP}{dZ} \right)_3 ,$$

and the gas specific leakage, Q , both as a function of the coal flowrate per unit area,

$$\left(\frac{\dot{m}}{a} \right)_3 ,$$

at the sprue outlet.

The specific coal flowrate yielding zero leakage is selected as the design point for each sprue length.

Fig. 4-104 Relationship Between Gas Flow Parameters and Coal Flowrate.
Sprue Area Ratio = 80, $k = 6 \times 10^{-12}$, 600-psi Outlet Pressure

Figure 4-105 shows the zero leakage producing coal mass fluxes as a function of sprue length for three sprue area ratios. The generic shape of all sprues is identical to the one used in Fig. 4-104.

Figure 4-106 shows calculated specific power requirements for the $AR = 60$, $L = 24$ -in. case using R_1 as a parameter. The specific mass flux corresponding to zero leakage is indicated in the figure and lines of constant wall slope angle have been superimposed.

Figure 4-106 indicates that, other things being equal, the rotor with the minimum R_1 requires the lowest power. On the other hand, as can be seen in Fig. 4-106, such a design is unsatisfactory because the sprue wall slope angles are much too steep for a design at the $Q = 0$ operating condition.

Based on available information, the wall slope angle should be less than approximately 20 deg. Thus, none of $L = 24$ in., $AR = 60$ designs is satisfactory for operations at the $Q = 0$ point. The other sprue length and area-ratio designs are similarly poor designs with regard to wall slope angle. One is therefore faced with the choice of going to even larger-diameter wheels or operating at higher than the $Q = 0$ coal efflux rate. Both options require higher power inputs and are therefore not desirable.

The remaining alternative is to change the basic sprue configuration. The wall-angle problem is due to the fact that for a 50-tons/h throughput, a large amount of total sprue inlet area is needed, leading to a thick rotor and steep wall angle as illustrated by configuration (a) of Fig. 4-107. The obvious solution is simply to stack up two or more tiers of sprues as shown in configuration (b) of Fig. 4-107. This permits use of the same total sprue inlet area with reduced wall angles.

The rotor configuration (b) shown in Fig. 4-107 is used for the remainder of this study. It should be noted that the same generic sprue shape is used in

Fig. 4-105 Coal Efflux as a Function of Sprue Length and Area Ratio for Obtaining Zero Gas Leakage. $k = 6 \times 10^{-12} \text{ ft}^2$, $P = 40.8$ (600 psia)

Fig. 4-106 Specific Power Requirements for 50-tons/h, 600-psia, Kinetic Extruder Rotor Designs as a Function of Sprue Outlet Coal Efflux Rate

CONFIGURATION (a)

CONFIGURATION (b)

Fig. 4-107 Rotor-Sprue Configurations

this stacked design scheme. Therefore, the data shown in Figs. 4-104 and 4-105 are still valid, and the calculations do not have to be repeated. Figures 4-108 and 4-109 show calculated data for this configuration for the $AR = 60$ case. Specific power and slope angle data are shown in Fig. 4-108. Numbers of sprues contained in the wheel and sprue outlet diameter data are presented in Fig. 4-109.

Note that according to the power curves shown in Fig. 4-108, the maximum permissible value of the wall slope angle, α , still determines the lower limit of R_1 and the required power. For a given angle α , the values of R_1 and L can be selected to minimize the power requirement. Slope angles could, of course, be further reduced by using more than two stacked sets of sprues, but this would lead to smaller sprue orifice sizes that could lead to plugging tendencies.

Similar calculations were carried out for area ratios of $AR = 45$ and $AR = 80$, and the minimum power requirement for a given wall slope angle was determined and the corresponding values of R_1 and L noted.

The power requirements as a function of AR are shown in Fig. 4-110. The solid lines represent the optimal choice of R_1 and L for the indicated slope angle and the dashed line corresponds to the nonoptimal design using fixed values of $R_1 = 15$ in. and $L = 18$ in. One may conclude from an examination of Fig. 4-110 that: (1) the maximum allowable slope angle remains a key parameter, and (2) a low sprue area ratio is desirable to minimize power.

Area ratios, however, cannot be reduced without a sacrifice in stability. Figure 4-111 shows the calculated pressure gradient distributions in the $AR = 45$ and $AR = 80$ sprues at both their design coal throughput, and in a plugged sprue condition ($\dot{m} = 0$). The body force distributions at design wheel speed are also shown. Note that for both area ratios the dP/dZ distributions are very stable when coal is flowing. That is, the body force is equal to or greater than dP/dZ . However, one should anticipate that in

Fig. 4-108 50-tons/h, 600-psia Kinetic Extruder Parameters for AR = 60

Fig. 4-109 50-tons/h, 600-psia Kinetic Extruder Parameters for $AR = 60$

Fig. 4-110 Specific Power Requirements for a 50-tons/h Kinetic Extruder Operating at the Zero Gas Leakage Point. Two stacked sets of sprues. Accumulator pressure: 600 psia, $k = 6 \times 10^{-12}$ ft²

Fig. 4-111 Pressure Gradient Distributions in Sprues of Different Area Ratios. Pressure ratio $P = 40.8$ (600 psia)

normal practice some sprues will occasionally become plugged by foreign objects. The sprues, therefore, should be designed also to be stable at zero coal flow. Pressure gradients at zero coal flow are plotted as dashed lines in Fig. 4-111. The $AR = 45$ case can be seen to be slightly unstable in that at the sprue inlet position the pressure gradient just exceeds the centrifugal force. Under this condition, coal flow would tend to reverse near the inlet position, possibly leading to a blowback.

Therefore, it is recommended that area ratios be chosen large enough so that stability is achieved at zero coal flow even if additional power is required.

It may be noted at this point that the sprue shape initially chosen is non-optimum in the sense that the pressure gradient curves (see Fig. 4-111) are much steeper than the body force. The problem is that the sprues are too sharply converging in the second section (i.e., A_{23} is too large). At this point a change should be made and another iteration of the design process carried through. Since the purpose of the present study was mainly to illustrate the optimization process, this was not carried further.

Useful Scaling Parameters. Scaling parameters may be derived from the theoretical sprue flow equations that are useful for quick estimates of the requirements for larger or higher pressure Kinetic Extruder units. For scaling to higher throughputs, at constant delivery pressure, the similarity parameter is

$$\frac{L\dot{m}}{k a_3} = C_1 = \text{a constant} \quad (4.7)$$

where

- L = sprue length
- $\dot{m}$ = total throughput
- a_3 = total sprue outlet area
- k = coal permeability

All geometrically similar sprue flows with the same value of $Lm/k a_3$ will exhibit the same pressure distribution. Thus, for example, Eq. (4.7) implies that to scale to a 10x larger $\dot{m}$, a_3 must also be increased by a factor of 10 to obtain similar performance (assuming L/k remains fixed).

Similarly for scaling up or down in delivery pressure at the same throughput, the parameter

$$\frac{P}{DL \omega^2} = C_2 = \text{a constant} \quad (4.8)$$

where

p = delivery pressure

ω = rotor speed

D = rotor diameter

Equation (4.8) says, for example, that to double the delivery pressure, either the speed ω must be increased by a factor of $\sqrt{2}$, or the rotor size must be increased so that DL is 2x higher, or some combination thereof.

Based on tests of the 1-ton/h experimental machines, approximate values for these scaling parameters are:

$$C_1 = 3 \times 10^{-13} \text{ tons/h/ft}^3$$

and

$$C_2 = 1.2 \times 10^{-4} \text{ atm-s}^2/\text{ft}^2$$

The C_2 value is restricted to pumping in a nitrogen atmosphere.

4.9 SUBSYSTEMS FOR THE KINETIC EXTRUDER

A Kinetic Extruder (KE) pressurizing system requires a number of subsystems to support the KE unit itself. These subsystems include transport systems for bringing the coal to and from the KE unit, a prime mover for supplying the required power to the KE, a lubrication and cooling system for the bearings and seals, a gas supply system for controlling the receiver pressure, supplying purge and fluidizing gases and providing control gas if fluidic turndown control is used, a rotor suction system to maintain the desired rotor inlet pressure, and an instrumentation and control system to provide automated operation of the feed system and integration into the overall plant operation.

A study was carried out to define the various auxiliary systems that are used. This study is documented in Appendix E. The main conclusion of the study is that commercially available equipment is satisfactory for all of the subsystems. Therefore, beyond the KE unit itself and some software development for the control system, no major development work on the subsystems appears to be necessary.

4.10 PROJECTED POWER REQUIREMENT FOR COMMERCIAL SIZE FEEDERS

Based on preliminary designs of rotor configurations, an estimated performance map was established for estimating feeder performance. This performance map is shown in Fig. 4-112. The specific power requirements, HP/ton/h, is shown as a function of feeder size for various delivery pressures, assuming a nitrogen gas environment. Superimposed are lines of temperature rise experienced by the coal as it passes through the feeder. These heating curves were calculated assuming that all the power input is transformed into heat energy and eventually transferred to the coal. No energy is lost.

The shape of these curves shows that low throughput feeders require much power to overcome the aerodynamic drag of the surrounding high-density nitrogen gas. As the feeder size increases, the specific power decreases until it reaches asymptotically the value associated with the internal work only.

Fig. 4-112 Kinetic Extruder - Projected Performance Map in N_2 Atmosphere

Similar calculations were carried out for a hydrogen atmosphere as is typically encountered in plants for converting coal into liquid products. The estimated performance map for this situation is shown in Fig. 4-113.

Comparison of Figs. 4-112 and 4-113 clearly indicates the drastic reduction in power requirements and a rapid dropoff of specific power to the asymptotic internal work value when hydrogen is used.

Fig. 4-113 Kinetic Extruder - Projected Performance Map in H_2 Atmosphere

Section 5
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Appendix A

COAL FEEDER BEARING AND SEAL DESIGN STUDY

This appendix describes the work carried out by Mechanical Technology Incorporated (MTI) of Latham, N. Y., under subcontract to Lockheed. The text was prepared from the report written by Mr. Robert L. Osterman of MTI.

A.1.0 OBJECTIVE

The objective of this phase of this task is to design and seal systems for commercial feeder units of 7 and 50 ton/h capacity. The design life objective is 8,000 h. The seals must prevent the incursion of the suspended coal powder into the seal and bearing areas. In addition to normal operation, the bearing systems must be designed to be capable of taking rotating unbalance loads of 4,600 and 112,000 lb, respectively, in the event that half the coal feeder sprues located on one-half of the wheel circumference, are suddenly emptied. The machines must be capable of safe shutdown after such an instance without damage to the bearings or seals.

A.2.0 SUMMARY

This appendix presents the results of a journal bearing and seal design study for the straddle mounted 7 and 50 ton/h coal feeder units.

The design is based on the use of water as the bearing lubricant, primarily because of its compatibility with the feeder. The use of oil was considered, but abandoned, since the failure of any shaft seals would produce serious consequences in terms of coal agglomeration. The water is also used as a coolant.

The basic bearing design selected is the shrouded-stepped pad configuration. This type of bearing has a number of desirable characteristics. The benefits of both

hydrodynamic and hydrostatic action can be obtained. The bearings can be designed so that hydrodynamic action alone is sufficient for normal operation, making the equipment failsafe in the event of lubrication system failure. This hybrid capability also provides the overload capacity to handle heavy unbalance conditions. In addition, the multiple pad bearing provides a high degree of stability against bearing whirl and wear.

The selected bearing and sealing system uses the basic shrouded step bearing as a pressure break down structure. This reduces the number of pressure seals to just one seal, the second seal being merely a dust seal, having no pressure sealing requirements.

Water at 625 to 650 psi is fed to the buffer zone between the face seal and the bearing. This same feed supplies the hydrostatic bearing portion of the bearing. The bearing designs developed here do analytically meet all design objectives. Verification testing prior to commitment to final design is recommended.

A.3.0 COAL FEEDER CONFIGURATION

Two coal feeder units are considered in the present study — one capable of feeding 50 tons of coal per hour, the other, 7 ton/h. For both units the pressure and temperature of the gas and powdered coal dust mixture within the wheel center is -3 to +1 psig and 200° F. The delivery pressure in the tank is 600 psig and the temperature is assumed to be 400° F. The coal is assumed to be transferred from the dryer at 200° F and experiences a 200° F temperature rise during the extrusion from the wheel.

Three bearings and three seal assemblies are required for a typical coal feeder. The bearings and sealing configuration can be seen in the coal feeder bearing/seal layout. Fig. A-1. One pair of bearings is needed to support the main rotor shaft, and the remaining bearing is needed to center the stationary feed pipe within the hollow section of the rotor. One of the three seal assemblies, designated the inner seal, is located inside the wheel, sealing between the stationary central inlet duct and the wheel. The other two seals, designated the outer seals, seal between the outside of the drive shaft and the stationary housing of tank.

A-3

Fig. A-1 Coal Feeder Bearing/Seal Layout

The work reported herein encompasses the bearing and seal designs for both 7- and 50-ton-per-hour coal feeder units.

A.4.0 50-T.P.H. UNIT

The 50-ton-per-hour unit, because of its large size and the quantity of coal it moves, imposes harsh conditions of operation on the bearing designs. In addition to normal operation, the bearing systems must be designed to be capable of taking a rotating unbalance load of 112,000 lb in the event that half of the coal feeder sprues, located on one-half of the wheel circumference, are suddenly emptied. The machine must be capable of safe shutdown after such an instance without damage to the bearings. The rotational speed range of this unit is 2140 rpm to 2350 rpm and the rotor shaft power is 1000 hp at 2400 rpm. The design life for the bearings and seals in this machine is 8,000 hours or eleven months of continuous operation.

A.4.1 Shaft Loading

Based on a wheel weight of 3,000 lb, normally operating at maximum residual unbalance of 13.50 oz-in. (average quality balancing for large flywheels), the main shaft bearing reactions at 2,350 rpm will each be 1,567 lb. At the emergency or "blowdown" condition, based on a 112,000-lb rotating load, the load at each bearing will be 57,500 lb. These are purely radial loads, rotating at a frequency of once per revolution. The extreme loading condition is, therefore, 57,500 lb at each main shaft bearing.

It was assumed that the "blowdown" condition could occur twice during a month's time of operation and that the wheel could stop rotating in less than 5 min. It follows, therefore, that, during eleven months' running, twenty-two five-minute pulses would occur. The normal loading condition (1,567 lb/bearing) would exist during 99.977 percent of the operating life of the unit, and emergency loading (57,500 lb/bearing) would be experienced 0.023 percent of the unit's operating life. Although it is evident that the normal operational mode predominates during the unit's lifetime, the main shaft bearings must be designed with excess capacity in order to withstand the overload conditions.

Since the stationary feed pipe, the feed pipe locating bearing, the hollow section of rotor and the main shaft bearing are all concentric, the loads which are imposed on the inner locating bearing are a function of the deflections of the feed pipe due to unsymmetrical loading and the amplitude of any vibrations which may be present in the pipe. Because the response of the feed pipe is largely uncertain at this time, the feed pipe locating bearing has been designed to have a stiffness of the same order as that of the main shaft bearing.

A.4.2 Hydrostatic Bearing Design

Water lubricated hydrostatic fluid film bearings are used to support the main rotor shaft and to center the stationary feed pipe within the hollow section of the rotor. The design approach is based on the use of water as the bearing lubricant primarily because of its compatibility with the feeder. The use of oil was considered but abandoned since the failure of any shaft seals in separating oil from the process coal would produce serious consequences in terms of coal agglomeration. Water is used throughout the sealing scheme described below, and also functions as a coolant. The particular bearing design utilized is the axial flow, hybrid-stepped pad configuration having both hydrostatic and hydrodynamic capability. All the journal bearings in the two feeder units are of this type.

The basic shrouded step design is shown in Fig. A-2. As adapted to a journal bearing, a number of step sections of the type shown, usually four, are located in a cylindrical array.

Referring to Fig. A-2, the bearing surface incorporates a depressed area. This area is located at a distance, Δ , below the surface forming a step in the direction of motion, at a distance, B , from the inlet groove which has a length, D , in the direction of motion. The total active length per section is C . Following the step, a section of length, A , seals the step area from the following feed groove. This section forms the familiar Rayleigh step bearing. It can be shown that it has a load capacity comparable to the more conventional tapered design.

Fig. A-2 Basic Shrouded Step Bearing Configuration

To improve the bearing performance, side leakage is restricted by introducing side shrouds of widths, L_1 and L_2 , which together with the width of the step area, W , form the total bearing width, L . If fluid is freely admitted to the feed grooves at a small feed pressure, the shrouded step bearing operates in a purely hydrodynamic mode. If feed orifices, DO , are interposed in the feed lines to the feed grooves, and if the feed pressure is substantially raised, then hydrostatic action is provided and the shrouded step bearing operates in its hybrid mode. In this mode, as the load decreases, film thickness, h_1 , is increased, flow through the bearing increases, the pressure drop across the orifice, DO , increases, and the pocket pressure decreases. This process continues until the bearing pressures match the imposed load. The reverse of this process takes place when the load increases.

The shrouded step bearing has a number of desirable characteristics which have been the reason for its selection for this design. One is its hybrid capability which provides for both hydrodynamic and hydrostatic action. The bearings are designed so that hydrodynamic action alone is sufficient for normal operation, making the equipment failsafe in the event of lubrication system failure. The hybrid capability then provides a high degree of stability against bearing whirl, a condition that must be avoided in low-load rotating machinery. A third very desirable characteristic of the shrouded step bearing is its stability against wear, if wear should occur. The load capability of the bearing as a function of step depth, Δ , reaches a maximum as Δ is increased, and then decreases rather slowly. Therefore, if the design value of Δ is placed beyond the maximum point, load capability does not suffer very much, but if wear occurs, and Δ is decreased, the bearing capacity may actually increase. The shaft bearings can also serve as pressure breakdown seals, thus only one pressure seal is needed per bearing at the interface with the coal suspension. Water at 625 to 650 psi is fed to the buffer zone between the face seal and the bearing, and this same feed supplies the bearing.

The configuration of the axial flow shrouded step bearing and the boundary conditions at the ends of the bearing are shown in Fig. A-3. The pocket width, W , comprises 50 percent of the bearing length, L ; the width of the boundary lands ($L_1 + L_2$) account

Fig. A-3 Axial Flow Shrouded Step Bearing Configuration

for the remainder. Existing at one side of the bearing is the buffer zone environment (PSUP), at the other are ambient pressure conditions (14.7 psia). Pressure tapped off the buffer zone is supplied to the bearing pocket via an inlet orifice (DO at PSUP). The low-pressure (ambient) side boundary land (L1) is intentionally made wider than the high pressure (buffer zone) side boundary land (L2). This procedure is followed to take maximum advantage of the hydrostatic load capacity that is produced by the pressure drop in the lubricant as it flows from the pocket to the ambient side of the bearing.

A.4.2.1 Main shaft bearing performance. Four key parameters were optimized in designing the axial flow shrouded step bearing. These parameters are: the groove inlet orifice diameter, DO; the step length ratio, B/A ; the step height ratio, h_2/h_1 ; and the shroud width ratio, $L1/L2$. Another key parameter in the bearing design is the length to diameter ratio, L/D , where D is the bearing diameter. Although the bearing efficiency, in terms of load capacity per unit of lubricant flow, improves as this ratio is increased, the ability of the bearings to handle misalignment has dictated that bearing length be kept down to an L/D less than or equal to 1.25.

The design calculations for the bearings were performed using existing MTI computer software for the shrouded step hybrid bearing which automatically accounts for turbulence and inertia effects. The program assumes as a variable input the boundary conditions and the land widths at either end of the bearing.

Ten in. diameter bearings, 12.5 in. long, were chosen to support the main shaft.

Flowrates are a strong function of bearing clearance, varying approximately as the cube of clearance. For this reason, the bearing clearances have been selected to be relatively tight. A radial clearance of 0.002 in. for the 10-in.-diameter bearing has been selected.

Power consumption in these bearings consists of the power consumed in the bearing itself, and the power consumed in pressurizing the water lubricant. In this particular design, the shaft power used by the bearings is low, on the order of 4 hp each. The power to pump the water to feed pressure at 650 psi is also low because the bearing leakage flows are low. The pressurization power for each main shaft bearing will run approximately 2.7 hp.

Figure A-3 shows the basic optimization of the inlet orifice diameter for a four-pocket axial flow shrouded step bearing design, as described above for an eccentricity of 0.8. Note that after a certain point, increasing the inlet orifice diameter does not result in a corresponding increase in load capacity for the bearing. Prior work has been used to set a value of circumferential step to land ration $(B/A) = 2.55$, and step height ratio $(h_2/h_1) = 1.30$. Slight deviations from these values have a negligibly low influence on the load capacity of the bearing. The value selected for $(L1 + L2)/L$ is more a function of balancing load capacity against leakage rate. In order to reduce leakage rate, a value of one half for $(L1 + L2)/L$ has been selected. Figure A-4 shows the optimization for shroud width ration $L1/L2$. A value of four has been selected for this quantity so that the bearing would still retain the desirable characteristics of a shrouded step bearing. The load capacity for the bearing, at an eccentricity ratio of 0.8, is approximately 85,000 lb at $L1/L2$ equal to four. Since the extreme loading condition corresponds to 57,500 lb per bearing, it is apparent from this curve that the design selected has an additional capacity available. The flow curves shown in Figs. A-4 and A-5 depict the leakage across the low-pressure side boundary land (L1). The leakage (at an eccentricity of 0.8), levels off at approximately 5.5 gallons per minute. This leakage can easily be received by a bearing drain, and thus the design functions well as a pressure breakdown device.

Figure A-6 shows load capacity and leakage flow plotted against eccentricity ratio for three different bearing clearances. For bearing clearances in the order of 4 to 6 mils in diameter, the operating eccentricity at emergency loading would be located below 0.9. Within this same clearance range, the bearing would normally operate at an eccentricity below 0.1. At the larger clearance, the leakage flow at no load is 70 percent greater than that at the nominal clearance of 4 mils (7.8 versus 4.6 gpm).

Fig. A-4 50-Ton/h Unit Main Shaft Journal Bearings, Inlet Orifice Diameter Optimization

Fig. A-5 50-Ton/h Unit Main Shaft Journal Bearings, Shroud Width Ratio Optimization

Fig. A-6 50-Ton/h Unit Main Shaft Journal Bearings, Effect of Bearing Clearance on Load Capacity and Leakage Flow

The stiffness of the bearing was established from the load eccentricity curve at nominal clearance as shown in Fig. A-7.

The bearing design dimensions are given in Table A-1; the bearing is shown in Fig. A-8.

A.4.2.2 Inner feed pipe locating bearing performance. One 7.5-in.-diameter bearing, 9.375 in. long, is used to center the stationary feed pipe within the hollow section of the rotor. The proportions of this bearing are the same as those of the main shaft bearing; the bearing surface is located on the outer diameter of the stationary feed pipe, and the journal surface is situated on the inner diameter of the hollow rotor. Thus, this bearing can be considered as a conventional journal bearing turned inside out.

The radial clearance selected for the 7.5-in. bearing is the same as that for the concentric main shaft bearing 0.002 in. The shaft power consumed by the bearing is in the order of 1.0 to 1.5 hp.

Figure A-9 shows the inlet orifice diameter optimization for the bearing with an eccentricity ratio of 0.8. The load capacity for the bearing, at this eccentricity is approximately 25,000 lb, and the axial leakage across the bearing from the seal buffer zone to the ambient side of the bearing is approximately 7.2 gpm. Normally, the bearing will be operating at an eccentricity of nearly zero. During normal operation the leakage from the bearing will be approximately 4.9 gpm.

The stiffness of the bearing was established from the load eccentricity curve at nominal clearance, as shown in Fig. A-10.

The bearing design dimensions are given in Table A-2. The bearing is shown in Fig. A-11.

Fig. A-7 50-Ton/h Unit Main Shaft Journal Bearings, Bearing Stiffnesses at Nominal Clearance

Table A-1
50 TON/H MAIN SHAFT BEARING DESIGN
(Ref. Fig. A-8)

Type	Shrouded Step
Feed Type	Hydrostatic
Lubricant	Water at 100°F
Feed Supply Pressure (psi)	650
Diameter (in.)	10
Length (in.)	12.5
Clearance, Radial (in.)	0.002
Number of Pockets	4
Subtended Angle, Feed Groove (rad)	0.0376
Subtended Angle, Step (rad)	1.1008
Subtended Angle, Land (rad)	0.4324
Low Pressure Side Land Width (in.)	5.00
High Pressure Side Land Width (in.)	1.25
Step Height (in.)	0.002
Orifice Diameter (in.)	0.1090
Maximum Load Applied (lb)	57,500
Eccentricity Ratio at Extreme Loading Conditions	0.70
Flow, no load, 100°F (gpm)	4.6
Stiffness at $e=0$ (lb/in.)	24.7×10^6

Fig. A-9 50-Ton/h Unit Inner Feed Pipe Locating Bearing, Inlet Orifice Diameter Optimization

Fig. A-10 50-ton/h Unit Inner Feed Pipe Locating Bearing, Bearing Stiffnesses at Nominal Clearance

Table A-2
50 TON/H INNER FEED PIPE LOCATING BEARING DESIGN
(Ref. Fig. A-11)

Type	Shrouded Step
Feed Type	Hydrostatic
Lubricant	Water at 100° F
Feed Supply Pressure (psi)	650
Diameter (in.)	7.5
Length (in.)	9.375
Clearance, Radial (in.)	0.002
Number of Pockets	4
Subtended Angle, Feed Groove (rad)	0.0376
Subtended Angle, Step (rad)	1.1008
Subtended Angle, Land (rad)	0.4324
Low Pressure Side Land Width (in.)	3.750
High Pressure Side Land Width (in.)	0.9375
Step Height (in.)	0.002
Orifice Diameter (in.)	0.1090
Maximum Load Applied (lb)	—
Eccentricity Ratio at Extreme Loading Conditions	—
Flow, no load, 100° F (gpm)	4.9
Stiffness at $e=0$ (lb/in.)	7.96×10^6

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Fig. A-11 Inner Locating Bearing

A.4.3 Seal Designs

The two types of seals required are the inner seal and the outer seal. The inner seal is located inside the wheel, sealing between the stationary central inlet duct and the wheel. The other, designated the outer seal, seals between the outside of the hollow drive shaft and the stationary housing.

A.4.3.1 Inner seal. The inner seal, on one side, is exposed to 650 psi water which is entering the locating bearing between the inlet duct and the hollow rotor, and, on the other side, to fine coal dust suspended in gas at a pressure of ± 1 psi gauge. The coal is assumed at 200°F, as fed from the coal crusher and dryer.

This environment dictates the use of a buffer type seal. Experience with fine particle suspensions indicated that fine particles could work their way upstream if a gas buffer were not used and could result in damage to the seal parts and eventually damage to the bearing. Furthermore, there is an additional condition to be met by this seal, namely, a blowdown condition in which the pressure inside the wheel rises to the tank pressure of 600 psi due to failure of one or more of the sprues to remain filled with packed coal. This pressure rise can drive particulate matter back into the seal if a gas buffer is used. For these reasons, water was considered as the buffer fluid.

In the case of using water as the buffer fluid, some water will be passed into the coal suspension. Information obtained indicates that water intrusion can be tolerated in small amounts, and that water intrusion at the rate of 1 percent of the coal feed rate, for both seals taken together, can be tolerated without damage. For a 50-ton/h coal feeder, this sets a limit of 1,000 lb/h for the water intrusion rate from the buffer seals.

A number of seal types were reviewed, singly and in combination, for this application. Some of the types considered were: lip seals, contact face seals, non-contact face seals, shaft riding seals, labyrinths, and packings. The selected design consists of a balanced face type contact seal which provides the primary separation between the

bearing lubricant and the interior of the wheel. Water at 625 to 650 psi is fed to the buffer zone between this face and the bearing and this same feed supplies the hydrostatic portion of the bearing. This sealing configuration is shown in the coal feeder bearing/seal layout, shown in Fig. A-1. The buffer fluid is provided by a constant displacement pump so that required flow to the buffer zone is maintained.

A.4.3.2 Outer seal. The outer seal, on one side, is exposed to 650 psi water which is entering the main shaft bearing, and, on the other side, to a coal suspension at 600 psi. The considerations are very similar to those for the inner seal, with the exception that the outer seal does not have to face the sudden pressurized condition. The selected design consists of the same components as the inner seal. The drawing of the coal feeder bearing/seal layout is shown in Fig. A-1.

A.4.4 Rolling Element Bearing Design

A bearing and seal design utilizing spherical roller bearings was examined for the straddle mounted 50-ton/h unit. The spherical roller bearings may have a cost advantage over the hydrostatic bearings, but will certainly require more complex sealing arrangements since roller bearings will have to be oil lubricated for this application.

The rolling element bearing support and seal layout is shown schematically in Fig. A-12. The main rotor shaft is supported by a pair of Torrington spherical roller bearings of 10.2362-in. bore. Bearing lubrication is by force-fed oil, circulated through the bearings via integral oil passages. The bearings are located outboard of the feeder housing.

Two sealing systems are required. One, designated the inboard seal, seals between the outside of the hollow drive shaft and the stationary housing. The other, designated the outboard seal, seals between the bearing oil drain cavity and the inboard sealing system buffer water drain cavity.

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Fig. A-12 Coal Feeder Seal Envelope Layout: Rolling Element BRC Support

The inboard sealing system consists of a balanced face type contact seal providing the primary separation between the bearing discharge and the interior of the feeder housing (coal suspension at 600 psi and 400° F). Outboard of this face seal is a shaft riding seal to control the exit flow from the buffer zone to the buffer fluid drain which is at atmospheric pressure. Water buffer fluid at 600+ psi is fed to the space between these two seals so that a flow of water exits from the outer shaft riding seal. The outer shaft riding seal has to support a pressure difference of 600+ psi from the water buffer zone to ambient. Since the flow of buffer fluid past this seal is not expected to be negligible, a buffer fluid drain cavity has been provided.

The outboard sealing system consists of a number of close clearance shaft seals, slingers, and drain cavities arranged in series. The lubricating oil drain cavity is the primary expedient in collecting the lubricating oil spray from the end of the roller bearing. The bearing drain cavity is defined by a close clearance seal on the other side of which is an oil slinger and backup drain. The oil is further retarded in its travel down the shaft by an oil windback which is drained internally. Oil leaking past the windback is contained by an additional slinger and bearing drain. This last slinger and one which is located in the buffer water drain cavity are arranged back to back, separated by a single labyrinth tooth.

Inserts of nonwetting teflon are used on the shaft as the mating surfaces for the close clearance shaft seals.

Oil leaking past the outboard sealing system would enter the inboard sealing system buffer water drain cavity. If this was found to occur, the discharge from the buffer water drain cavity would either be directed to an oil/water separator or to waste depending on the quantity of oil contained in the water.

The operating speed range of the 50-ton/h coal feeder is approximately three times the limiting speeds of the spherical roller bearings available in a 10-in. bore. However, the loads imposed on the bearings are light compared to the basic dynamic

capacity of these bearings. For the particular spherical roller bearing selected, the standard Torrington B₁₀ fatigue life calculation yielded a bearing life of 2,900 years. This life calculation does not take into account the centrifugal forces acting on the rollers due to the high speeds. It is therefore recommended that if a rolling element bearing design is to be seriously considered, further analysis on the bearing application be performed. The study should have as its main objective the determination of whether the fatigue life of the rolling element bearings is fully comparable to that of the alternative fluid film bearings.

A.5.0 7 TON/H UNIT

The 7-ton/h coal feeder is the smaller of the two extruder units considered in this study. Because of its smaller size, this unit imposes relatively moderate loading conditions, during operation, on the bearing designs. In addition to normal operation, the bearing systems must be designed to be capable of taking a rotating unbalance load of 4,600 lb corresponding to the "blowdown" condition described for the 50 ton/h unit. The machine must be capable of safe shutdown after such an instance without damage to the bearings. The rotational speed of this unit is 3700 rpm and the rotor shaft power is 180 hp at 3700 rpm. Design life for the bearings and seals in this machine is 8,000 h, or 11 months' continuous operation.

A.5.1 Shaft Loading

Based on a wheel weight of 750 lb, normally operating at a maximum residual unbalance of 3.0 oz-in. (average quality balancing for large flywheels), the main shaft bearing reactions at 3700 rpm will each be 413 lb. At the emergency or "blowdown" condition, based on a 4,600-lb rotating load, the load at each bearing will be 2,675 lb. These are purely radial loads, rotating at a frequency of once-per-revolution. The extreme loading condition, therefore, is 2,675 lb at each main shaft bearing.

It was assumed that the "blowdown" condition occurs in the 7-ton/h unit with the same frequency as it occurs in the 50-ton/h unit.

Since the loading imposed on the inner locating bearing by the stationary feed pipe is uncertain in this unit as well, the locating bearing was designed with a stiffness of the same order as that of the main shaft bearings.

A.5.2 Bearing Designs

Axial flow, shrouded step bearings are used to support the main rotor shaft and to center the stationary feed pipe within the hollow section of the rotor.

The configuration and proportions of the bearings used in the 7-ton/h unit and 50-ton/h unit are identical since the design considerations are the same.

A.5.2.1 Main shaft bearing performance. Two 6-in. -diameter bearings, 7.5 in. long, were chosen to support the main shaft. The bearing configurations are the same as those of the 50-ton/h main shaft bearings, although the various dimensions have been scaled down to reflect the new shaft size and loading.

To keep the bearing leakage to a reasonable level, a relatively tight radial clearance of 0.00125 in. for the 6-in. -diameter bearing was selected. The shaft power used by the bearings is of the order of 2.5 hp each.

Figure A-13 shows the inlet orifice diameter optimization for the bearing with an eccentricity of 0.8. The load capacity for the bearing, at an eccentricity ratio of 0.8, is approximately 41,600 lb. During normal operation, the bearing will be running at an eccentricity of nearly zero. The axial leakage across the bearing from the seal buffer zone to the ambient side of the bearing will be approximately 1.4 gpm for all conditions of operation encountered in this unit.

The stiffness of the bearing was established from the load eccentricity curve at nominal clearance as shown in Fig. A-14.

The bearing design dimensions are given in Table A-3. The bearing is shown in Fig. A-8.

Fig. A-13 7-ton/h Unit Main Shaft Journal Bearings, Inlet Orifice Diameter Optimization

Fig. A-14 7-ton/h Unit Main Shaft Journal Bearings, Bearing Stiffnesses at Nominal Clearance

Table A-3
7 TON/H MAIN SHAFT BEARING DESIGN
(Ref. Fig. A-8)

Type	Shrouded Step
Feed Type	Hydrostatic
Lubricant	Water at 100° F
Feed Supply Pressure (psi)	650
Diameter (in.)	6
Length (in.)	7.5
Clearance, Radial (in.)	0.00125
Number of Pockets	4
Subtended Angle, Feed Groove (rad)	0.0376
Subtended Angle, Step (rad)	1.1008
Subtended Angle, Land (rad)	0.4324
Low Pressure Side Land Width (in.)	3.00
High Pressure Side Land Width (in.)	0.75
Step Height (in.)	0.001
Orifice Diameter (in.)	0.0780
Maximum Load Applied (lb)	2,675
Eccentricity Ratio at Extreme Loading Condition	0.11
Flow, no load, 100° F (gpm)	1.4
Stiffness at $e=0$ (lb/in.)	17.93×10^6

A. 5. 2. 2 Inner feed pipe locating bearing performance. One 3.5-in. -diameter bearing, 4.375 in. long, is used to center the stationary feed pipe within the hollow section of the rotor. The general configuration of this bearing is the same as that of the 50-ton/h mainshaft bearings. As in the 50-ton/h unit, the bearing surface is situated on the outer diameter of the stationary feed pipe, and the journal surface is located on the inner diameter of the hollow rotor.

The radial clearance selected for the 3.5-in. bearing is the same as that for the concentric main shaft bearing, 0.00125 in. The shaft power consumed by the bearing is of the order of 1/3 hp.

Figure A-15 shows the inlet orifice diameter optimization for the bearing with an eccentricity of 0.8. The load capacity for the bearing, at an eccentricity ratio of 0.8, is approximately 4,500 lb. The axial leakage, at this same eccentricity, across the bearing from the seal buffer zone to the ambient side of the bearing is approximately 2.2 gpm. Normally, the bearing will be operating at an eccentricity of nearly zero. During normal operation, the leakage from the bearing will be approximately 1.4 gpm.

The stiffness of the bearing was established from the load eccentricity curve at nominal clearance as shown in Fig. A-16.

The bearing design dimensions are given in Table A-4. The bearing is shown in Fig. A-11.

A. 5. 3 Seal Designs

The sealing requirements for the 7-ton/h unit are identical to those for the 50-ton/h unit, that is, two types of seals are required. One, designated the inner seal, is located inside the wheel, sealing between the stationary central inlet duct and the wheel. The other, designated the outer seal, seals between the outside of the hollow drive shaft and the stationary housing.

Fig. A-15. 7-ton/h Unit Inner Feed Pipe Locating Bearing, Inlet Orifice Diameter Optimization

Fig. A-16 7-ton/h Unit Inner Feed Pipe Locating Bearing, Bearing Stiffnesses at Nominal Clearance

Table A-4
7 TON/H INNER FEED PIPE LOCATING BEARING DESIGN
(Ref. Fig. A-11)

Type	Shrouded Step
Feed Type	Hydrostatic
Lubricant	Water at 100° F
Feed Supply Pressure (psi)	650
Diameter (in.)	3.5
Length (in.)	4.375
Clearance, Radial (in.)	0.00125
Number of Pockets	4
Subtended Angle, Feed Groove (rad)	0.0376
Subtended Angle, Step (rad)	1.1008
Subtended Angle, Land (rad)	0.4324
Low Pressure Side Land Width (in.)	1.750
High Pressure Side Land Width (in.)	0.4375
Step Height (in.)	0.001
Orifice Diameter (in.)	0.0470
Maximum Load Applied (lb)	—
Eccentricity Ratio at Extreme Loading Condition	—
Flow, no load, 100° F (gpm)	1.4
Stiffness at e=0 (lb/in.)	2.62×10^6

A.5.3.1 Inner seal. As in the 50-ton/h unit, the design consists of a balanced face type contact seal which provides the primary separation between the bearing lubricant and the interior of the wheel. Water at 625 psi is fed to the buffer zone between this face seal and the bearing, and this same feed supplies the hydrostatic portion of the bearing. This sealing configuration can be clearly seen in the coal feeder bearing/seal layout, Fig. A-1. The buffer fluid is to be the direct output of a constant displacement pump so that flow is maintained to the buffer zone.

A.6.0 MATERIALS

The material specification for the bearings and seals in both units is as follows:

- Hydrostatic Journal Bearings

- Journal: 317 stainless steel
- Sleeve: 317 S.S.
- Special coatings on shaft and sleeve: chrome oxide over nichrome 5
- Housing: 317 S.S.

- Seals

- Stator
 - . Face insert: carbon graphite*
 - . Carrier: 316 stainless steel
 - . Cartridge: 316 S.S.
- Rotor
 - . Face Insert: LW-15 (flame sprayed tungsten carbide)*
 - . Carrier: 316L S.S.

* Lapped to within a flatness of two helium light bands.

A.7.0 THERMAL ANALYSIS

An approximate thermal analysis was conducted for the 50-ton/h unit. The vertical half of the machine opposite the feed pipe was selected for study. This half of the machine operates at the highest temperature since the inner locating bearing and its heat-absorbing flow of water are absent in this half.

All surfaces, within the machine, exposed to the 400°F coal have high heat-transfer coefficients associated with them. The outside surface of the machine, exposed to stagnant room-temperature air (taken to be 100°F), has poor heat-transfer characteristics. The inside surface of the wheel, adjacent to the end of the feed pipe and exposed to 200°F coal, was taken to be insulated.

Since the heat-transfer to the rotor and housing surfaces is excellent within the machine and poor outside, the main shaft and bearing shell was assumed to be 400°F throughout. Under these conditions, the analysis revealed that the bearing lubricant will reach the boiling point as it exits in the outboard end of the main shaft bearing. The temperature of the lubricant water, when it reaches the outboard end of the bearing, was calculated to be 260°F so that when it leaves the bearing and enters the bearing drain, which is at atmospheric pressure, some water will flash into steam. The phase transition zone is of finite length and exists for a very short distance at the end of the bearing. The lubricant inside the bearing is under pressure so that no phase change will occur in the lubricant while within the bearing proper.

The water flashing at the end of the main shaft bearing should not be detrimental to the bearing's performance. This steam will condense in the bearing drain lines. However, the bearing drain surfaces must be protected from the effects of potential cavitation erosion. The bearing lands and shaft are to be coated with plasma-sprayed chrome oxide for wear resistance. Chrome oxide provides excellent protection against erosion and it would be a simple matter to coat the bearing drain surfaces as well. These coating specifications are given on the engineering drawings for the main shaft and inner locating bearings.

No thermal analysis was conducted for the vertical half of the machine containing the feed pipe, inner locating bearing and hollow section of rotor. It is expected that the additional water circulation through the structure, by means of the inner bearing, will carry heat away from the hollow rotor and, therefore, lower bearing temperatures in this half of the machine.

A.8.0 CONCLUSIONS

1. The bearing and seal system tradeoff studies yielded a design approach, based on the use of water as the bearing lubricant, because of its compatibility with the system.
2. The tradeoff study identified the axial flow, hybrid-stepped pad configuration as being the bearing design ultimately compatible with the given system. This particular bearing functions as a shaft seal, while providing both hydrodynamic and hydrostatic load-carrying capability. This bearing also provides a high degree of stability against bearing whirl and wear, if wear should occur.
3. An optimum bearing geometry was generated which should be capable of safely handling the stipulated load, speed, and dynamic requirements. The operating eccentricities, for all the journal bearings in the two feeder units, at emergency loading conditions would be located below 0.9. The bearings will normally operate at eccentricities below 0.1.
4. Based on a review of the seals compatible with the bearing system, balanced, face-type contact seals were selected to provide the primary separation between the bearing lubricant and the pulverized coal. A buffer zone of water between the contact seal and bearing was dictated by the environment. Water intrusion into the process mixture from the buffer seals is expected to be significantly less than the stipulated maximum tolerable amount of 1 percent of the coal feed rate, total.
5. An approximate thermal analysis conducted for the vertical half of the 50-ton/h unit opposite the feed pipe revealed that the bearing lubricant will flash as it exits in the outboard end of the main shaft bearing present in this half of the machine.

The lubricant water flashing off at the end of the main shaft bearing should not be detrimental to the performance of the bearing, if the bearing surfaces are protected by a hard facing alloy coating.

A.9.0 RECOMMENDATIONS

- Any thrust bearing or thrust collar arrangement should be located at one end of the main shaft only, to reduce effects of thermal growth of the shaft during operation.
- The chrome oxide coatings on the O.D. and I.D. of the main shaft should overlap the ends of the matching coatings on the respective bearing surfaces, by an amount slightly greater than the rotor-to-housing axial running clearance.
- The theoretical minimum startup pressure for the main shaft bearings in the 50-ton/h unit (which has the greater static bearing loads of the two units) is 45 psig. The minimum startup pressure for the bearings in each coal feeder unit should be verified by test.
- The 50-ton/h outer seal should be component-tested under actual operating conditions. The test configuration should be such that the sudden blowdown condition can be imposed upon the seal.
- The effects of thermal deflection and pressure moment on leakage through the seal should be further investigated. Unbalanced vibration at the seal rotor and contamination of the buffer water being sealed should also be investigated for their effects on the performance of the seal.
- During the bearing and seal tests, particular attention should be paid to the effects of cavitation erosion and wear, since the limitations imposed by these factors on seal life cannot be analytically predicted.

Appendix B

KINETIC EXTRUDER POWER REQUIREMENTS

The total power required to operate the Kinetic Extruder is composed of two parts:

(1) What we term "internal work," that is, the work done on the coal while inside a sprue in accelerating it up to the rotor rim speed and in overcoming friction and pressure forces; and (2) work done to overcome the aerodynamic windage on the exterior surfaces of the rotor, the "external work." The internal work is directly proportional to coal throughput; while the windage is a more or less fixed parasitic loss and only increases with throughput insofar as wheel size increases. In small-scale (e.g., 1 to 5 tons/hr), high-pressure feeder units, the aerodynamic drag loss is, by far, the dominant term. In large units, where the windage is distributed over a large number of tons/hr, it no longer dominates.

● Internal Work

The internal work is calculated as follows. Consider an element of coal initially placed at position r_1 near the axis (the sprue entrance). In time, it will be spun outward and reach position r_2 (the sprue outlet); there it will leave the sprue with the radial velocity U_2 and a tangential velocity $r_2\omega$, where ω = rotor angular velocity.

The radial force balance on the coal element of mass m at the position r is given by:

$$F_r = m \left(\frac{d^2r}{dt^2} - r\omega^2 \right)$$

where F_r represents the net force acting on the coal element in the radial direction due to friction, gas pressure, etc.

First, consider the case without friction or other radial forces, i.e., $F_r = 0$. The equation of motion is thus reduced to:

$$\frac{d^2 r}{dt^2} - r\omega^2 = 0$$

This differential equation has the general solution

$$r(t) = C_1 e^{\omega t} + C_2 e^{-\omega t}$$

Using the following boundary conditions:

$$r(0) = r_1$$

$$\frac{dr(0)}{dt} = 0$$

we obtain

$$r = \frac{r_1}{2} (e^{\omega t} + e^{-\omega t}) = r_1 \cosh(\omega t)$$

$$\frac{dr}{dt} = \frac{r_1 \omega}{2} (e^{\omega t} - e^{-\omega t}) = r_1 \omega \sinh(\omega t)$$

For $r = r_2$

$$\frac{dr}{dt} = U_2 = r_1 \omega \sinh\left(\cosh^{-1} \frac{r_2}{r_1}\right)$$

Since

$$\cosh^{-1} X = \ln 2X - \frac{1}{2} \frac{1}{2X^2} - \frac{1}{2} \cdot \frac{3}{4} \cdot \frac{1}{4X^4} \dots$$

and

$$\frac{r_2}{r_1} \gg 1$$

we have approximately:

$$\cosh^{-1} \left(\frac{r_2}{r_1} \right) = \ln \left(\frac{2 r_2}{r_1} \right)$$

Thus,

$$U_2 \doteq r_1 \omega \sinh \left(\ln \frac{2 r_2}{r_1} \right)$$

and since $\sinh x = \frac{e^x - e^{-x}}{2}$

$$U_2 = \omega \left(r_2 - \frac{r_1^2}{4 r_2} \right)$$

And since $r_2 \gg r_1$, we have simply

$$U_2 \doteq r_2 \omega$$

Thus, in the frictionless case, to a good approximation, the coal mass element leaves the sprue with radial velocity

$$U_2 = r_2 \omega$$

The tangential velocity is equal to the rotor rim velocity

$$V_2 = r_2 \omega$$

The total kinetic energy change, which represents the work done on the element by the rotor, is then given by

$$\begin{aligned}\Delta KE &= \frac{1}{2} m \left(\sqrt{U_2^2 + V_2^2} \right)^2 - \frac{1}{2} m r_1^2 \omega^2 \\ &= m r_2^2 \omega^2 - \frac{1}{2} m r_1^2 \omega^2\end{aligned}$$

Thus, when $r_2 \gg r_1$, $r_2^2 \omega^2$ is the work done per unit mass of coal in the frictionless case.

Now, we can also show that the energy requirement is the same when friction or other retarding forces are present. Consider the special case where the net force constrains the coal elements' motion to a constant velocity U_r . Thence,

$$F_r = m(-r \omega^2)$$

and the work done to overcome friction is

$$-\int_{r_1}^{r_2} F_r dr = \int_{r_1}^{r_2} m r \omega^2 dr = \frac{m}{2} (r_2^2 - r_1^2) \omega^2$$

The work done by the rotor is

Total Work = change in kinetic energy + friction work

$$\begin{aligned}
 &= \frac{1}{2} m \left(\sqrt{r_2^2 \omega^2 + U_r^2} \right)^2 - \frac{1}{2} m \left(\sqrt{r_1^2 \omega^2 + U_r^2} \right)^2 + \frac{1}{2} m (r_2^2 - r_1^2) \omega^2 \\
 &= m (r_2^2 - r_1^2) \omega^2
 \end{aligned}$$

which for $r_2 \gg r_1$ is the same result as the frictionless case. We have worked out other special cases with identical results and the relation apparently holds in the general case.

Therefore, regardless of the details of the internal forces, the total internal work done by the rotor per unit mass of coal always closely approximates $r_2^2 \omega^2$. This result applies whether the retarding forces are frictional or gas-pressure induced. So long as the bulk coal can be considered incompressible, extra PV work terms do not need to be added. The total internal horsepower requirement for the Kinetic Extruder may then be calculated from:

$$HP_I = \frac{0.555}{550} \frac{r_2^2 \omega^2}{g} \dot{m}_c \quad (\text{hp}) \quad (4)$$

where:

$$\begin{aligned}
 r_2 &= \text{rotor radius (ft)} \\
 \dot{m}_c &= \text{coal feed (tons/hr)} \\
 g_c &= \text{gravitational constant (ft/s}^2\text{)}
 \end{aligned}$$

Power requirements computed from the above equation were compared with experimental data obtained in tests with the sprue evaluation rotor. "Internal work" power was determined at atmospheric pressure by subtracting shaft power input without coal flow from the power consumption observed after coal flow was established. These data are given in Fig. B-1 together with the theoretical curve. The data

average about 1 hp/ton/hr higher than the theoretical curve. This small increase is probably due to friction by the coal in the space between the inner stationary hub and the rotating surfaces of the rotor.

- External Work

The drag torque for a thick rotor wetted on both sides is normally expressed as:

$$T = C_m \frac{1}{2} \rho \omega^2 r_2^5 \left(1 + \frac{5}{2} \frac{h}{r_2} \right) \quad (5)$$

and the required power is given by:

$$V = T\omega$$

where

- T = aerodynamic skin drag torque
- V = aerodynamic power dissipation
- ω = rotor speed
- ρ = external gas density
- r_2 = rotor radius
- C_m = dimensionless torque coefficient
- h = rotor rim thickness

The functional form of the thickness correction term $\left(1 + \frac{5}{2} \frac{h}{r} \right)$ is derived based on the assumption of uniform skin friction coefficient over the disk surface.

The torque coefficient C_m is best determined from experimental measurements. If no experimental data is available, one can estimate the torque coefficient from theoretical considerations.

Fig. B-1 Comparison of "Internal Work" Data With Theory

The torque coefficient C_m is a function of the rotor Reynolds number $Re = \frac{\rho \omega r_2^2}{\mu}$ and geometric factors. If the disk is hydraulically smooth, at high Reynold numbers a $Re^{-1/5}$ scaling of C_m is found. For a free disk the old Von Karmen theoretical expression, $C_m = 0.146 Re^{-1/5}$, is often used (Ref. 4). If the disk is running in a tight housing, the drag is expected to be lower. For example, Daily (Ref. 5) gives:

$$C_m = 0.102 \left(\frac{s}{r_2} \right)^{1/10} Re^{-1/5} \quad (6)$$

where

s = housing clearance at sides of disk

The above expressions for C_m apply to rotors with hydraulically smooth surfaces. For rough surfaces, torque coefficients that are somewhat higher than the above expressions, and exhibiting no Reynolds number dependence, would be expected.

Figures B-2 and B-3 show drag data from Theodorsen (Ref. 4) for disks and cylinders of varying roughness. The general shape of the curves for the rough surfaces is quite similar to the appearance of the data obtained in the present rotor drag tests (see Sec. 4.2.3).

One may try to quantitatively estimate the influence of roughness in the present tests from pipe flow data. Figure B-4 shows the classical "Moody diagram" for pipe flow roughness effects, which gives the skin friction coefficient λ as a function of Reynolds number with wall roughness as a parameter. The average λ for the rotor can be shown to be proportional to C_m . The Reynolds number in Fig. B-4 is based on pipe diameter; in order to apply these data to the rotor, a corresponding Reynolds number must be determined for the rotor.

The Reynolds number to be used should be based on the rotor boundary-layer thickness, δ .

$$Re_\delta = \frac{\rho r \omega 2\delta}{\mu}$$

Fig. B-2 Moment coefficient $C_m = \frac{M}{\frac{1}{2} \rho \omega^2 a^3}$ for disks as function of Reynolds number. (Theodensen, 1944.)

Fig. B-3 Effect of surface roughness, or grain size, on the drag coefficient. Saturation density of particles. (Theodensen, 1944.)

The theoretical hydraulically smooth free disk boundary-layer thickness is given by:

$$\delta = 0.53 r Re^{-1/5}$$

Thus,

$$Re_{\delta} = 1.06 \frac{\rho r^2 \omega}{\mu} Re^{-1/5}$$

The analogous range of Re_{δ} Reynolds numbers in the rotor tests may then be calculated.

The relative wall roughness k_s/d to be used in conjunction with Fig. B-4 may be estimated from the following:

$$\frac{k_s}{d} = \frac{rms}{2 \delta}$$

where

rms = profilometer measurement of rotor surface finish

Based on an estimated rms range of 25 μ -in. (polished) to 500 μ -in. (covered with coal dust layer) the regime of the rotor tests falls into the shaded zone in Fig. B-4.

Accordingly, hydraulically rough behavior should be expected.

As indicated above, the drag coefficient should be determined experimentally, if the proper value of the coefficient is of importance. For small coal feeders, where the aerodynamic friction is the predominant factor in determining the power requirement a fairly accurate knowledge of the value of the torque coefficient is of importance. This is why the studies described in Sec. 4.2.3 were conducted.

Fig. B-4 Moody Diagram for Wall Roughness Effects in Pipe Flow

Appendix C
A CHANNEL FLOW ANALYSIS FOR POROUS BEDS
MOVING UNDER HIGH G-FORCES

by
J. W. Meyer

ABSTRACT

An analysis is made of centrifugal flow of compacted porous beds through channels against high gas back-pressure. One-dimensional compressible gas flow solutions are presented for cocurrent and counter-current solids/gas motion through the variable area channels. The physical process analyzed represents the flow through a newly developed centrifugal "pump" for feeding dry pulverized material against a pressure barrier. The objective of the analysis is to gain a basic theoretical understanding of this type of device. Good agreement is found between analytical predictions and test data obtained with an experimental "pump" feeding coal.

* This is the text of a paper which has been submitted to the American Institute of Chemical Engineering Journal for publication

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A CHANNEL FLOW ANALYSIS FOR POROUS BEDS MOVING UNDER HIGH G-FORCES

by

J. W. Meyer

Applied Mechanics Laboratory
Lockheed Palo Alto Research Laboratory
3251 Hanover Street
Palo Alto, California 94304

SCOPE

A new centrifugal "pump" has been recently developed for transporting dry pulverized material across a gas pressure barrier (Bonin et al. 1977). The experimental 1 ton/hr scale machine has successfully pumped coal from atmospheric pressure to delivery pressures as high as 2.8 Mpa (400 psig). The heart of the pump is a high-speed rotor that contains many converging radial channels or "sprues." Pulverized material is transported into the eye of the rotor and is centrifuged outward into the sprues. A compacted moving plug of porous material forms in the sprues and creates a seal against the high-pressure gases. The sealing action is a combined effect of both the motion of the plug and its relatively low permeability.

The objective in the present paper is to develop a theoretical description of the basic operation of this new device, as well as a capability for predicting its performance. Specifically, the pump channel flow is analyzed and solutions to the basic porous media flow problem are presented. A one-dimensional gas flow model is developed representing cocurrent or countercurrent solids/gas motion through the variable area channels at high-pressure ratios. Experimental data obtained in tests of the

experimental machine pumping pulverized coal are presented, and comparisons with the theory are made. Attention here is focused exclusively on the rotor channel flow. Other aspects of the design of practical pump hardware are discussed elsewhere [Bonin et al. 1977, Lockheed (July 1977), Lockheed (December 1977)].

CONCLUSIONS AND SIGNIFICANCE

A theoretical approach has been developed to describe the porous media flow situation in the flow channels of a pulverized material pump. Theoretical model predictions based on uniform bed properties are in good agreement with operational experience with an experimental machine. The apparent permeability of the flowing material in the pump channels is found to be somewhat higher than measured in tests on static material plugs.

The theoretical model can be taken to be sufficiently validated to be useful in designing solids pumping machines to particular sets of requirements.

1.0 CHANNEL FLOW ANALYSIS

The flow of material in the pump rotor sprues has a basic similarity to the packed bed gravity transport of solids through standpipes or solid/fluid contacting columns that are widely used in the chemical industry. Stresses and friction forces in such columns have been investigated previously, using both one-dimensional (Delaplane 1956, Brandt and Johnson 1963) and two-dimensional models (Grossmann 1975). This work has been drawn upon in the present stress analysis.

Consider a tall cylindrical or gently tapered bed of granular material in a radially oriented pump channel, such as illustrated by Figure 1. A control volume

Fig. C-1 Granular Bed Column Stresses

of bed material is subjected to centrifugal g-forces which are opposed by a combination of gas pressure and granular material stresses. Assuming that the material flow velocity remains low, radial and coriolis accelerations can be neglected and the equilibrium force balance may be written as follows:

$$\rho_b G - \frac{dp}{dy} - \frac{d\sigma_y}{dy} - \frac{4\tau}{D} = 0 \quad (1)$$

where σ_y is interpreted as the normal stress in the y-direction averaged across the column cross section.

Following a development similar to that applied in the case of gravity standpipes [Delaplane (1956), Brandt and Johnson (1963)], the stress ratios μ_1 and μ_2 are introduced,

$$\mu_1 = \frac{\tau}{\sigma_r} \quad ; \quad \mu_2 = \frac{\sigma_r}{\sigma_y}$$

where σ_r is the material stress normal to the wall. Using these ratios, Eq. (1) becomes

$$\rho_b G - \frac{dp}{dy} - \frac{d\sigma_y}{dy} - \frac{4\mu_1\mu_2\sigma_y}{D} = 0 \quad (2)$$

If

$$\left(\rho_b G - \frac{dp}{dy} \right), \mu_1, \text{ and } \mu_2$$

are assumed constant the solution of Eq. (2) takes the form

$$\sigma_y = \frac{\left(\rho_b G - \frac{dp}{dy}\right) D}{4 \mu_1 \mu_2} \left[1 - \exp\left(\frac{-4 \mu_1 \mu_2 y}{D}\right) \right] \quad (3)$$

which in the absence of the dp/dy term is the well known Janssen's Equation.

According to Eq. (3), the normal stress has an asymptotic value of

$$\sigma_y(\infty) = \frac{\left(\rho_b G - \frac{dp}{dy}\right) D}{4 \mu_1 \mu_2} \quad (4)$$

for large y .

The coal feedstock used in the present experiments has a wall friction coefficient $\mu_1 \approx 0.5$; however, the exact value for the Janssen constant μ_2 was not determined. Jenike et al. (1972) recommend $\mu_2 = 0.4$ as representative of most materials. Using these data, $\frac{1}{4 \mu_1 \mu_2} = 1.25$, in other words, the asymptotic stress $\sigma_y(\infty)$ represents the hydrostatic head only 1.25 diameters below the bed surface. Wolf and Hohenleiten (1945) have published experimental column load data for pulverized coal which show an asymptotic head of about 2 diameters.

Length-to-diameter ratios in the experimental rotor channels were quite large. The sprues tested were 21.3 cm in length with the channel span converging from 5 cm at the inlet to 1.3 cm at the midpoint, and then to from 0.6 to 0.9 cm at the outlet, depending on the particular set of hardware. Normally, most of the gas pressure drop takes place in the last 10 cm of the sprue where the average diameter is only about 10% greater than the outlet. The conclusion to be drawn is that any build up of σ_y in the sprues is very limited in magnitude and consequently, $d\sigma_y/dy$ is small

throughout most of the flow channel, relative to the other terms in Eq. (1). Dropping this term, the stress equation reduces to

$$\tau \approx \frac{D}{4} \left(\rho_b G - \frac{dp}{dy} \right) \quad (5)$$

This approximation can be expected to hold reasonably well everywhere except close to the inlet end of the sprue.

Evidently, over most of the sprue the force balance is quite simple. In the absence of a pressure gradient, the centrifugal body force and wall friction are in equilibrium. Imposing a gas pressure gradient merely reduces τ (and σ_y). This presumably has little effect until such time that $\frac{dp}{dy} = \rho_b G$ and the bed stresses completely disappear, i.e., the material is fluidized. The introduction of the additional complications present in the actual pump rotor, namely those of moderate variations of body force, pressure gradient, and cross-sectional area along the channel, does not change this situation. Thus, the main requirement for pump channel design is just a proper matching between the body force distribution and the gas pressure gradient distribution in the channel. As long as the body force exceeds the pressure gradient so that the bed remains stressed to some degree, the pump will operate stably. This requirement is virtually local in character because of the short effective distances over which bed stresses can be transferred. Bed stresses could only be important insofar as they may influence the gas flow by changing bed porosity and permeability. Based on the experimental results to be discussed, this effect does not appear to have had a significant impact.

1.1 GAS FLOW SOLUTION

A one-dimensional porous media gas flow analysis may be applied to the sprue flow. The theoretical model is formulated to handle gas bleed flows injected through the sprue walls into the medium, in addition to gas permeating upstream from the high pressure region. However, bleed flows were not used in the experiments discussed in the present paper. Using the nomenclature shown in Figure 2, where the origin is now fixed at the sprue outlet, the following equations apply to the problem. The Darcy equation is:

$$v + \epsilon u = - \frac{k}{\mu} \frac{dp}{dz} \quad (6)$$

the gas and coal continuity equations are:

$$\begin{aligned} -\rho q + \frac{\partial (\rho v a)}{\partial z} &= -\epsilon \frac{\partial (\rho a)}{\partial t} \\ \rho_s u a &= \dot{m} = \text{constant} \end{aligned} \quad (7)$$

and the gas equation of state under isothermal conditions is:

$$\frac{p}{\rho} = \frac{p_1}{\rho_1} = \text{constant} \quad (8)$$

An unsteady form of the gas continuity equation is used here because the final equations are solved by means of a time asymptotic relaxation method. The bed parameters of bulk density, porosity, and permeability, as well as the gas viscosity

Fig. C-2 Schematic of Sprue Flow Situation Modeled

are normally assumed to be constants in a given calculation. Introducing the following nondimensional variables of

$$A = \frac{a}{a_2} \quad , \quad P = \frac{p}{p_1} \quad , \quad Z = \frac{z}{L}$$

and defining

$$\beta \equiv - \frac{k p_1}{\mu L}$$

yields

$$v + \frac{\epsilon \dot{m}}{\rho_s a} = \beta \frac{dP}{dZ} \quad (9)$$

$$- \frac{\rho q}{a_2} + \frac{\partial (\rho v A)}{L \partial Z} = - \epsilon \frac{\partial (\rho A)}{\partial t} \quad (10)$$

$$\rho = \rho_1 P \quad (11)$$

Combining Eqs. (9), (10), and (11), a single, second-order differential equation for the pressure ratio is derived and is

$$\begin{aligned} - \epsilon A L \frac{\partial P}{\partial t} &= \beta A P \frac{\partial^2 P}{\partial Z^2} + \beta A \left(\frac{\partial P}{\partial Z} \right)^2 \\ &+ \left[\beta \left(\frac{dA}{dZ} \right) P - \frac{\epsilon \dot{m}}{\rho_s a_2} \right] \frac{\partial P}{\partial Z} - \frac{P q L}{a_2} \end{aligned} \quad (12)$$

The boundary conditions are

$$P(0) = P_r \quad , \quad P(1) = 1$$

In Eq. (12), q is the injected bleed gas per unit length of sprue. The bleed gas is modeled by a simple incompressible orifice flow equation.

$$q_i = \frac{1}{\Delta Z_B^i} 0.6 a_B^i \left[\frac{2 (p_P^i - p_1 P)}{\rho_P^i} \right]^{1/2} \quad \text{for} \quad Z_B^i \leq Z \leq Z_B^i + \Delta Z_B^i \quad (13)$$

where q_i is the volumetric flow per unit sprue length; Z_B^i is the bleed port location; ΔZ_B^i is the length over which bleed gas injection takes place; a_B^i is the i th bleed gas orifice area; and p_P^i , ρ_P^i is the plenum pressure and density of the bleed gas.

The time asymptotic solution of Eqs. (12) and (13) constitutes the steady-state sprue gas-flow solution for a given problem. The computer code used to implement the solution uses a Z mesh of 129 points. An arbitrary (typically linear) pressure distribution is used as initial conditions and the equations are integrated forward in time until a steady state is reached. The final solution yields the pressure distribution along the sprue, the pressure gradient distribution along the sprue, the bleed gas flow rates (if any), and the gas leakage rate through the sprue.

Examples of results for a typical set of calculations without bleeds are given in Figures 3 through 6. The following set of constant parameters is assumed for the example: sprue length = 21.3 cm; solids bed permeability = $2.6 \times 10^{-9} \text{ cm}^2$; solids bed density = 0.67 gm/cm^3 ; solids bed porosity = 0.53; gas viscosity = $1.8 \times 10^{-4} \text{ gm/cm-s}$; sprue inlet gas pressure = 0.1 MPa (1 atm); and sprue outlet gas pressure = 2.8 MPa. Figure 3 gives the assumed sprue shape in terms of (area ratio) $^{1/2}$ as a function of position. Figures 4 and 5 give the calculated pressures and pressure gradients throughout the sprue as a function of position. The solids mass flux at the sprue outlet (i. e., $\dot{m}/a_2$) is the parameter which distinguishes the several curves shown in each graph. Finally, Figure 6 gives the gas flow through the sprue in terms

Fig. C-3 Sprue Shape Profile for Example Calculation (Area Ratio)^{1/2}
Versus Position

Fig. C-4 Sprue Gas Pressure as Function of Position for the Example.
Sprue Outlet Solids Mass Flux as a Parameter

Fig. C-5 Sprue Gas Pressure Gradient as a Function of Position for the Example. Sprue Outlet Mass Flux as a Parameter

Fig. C-6 Sprue Inlet Gas Velocity as a Function of Outlet Solids Mass Flux for the Example

of the superficial gas velocity at the sprue inlet ($Z = 1$ position) as a function of solids mass flux. It may be noted that there is a particular value of mass flux, in the example it is about 550 kg/s/m^2 , for which there is no gas flow whatsoever through the sprue ($v_1 = 0$). This is a desirable operating point to aim for in designing a pump.

It is clear from the plots in Figures 4 and 5 that the pressure distribution is sensitive to the mass flux, especially beyond the $v_1 = 0$ operating point. High mass flux tends to concentrate the pressure drop toward the outlet of the sprue. It is also found that variation of the medium permeability at constant mass flux produces similar looking families of pressure curves, and actually it is the ratio of mass flux to permeability that is the key factor. High mass flux/low permeability situations yield highly peaked pressure gradient distributions which are a poor match to the body force.

As discussed, for a pump to operate, the centrifugal body force in the medium must exceed the gas pressure gradient. In terms of the dimensionless variables, this requirement may be stated as:

$$\left| \frac{dP}{dZ} \right| < \left| \frac{\rho_b r \omega^2 L}{p_1} \right| \quad (14)$$

The body force is linear in r (and Z) if the bed density in the sprue is constant.

In the experimental rotor, an essential component is a short endhopper or "control orifice" placed down-stream of the sprue outlet as shown in Figure 7. This device acts to stabilize and to meter the material velocity through the sprue. Pressure equalization ports assure that there is no significant pressure difference across the

Fig. C-7. Control Orifice Configuration

control orifice itself. The flow of solids through the control orifice has been found to obey a Rausch-type gravity flow equation (Zenz 1960), scaled by a $G^{1/2}$ factor to account for higher G-forces, i.e.,

$$\begin{aligned}\dot{m}_h &= C d^{5/2} G^{1/2} \\ &= C d^{5/2} (r)^{1/2} \omega\end{aligned}\tag{15}$$

where C is an empirical constant and d is the orifice outlet diameter.

Thus, up to a certain limiting pressure, the endhopper runs full and the throughput is independent of delivery pressure, being only a function of rotor speed as given by Eq. (15). Under conditions where the centrifugal force is insufficient according to Eq. (14) criteria, and the maximum mass flow which can be delivered by the sprue is less than $\dot{m}_h$, the end hopper would not run full. If this occurs, it has been found experimentally that the material plug in the sprue is unstable, and can result in flow stoppages or "blowbacks" of high pressure gases into the rotor due to a complete loss of the integrity of the sprue plug.

1.2 EFFECT OF GAS BLEEDS

One set of calculated results is presented in Figure 8 to generally illustrate the effect of gas bleeds distributed along the sprue wells. Four bleeds are assumed, all from a common plenum having a pressure of 3.62 MPa. All bleed orifices are assumed to be $1.9 \times 10^{-7} \text{ m}^2$ in area and a sprue outlet area of $4.9 \times 10^{-4} \text{ m}^2$ is specified. Other data are identical to the previous example without bleeds.

Comparing Figure 8 and Figure 5, it can be seen that the effect of the gas injection is to reduce the pressure gradient in the region where gas is introduced, as would be expected.

1.3 MATERIAL PROPERTIES

The bulk material properties needed for theoretical calculations are the bulk density, porosity, and permeability. Porosity is directly related to the bulk density by $\epsilon = (\rho_T - \rho_s)/\rho_T$ where ρ_T = true material density. Therefore, only the bulk density and permeability at the state of compaction of the sprue material need to be estimated.

A laboratory apparatus was assembled to measure permeability of a compacted column of material. The schematic of the setup is shown in Figure 9. To measure permeability, the material sample is placed in the sample tube and compacted in a laboratory centrifuge. The sample tube is then placed in the Figure 9 apparatus and the flow rate of gas permeating through the column is measured at a given pressure drop. The pressure drops are kept small so that the permeability may be computed from an incompressible form of the Darcy equation. Specifically:

$$k = \frac{Q_s \mu L_s}{A_s \Delta p_s}$$

where Q_s is the volume flow rate through sample, L_s is the material column length, Δp_s is the pressure drop across column, and A_s is the column cross-sectional area. The sample tubes are 2.2 cm in diameter and a typical compacted column length was 5 to 6 cm.

Fig. C-8 Pressure Gradient Distributions Obtained with Gas Bleeds Through the Sprue Wall at $Z = 0.1, 0.2, 0.3$, and 0.4

Fig. C-9. Permeability Test Apparatus Schematic

Figure 10 shows permeability data obtained by the above procedure as a function of compacting G-force in the centrifuge. The test materials are pulverized coal samples taken in conjunction with the coal pump experimental runs discussed in the next sections.

It is not altogether clear how to apply the Figure 10 data to the sprue flow situation. One approach would be assume the validity of Eq. (4)

$$\sigma_y = \frac{\left(\rho_s G - \frac{dp}{dy} \right) D}{4 \mu_1 \mu_2}$$

and assume k is directly related to σ_y . Thus, in the absence of a pressure gradient, $k = f(\rho_s G D)$. The average G-force in the sprues in the experiments described in the following section was approximately 3000 (g), which is 3 times higher than that which could be obtained in the lab centrifuge. On the other hand, over much of their length, the sprue diameters are 2 to 4 times smaller than the centrifuge tubes. Thus, using the $\rho_s G D$ scaling, the sprue material permeability would be expected to be similar to that near the 1000-g compaction level in the lab tests. Finally, if the sprue pressure gradient represents a significant fraction of the sprue body force, a proportional reduction in the effective G-force could be made.

2.0 EXPERIMENTAL HARDWARE

The coal pump hardware and test loop have been described elsewhere (Bonin et al. 1977) and will be only briefly reviewed here. The installation is shown schematically in Figure 11. The essential items are the atmospheric feed hopper, a feed

Fig. C-10. Coal Bulk Density and Permeability Measured in Lab Centrifuge Tests. 2.2 cm ID Test Tubes

Fig. C-11. Experimental Test Loop Schematic

tube for transporting fluidized material into the eye of the rotor, the rotor and drive system, a vent line and an eductor system for removing excess gases from the eye of the rotor, and, finally, a high-pressure receiver hopper. The feed hopper is zone fluidized near the feed tube inlet.

Tests of the coal pump at high delivery pressure are conducted on a batch basis. The pump cannot be started with a significant pressure difference across the sprue channel. Coal flow must be first initiated while the receiver hopper is at atmospheric pressure, and once the sprue plug is established, the hopper may be pressurized to the test level. To shut down, the procedure is reversed. Each of the vessels could hold approximately one-half ton of coal. However, allowing for ullage and for time for pressurization and depressurization, steady-state run times were normally about 5 to 10 minutes in duration.

The prototype rotor tested was 71.1 cm in diameter and contained 12 sprues which were replaceable parts so that shapes and cross-sectional areas could be varied. A rotor subassembly is shown in Figure 12. The sprues are constructed in two approximately equal-length sections -- a cast funnel shape in conjunction with a machined cylinder. The wall angles in both sections are everywhere less than the material angle of slide so that a uniform flow is obtained. In the experiments discussed in this paper, the funnel sections remained the same and the sprue area profile changes were made only in the machined section.

3.0 EXPERIMENTAL RESULTS WITH PULVERIZED COAL

Complete validation of the computer analysis for the sprue flow is a difficult task. Definitive experimental data, for example, the sprue pressure distributions,

Fig. C-12. Experimental Rotor Partial Assembly

cannot be obtained practically. Instead, indirect criteria were relied on, in this case gas back leakage through the sprues and maximum delivery pressure capability. The following pertinent data were obtained during experimental runs: (1) coal throughout, (2) delivery pressure, (3) rotor speed, (4) rotor suction line flowrates. Throughputs are obtained by monitoring the weight change of the feed hopper, which is mounted on load cells.

Rotor suction line flowrates give a measure of the gas permeation flowrate through the sprue. In the experimental unit, excess gases are vented from the eye of the rotor through from one to three 0.46 cm ID channels. These vent lines are connected to a small eductor that discharges back into the feed hopper. Typically, the pressure at the eye of the rotor is kept at 0.07 to 0.09 MPa (-2 to -5 psig) to assure a constant flow into the rotor from the atmospheric pressure feed hopper. The minimum venting requirement is to remove the gas produced by compacting the coal from a fluid bed density, at which it enters the rotor, to the compacted bed density which it assumes in the sprues. This flow may be estimated from a gas mass balance on the rotor eye as follows: Coal enters the rotor eye at a fluidized density of approximately 400 kg/m^3 (72% porosity) and leaves at approximately 710 kg/m^3 (50% porosity). At a typical flowrate of 1000 kg/hr, the minimum venting requirement is thus:

$$Q_{\min} = [1000 (0.72)/400] - [1000 (0.50)/710] = 1.1 \text{ m}^3/\text{hr}$$

Similarly, at the zero leakage point, the sprue gas velocity (v in Eq. 6) is zero and all the gas entering through the feed pipe must be removed through the vent lines. In this case, $Q = 1.8 \text{ m}^3/\text{hr}$.

The vent lines were calibrated so that flowrates could be computed from the pressure drop between the eye of the rotor and the eductor suction inlet. To properly correct for variations in gas density at the vent line inlet, the following pipe pressure drop equation was fitted to the line calibration data and was used to compute flows

$$Q = C (P_1 - P_2)^{1/2} (P_1)^{1/2}$$

where

C = calibration constant = $0.97 \text{ nm}^3/\text{hr}/\text{MPa}^{1/2}$

Q = vent line flow (nm^3/hr)

P_1 = rotor eye pressure (MPa)

P_2 = eductor inlet pressure (MPa)

It must be noted that the calibration was made using clean gas whereas in coal pump operation a small loading of coal dust is entrained in the vent flows. The effect is believed to be small; however, strictly speaking, the apparent flow rates in tests must be treated as upper bounds.

Experimental results obtained in a particular series of four successive coal pump runs, using the same well-characterized batch of coal, comprises the data set that will be used here to compare with theory. The feedstock was a Utah bituminous coal with a nominal 70% passing 200 mesh grind and the pressurizing gas was nitrogen. All the runs were made at close to a one ton/hr rotor throughput and similar rotor speeds. Three sprue outlet sizes were tested. The outlet area of the largest size was 2.23 times that of the smallest, yielding a similar spread in the outlet velocity of

the coal. Figure 13 shows the shape profiles of the sprues. Table C-1 summarizes the experimental data for the series of tests. Estimates for the experimental error in the data are as follows: delivery pressure $\pm 2\%$, mass flux $\pm 5\%$, rotor speed $\pm 1\%$, vent flowrate $\pm 15\%$.

In Run 129 the largest set of sprues was used (outlet diameter $D_2 = 0.878$ cm). At the observed throughput, the corresponding coal mass flux at the sprue outlet was 358 kg/s/m^2 and a delivery pressure of 1.79 MPa was reached. At this point the total indicated rotor vent flow was $4.2 \text{ nm}^3/\text{hr}$. In Run 130, a much smaller sprue set was installed ($D_2 = 0.587$ cm) which yielded an outlet mass flux of 805 kg/s/m^2 . The rotor vent flow was then minimal and the pressure limit was formed by the sprue pressure gradient exceeding the centrifugal force. As discussed, this has been found to yield unstable flow behavior and, in this case, a blowback.

In Runs 131 and 132, the midsize sprue set was installed ($D_2 = 0.719$ cm) and a third rotor suction line was activated to increase venting capacity. This enabled pressure levels of about 2.8 MPa to be reached.

4.0 COMPARISON WITH THE ONE-DIMENSIONAL THEORY

In this section comparisons between the experimental data for Runs 129 through 132 and predictions of the one-dimensional theory are made. It was found by trial and error that a single set of bed properties gives calculated results matching the data for all three sprue sizes.

To analyze Runs 129 and 130, the theoretical model was exercised for the test pressures and mass fluxes with coal properties of $\rho_s = 0.71 \text{ gm/cm}^3$, $\epsilon = 0.507$ and a range of permeabilities: $k = 3.2 \times 10^{-9}$, 3.7×10^{-9} , and $4.3 \times 10^{-9} \text{ cm}^2$.

Fig. C-13. Sprue Shape Profiles Used in the Experimental Runs

Table C-1
COAL PUMP TEST DATA USED IN COMPUTER MODEL VALIDATION

Run No.	Sprue Outlet Diameter (cm)	Delivery Pressure (MPa)	Rotor Speed (rads/s)	Sprue Outlet Mass flux (kg/s/m ²)	Observed Rotor Vent Flow (nm ³ /hr)	Run Duration at Full Pressure (min)	Comment
129	0.876	1.79	370	358	4.2	5	2 vent lines, pressure limit due to excessive gas backflow
130	0.587	2.23	376	805	Small	1	dP/dZ limiting behavior, - unstable/ blowback
131	0.719	2.79	376	537	5.6	6	3 vent lines, stable operation
132	0.719	2.82	376	537	6.5	7	3 vent lines, stable operation

This range spans what might represent conceivable states of compaction, based on the permeability tests on samples taken after each run. (See Figure 10.) The bed density value used is compatible with what is later shown to be the best-fit permeability. A measurement of the true material density of the coal was made using a gas pycnometer, yielding $\rho_T = 1.44 \text{ gm/cm}^3 \pm 1\%$. The porosity of $\epsilon = 0.507$ corresponds to this value with $\rho_s = 0.71 \text{ gm/cm}^3$.

Figure 14 illustrates the results for the Run 129 conditions. The graph shows the theoretical gas backflow through the sprues as a function of coal permeability. The measured vent flow for Run 129 was $4.2 \text{ nm}^3/\text{hr}$, which would correspond to the theoretical quantity if $k = 4.1 \times 10^{-9} \text{ cm}^2$. Since the measured vent flow must be interpreted as an upper limit, the apparent sprue medium permeability also constitutes an upper bound, i.e., $k \text{ (apparent)} < 4.1 \times 10^{-9} \text{ cm}^2$.

Figure 15 shows the theoretical pressure gradient distributions for the Run 129 case with $k = 3.6 \times 10^{-9} \text{ cm}^2$, which is later shown to be most representative permeability. The centrifugal body force in comparable units also is plotted, showing that it greatly exceeds the pressure gradient throughout the sprue.

Figure 16 illustrates the key theoretical results for the Run 130 conditions. This figure gives a comparison of the centrifugal body force with the theoretical pressure gradient distributions obtained for the three k values. A blowback occurred in the test which is interpreted as dP/dZ having approached the body force too closely. According to the theory, the outlet dP/dZ would match the body force if $k \approx 3.1 \times 10^{-9} \text{ cm}^2$. This should be treated as a lower bound for the permeability in the sprue.

Fig. C-14. Predicted Gas Backflow, as a Function of Bed Permeability for Run 129.

Fig. C-15. Theoretical Sprue Pressure Gradient Distribution
For Run 129. $k = 3.6 \times 10^{-9} \text{ cm}^2$

Fig. C-16. Predictions for Run 130 Test. Pressure Gradients for Three Values of Bed Permeability Computed With Body Force

Therefore, it appears that if the theory and data are to be reasonably consistent for both sprue sizes, the actual sprue coal permeability must lie in the range

$$3.1 \times 10^{-9} \text{ cm}^2 < k \text{ (apparent)} < 4.1 \times 10^{-9} \text{ cm}^2$$

The theoretical model was then exercised for the Run 131 and 132 conditions (2.8 MPa delivery pressure), with the permeability in the middle of the apparent range, $k = 3.6 \times 10^{-9} \text{ cm}^2$. These results are given in Figures 17 and 18. Figure 17 illustrates the gas pressure gradient distributions for a range of mass fluxes as compared to the body force in the run. For the test mass flux, the body force everywhere exceeds the calculated dP/dZ curve except for a very small region near $Z = 0.4$. The region is only about one half a diameter in length so that it is plausible that a weak normal stress gradient could have forced the coal through this area. Figure 18 shows the comparison of theoretical and experimental gas backflow for the two runs. In this figure, the theoretical vent flow profiles are shown as a function of pressure and sprue mass flux. Measured vent flows in both runs are only slightly higher than the theoretical values.

A comparison of data and theoretical predictions for the runs for all three sprue sizes with $k = 3.6 \times 10^{-9} \text{ cm}^2$ are summarized in Table C-2. Overall, the simple one-dimensional model calculations with uniform bed properties seem to be in good agreement with the observed test results. As can be seen, the high gas flow at low back pressure for the large sprue is predicted by theory, the blowback with the small sprue is predicted, and finally the successful operation and leakage rates at 2.8 MPa, using the optimal sprues, is predicted reasonably well.

Fig. C-17. Predicted Gas Pressure Gradient Distributions Compared With Body Force, $k = 3.6 \times 10^{-9} \text{ cm}^2$, Run 131/132 Sprues

Fig. C-18. Comparison of Theoretical Vent Flows With Observed Flows In Runs 131, 132. Predictions for $k = 3.6 \times 10^{-9} \text{ cm}^2$

Table C-2

THEORETICAL MODEL PREDICTIONS FOR $k = 3.6 \times 10^{-9} \text{ cm}$, $\rho_b = 0.71 \text{ gm/cm}^3$, $\epsilon = 0.507$,
COMPARED WITH TEST RESULTS

Run No.	Delivery Pressure (MPa)	Sprue Outlet Mass flux (kg/s/m ²)	Theoretical Prediction	Experimental Results
Run 129	1.79	358	Vent flow of 3.3 nm ³ /hr, ample body force	Vent flow of 4.2 nm ³ /hr stable operation
Run 130	2.23	805	No gas backflow, outlet dP/dZ only 15% less than body force. Danger of blowback if slight decrease in permeability	No gas backflow, operated for short period then sudden blowback
Run 131	2.79	537	Body force 30% greater than dP/dZ at outlet but just below dP/dZ at Z = 0.4, 5.2 nm ³ /hr vent flow	Smooth runs with no blowback. Vent flows of 5.6 and 6.5 nm ³ /hr for Runs 131, 132 respectively
Run 132	2.86	537	Same as above, 5.4 nm ³ /hr vent flow	

It should be noted that the sprue material permeability value that provides a good fit to the data, $k = 3.6 \times 10^{-9} \text{ cm}^2$, is somewhat higher than might be expected based on the lab centrifuge measurements. The reduction in compaction stress due to the partial support of the sprue material by the gas pressure gradient does not appear to be nearly enough to account for the increase. This is especially clear in Run 129 which utilized the largest diameter sprues and the largest excess of body force over the pressure term. Relatively speaking, the compaction of the coal should have been the greatest, and the apparent permeability the least, in this case. However, no such effect can be discerned in the data. It may be speculated that the high apparent permeability is mainly due to the fact that the lab tests were performed on stationary compacted plugs, whereas the material remains in motion in the coal pump sprues. The dilatation associated with the motion reduces the compaction density to less than that obtained in a static condition and washes out the effect of stress level variations.

ACKNOWLEDGMENT

This research was supported by the U.S. Department of Energy under Contract No. E(49-18) - 1792 and program manager R. R. Fleischbein. The author also wishes to acknowledge the contribution of J. J. Kohfeld who ably programmed the computer code.

NOMENCLATURE

a	=	channel area
A	=	area ratio a/a_2
d	=	control orifice outlet diameter
D	=	channel diameter
G	=	$r\omega^2$ centrifugal acceleration
k	=	bed permeability
L	=	sprue length
$\dot{m}$	=	solids mass flowrate
p	=	gas pressure
P	=	pressure ratio p/p_1
Pr	=	total pressure ratio across sprue
q	=	bleed volume flowrate per unit sprue length
Q	=	rotor vent gas flowrate
r	=	distance from rotational axis
t	=	time
u	=	solids radial velocity
v	=	gas superficial radial velocity
y	=	depth below bed surface
z	=	distance from sprue outlet
Z	=	z/L

β = $-k p_1 / \mu L$
 ϵ = bed porosity
 ρ_T = true material density
 ρ_b = total bed density
 ρ_s = bed solids density
 ρ = gas density
 μ = gas viscosity
 μ_1, μ_2 = Jannsen stress ratios
 τ = wall shear stress
 σ_r = material stress normal to wall
 σ_y = average material stress in y-direction

Subscript

1 sprue inlet
 2 sprue outlet

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Appendix D
DETAILS OF THE KINETIC EXTRUDER MODEL NO. 4

This appendix contains all drawings of parts and assemblies used for the manufacturing of the Model 4 feeder. Figure D-1 shows the drawing breakdown structure. All drawings called out in Fig. D-1 are included in this appendix.

1 UNBRAM0 OR EQUIVALENT.
2 SMIA CORP. MODEL BR-10
FIBER OPTIC SENSOR SYSTEM.
3 PARKER HANNIFIN CO. OR EQUIVALENT.

- (1) REMOVE ONE (9537910) NOZZLE BLOCK FROM THE (1) 9: ROTOR ASSY AND INSTALL LOWER OMS NO. 999991 Rotor LIFTING EYE.
- (2) LOWER THE ROTOR ASSY INTO THE (1) 2: HOUSING. APPROXIMATELY CENTER AS SHOWN.
- (3) INSERT THE (1) 2: VAL. IMPT. BEARING ASSY AND THE (1) 2: 9: POWER IMPT. BEARING ASSY INTO THE HOUSING. INSURE THE 1/2" SHOULDER SCREW LOCKWING IS OBTAINED.
- (4) INSTALL ITEM 10.
- (5) CENTER THE ROTOR IN THE HOUSING AND DETERMINE GEAR REQUIRED BETWEEN ITEMS 7 & 9 W. T. G.
- (6) INSTALL ITEMS 7 AND 12 AND TURN TO 50 TO 10-LB.
- (7) REINSTALL THE (9537910) NOZZLE BLOCK.
- (8) INSTALL ITEMS 9, 9, 10 AND 11. TURN TO 200 TO 10-LB.

[illegible]

U.S. Department of Energy

[illegible]

55335-1
DE5722

28
2CF

SECTION A - A
SCALE 1/4" = 1"

APPROVED: [Signature] DATE: 10/1/55	
COAL FEEDER ASSEMBLY	
E 10771 55335-1	1
FILE NO. 4001-42-11	

NOTES:

1. ALL MACHINE SURFACES TO $\sqrt{125}$ OR BETTER AND NOTED
2. ALL PLATE MATERIAL ASTM A70.
3. DO NOT PAINT INTERNAL OR EXTERNAL MACHINED FLANGES.

-3 RETAINER
MATERIAL: ASTM A70 STL
SCALE: 1/2

-1 COVER
MATERIAL: ASTM A70 STL
SCALE: 1/4

REV	DATE	DESCRIPTION	BY	CHKD
1		REVISION		
2		REVISION		
3		REVISION		
4		REVISION		
5		REVISION		
6		REVISION		
7		REVISION		
8		REVISION		
9		REVISION		
10		REVISION		

U.S. DEPARTMENT OF ENERGY

INTERPRET AND PER

DESIGNER: []

ENGINEER: []

DRWING: []

DATE: []

PROJECT: []

COVER, RETAINER

FOR HORIZONTAL FEEDER

FILE NO: 4001.92-114

1. ALL MACHINE SURFACES TO BE Oiled ✓
2. ALL PLATE MATERIAL AS PER A70.
3. DO NOT PAINT INTERNAL OR EXTERNAL MACHINED FLANGES

[illegible]

U.S. DEPARTMENT OF COMMERCE

[illegible]

1000

NOTES:

1. PAINT PART NO. AND REV LTR TO WHICH REF IN APPROX LOC SHOWN, USING WHITE ACRYLIC ENAMEL.
2. UNLESS OTHERWISE SHOWN, ALL MACHINING SURFACE ROUGHNESS SHALL BE 125 RMS, OR BETTER.
3. ALL WELDS SHALL BE LEAK PROOF.
4. WELD PER BEST SHOP PRACTICE.
5. NO STRESS RELIEF REQUIRED.
6. HYDROSTATIC PROOF TEST TO A MIN OF 650 PSIG.
7. 100% CLEAN INTERNAL AND EXTERNAL SURFACES PRIOR TO NOTE 8.
8. PAINT EXTERNAL SURFACE ONLY, USING FLAT BLACK ACRYLIC PRIMER, EXCEPT AS NOTED.
9. DO NOT PAINT SURFACE SHOWN.
10. RECEIVER VESSEL WEIGHTS:
EMPTY: 1663 LB
FULL OF WATER: 1787 LB.
11. MAKE PROOF OF BLIND R.F. PIPE FLANGE: 600 LB, ASTM A70.

DETAIL ITEM 1 (-901)

SECTION A - A

ITEM	DESCRIPTION	STUD	STEEL	CHARL. Y.	REV
1	FLANGE	2" PLT STL	ASTM A70	1	1
2	FACE PLATE	2" PLT STL	ASTM A70	1	1
3	SIDE PLATE	2" PLT STL	ASTM A70	1	1
4	RING	2" PLT STL	ASTM A70	1	1
5	BASE	2" PLT STL	ASTM A70	1	1
6	WELDMENT	2" PLT STL	ASTM A70	1	1

U.S. DEPARTMENT OF ENERGY

WELDON-RECEIVER VESSEL

6537550

DATE 22 JUL 73

BY DANIEL

REVISION 1

REVISION 2

REVISION 3

REVISION 4

REVISION 5

REVISION 6

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REVISION 8

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REVISION 98

REVISION 99

REVISION 100

1. ALL MACHINE SURFACES TO ¹²⁵✓
OR BETTER AND NOTED

2. BREAK ALL EDGES AND ROUGH CORNERS

3. APPLY NITRIDED GAS SALT BATH
DIFFUSION COATING .0007 MIN
THICK TO OBTAIN A RADIAL
CLEARANCE OF .0015/.0020 I.D.
OF 5533525

[illegible]

FILE ID: ACD1.92-14

NOTES:

1	1/2-18 X 1/2-18	CAP SCREW				10
2	5533962	WASHER				12
3	2-122	O-RING	PARKER	BUNA-N		11
4	2-251	O-RING	PARKER	BUNA-N		10
5	2-251	O-RING	PARKER	BUNA-N		9
6	1/8-18 X 1/8-18	CAP SCREW				8
7	US 900	WASHER	SHALLEY			7
8	5533930	HOLD DOWN				6
9	5533930	BEARING ASSY				5
10	5533937	SHAFT ASSY				4
11	5533926	SHAFT				3
12	5533930	BEARING ASSY				2
13	5533930	BEARING ASSY				1

PREPARED FOR
U.S. DEPARTMENT OF ENERGY

INTERVIEW DATE	DATE OF INTERVIEW	BY	FOR
10/10/77	10/10/77	W. DANIEL	W. DANIEL
10/10/77	10/10/77	W. DANIEL	W. DANIEL
10/10/77	10/10/77	W. DANIEL	W. DANIEL
10/10/77	10/10/77	W. DANIEL	W. DANIEL
10/10/77	10/10/77	W. DANIEL	W. DANIEL
10/10/77	10/10/77	W. DANIEL	W. DANIEL
10/10/77	10/10/77	W. DANIEL	W. DANIEL
10/10/77	10/10/77	W. DANIEL	W. DANIEL
10/10/77	10/10/77	W. DANIEL	W. DANIEL

BEARING ASSY
FEED SIDE

10/10/77 5533936

FILE ID: RDD-92-14

D-10

NOTES:

- 1 REAR SEAL DIVISION
BY GLEN A.E.
WHEELING, ILL. 60090

DETAIL D
SCALE 2/1

SECTION A - A

ITEM	DESCRIPTION	QUANTITY	UNIT	REMARKS
1	TERMOUCOUPLE	1		
2	2-540	1		
3	2-540	1		
4	2-540	1		
5	2-540	1		
6	2-540	1		
7	2-540	1		
8	2-540	1		
9	2-540	1		
10	2-540	1		
11	2-540	1		
12	2-540	1		
13	2-540	1		
14	2-540	1		
15	2-540	1		
16	2-540	1		
17	2-540	1		
18	2-540	1		
19	2-540	1		
20	2-540	1		

U.S. DEPARTMENT OF ENERGY

SHAFT ASSY
COAL INPT
COAL FEEDER

DATE: 10/1/70
BY: J. DANIEL
CHECKED: 10/1/70
BY: J. DANIEL

FILE NO: 400-12-11

1. ALL MACHINE SURFACES TO ¹²⁹✓
OR BETTER AND NOTED

2. BREAK ALL EDGES AND ROUGH CORNERS

1 MAY BE PURCHASED FROM HYERSON
ENERGYVILLE, CA.

2 PLF OUC. 9433916 APPLY NITRIDED
GAS SALT BATH DIFFUSION COATING
.0007 INH THICK TO OBTAIN A
RADIAL CLEARANCE OF .00015-.00020
WITH THE I. O. OF 9433925

DETAIL D
SCALE 2/1

DETAIL E

SCALE 0/1
TYP (2) PLACES

[illegible]

FILE ID: A001,52-1

NOTES:

SECTION A - A

[illegible]

1 HERNANDO SEAL DIVISION
674 GLENN AVE
CHERLING, ILL. 60090

[illegible]

PREPARED FOR
U.S. DEPARTMENT OF ENERGY

[illegible]

FD-302 (Rev. 11-27-70)

D-14

1 TAP 100-9200F-00 1 .975 DEEP
0 PLACES
TO RATE WITH 20-9750-9

DETAIL 1
SCALE: 7:1

		-001	SWIFT	TON CRES	AMS 5670		1
QTY BLDG	CODE IDENT	PART CO IDENTIFYING NO	NAME, A, B, C, D S, C, R, E, P, T	ITEM NO, C, O, D, E P, O, S, T, O, F, I, C, E	ITEM NO, C, O, D, E S, P, E, C, I, F, I, C, A, T, I, O, N	CODE	ITEM NO

PREPARED FOR

U.S. DEPARTMENT OF ENERGY

[illegible]

FILE ID: 62-92-14

D-15

8 7 6 5 4 3 2 1

REVISED: 11/11/79

DATE: 11/11/79

BY: J. G. WILKINS

1. BREAK ALL SHARP EDGES

SECTION A-A

DETAIL B

DETAIL C

ITEM	QTY	DESCRIPTION	UNIT
1	1	HUB	PC
2	1	SHAFT	PC
3	1	KEY	PC
4	1	KEYWAY	PC
5	1	KEYWAY	PC
6	1	KEYWAY	PC
7	1	KEYWAY	PC
8	1	KEYWAY	PC
9	1	KEYWAY	PC
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11	1	KEYWAY	PC
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97	1	KEYWAY	PC
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99	1	KEYWAY	PC
100	1	KEYWAY	PC

U.S. DEPARTMENT OF ENERGY

LOCKHEED MISSILES & SPACE COMPANY, INC.

47077 5533526

FILE ID: R001.52-14

FILE ID: A001,52-15

D-17

NOTES:

1	10-2000-1	CAP SCREW			
2	10-2000-2	CAP SCREW			
3	2-200	O-RING	PARCEL	13,440-0	
4	1-4000-10	SEAL ASSY			
5	5533531	SPACER BEARING			
6	5533532	RETAINER SEAL			
7	5533533-1	O-RING			
8	5533534	O-RING			
9	5533535	O-RING			
10	5533536	O-RING			
11	5533537	O-RING			
12	5533538	O-RING			
13	5533539	O-RING			
14	5533540	O-RING			
15	5533541	O-RING			
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94	5533620	O-RING			
95	5533621	O-RING			
96	5533622	O-RING			
97	5533623	O-RING			
98	5533624	O-RING			
99	5533625	O-RING			
100	5533626	O-RING			
101	5533627	O-RING			
102	5533628	O-RING			
103	5533629	O-RING			
104	5533630	O-RING			
105	5533631	O-RING			
106	5533632	O-RING			
107	5533633	O-RING			
108	5533634	O-RING			
109	5533635	O-RING			
110	5533636	O-RING			
111	5533637	O-RING			
112	5533638	O-RING			
113	5533639	O-RING			
114	5533640	O-RING			
115	5533641	O-RING			
116	5533642	O-RING			
117	5533643	O-RING			
118	5533644	O-RING			
119	5533645	O-RING			
120	5533646	O-RING			
121	5533647	O-RING			
122	5533648	O-RING			
123	5533649	O-RING			
124	5533650	O-RING			
125	5533651	O-RING			
126	5533652	O-RING			
127	5533653	O-RING			
128	5533654	O-RING			
129	5533655	O-RING			
130	5533656	O-RING			
131	5533657	O-RING			
132	5533658	O-RING			
133	5533659	O-RING			
134	5533660	O-RING			
135	5533661	O-RING			
136	5533662	O-RING			
137	5533663	O-RING			
138	5533664	O-RING			
139	5533665	O-RING			
140	5533666	O-RING			
141	5533667	O-RING			
142	5533668	O-RING			
143	5533669	O-RING			
144	5533670	O-RING			
145	5533671	O-RING			
146	5533672	O-RING			
147	5533673	O-RING			
148	5533674	O-RING			
149	5533675	O-RING			
150	5533676	O-RING			
151	5533677	O-RING			
152	5533678	O-RING			
153	5533679	O-RING			
154	5533680	O-RING			
155	5533681	O-RING			
156	5533682	O-RING			
157	5533683	O-RING			
158	5533684	O-RING			
159	5533685	O-RING			
160	5533686	O-RING			
161	5533687	O-RING			
162	5533688	O-RING			
163	5533689	O-RING			
164	5533690	O-RING			
165	5533691	O-RING			
166	5533692	O-RING			
167	5533693	O-RING			
168	5533694	O-RING			
169	5533695	O-RING			
170	5533696	O-RING			
171	5533697	O-RING			
172	5533698	O-RING			
173	5533699	O-RING			
174	5533700	O-RING			
175	5533701	O-RING			
176	5533702	O-RING			
177	5533703	O-RING			
178	5533704	O-RING			
179	5533705	O-RING			
180	5533706	O-RING			
181	5533707	O-RING			
182	5533708	O-RING			
183	5533709	O-RING			
184	5533710	O-RING			
185	5533711	O-RING			
186	5533712	O-RING			
187	5533713	O-RING			
188	5533714	O-RING			
189	5533715	O-RING			
190	5533716	O-RING			
191	5533717	O-RING			
192	5533718	O-RING			
193	5533719	O-RING			
194	5533720	O-RING			
195	5533721	O-RING			
196	5533722	O-RING			
197	5533723	O-RING			
198	5533724	O-RING			
199	5533725	O-RING			
200	5533726	O-RING			
201	5533727	O-RING			
202	5533728	O-RING			
203	5533729	O-RING			
204	5533730	O-RING			
205	5533731	O-RING			
206	5533732	O-RING			
207	5533733	O-RING			
208	5533734	O-RING			
209	5533735	O-RING			
210	5533736	O-RING			
211	5533737	O-RING			
212	5533738	O-RING			
213	5533739	O-RING			
214	5533740	O-RING			
215	5533741	O-RING			
216	5533742	O-RING			
217	5533743	O-RING			
218	5533744	O-RING			
219	5533745	O-RING			
220	5533746	O-RING			
221	5533747	O-RING			
222	5533748	O-RING			
223	5533749	O-RING			
224	5533750	O-RING			
225	5533751	O-RING			
226	5533752	O-RING			
227	5533753	O-RING			
228	5533754	O-RING			
229	5533755	O-RING			
230	5533756	O-RING			
231	5533757	O-RING			
232	5533758	O-RING			
233	5533759	O-RING			
234	5533760	O-RING			
235	5533761	O-RING			
236	5533762	O-RING			
237	5533763	O-RING			
238	5533764	O-RING			
239	5533765	O-RING			
240	5533766	O-RING			
241	5533767	O-RING			
242	5533768	O-RING			
243	5533769	O-RING			
244	5533770	O-RING			
245	5533771	O-RING			
246	5533772	O-RING			
247	5533773	O-RING			
248	5533774	O-RING			
249	5533775	O-RING			
250	5533776	O-RING			
251	5533777	O-RING			
252	5533778	O-RING			
253	5533779	O-RING			
254	5533780	O-RING			
255	5533781	O-RING			
256	5533782	O-RING			
257	5533783	O-RING			
258	5533784	O-RING			
259	5533785	O-RING			
260	5533786	O-RING			
261	5533787	O-RING			
262	5533788	O-RING			
263	5533789	O-RING			
264	5533790	O-RING			
265	5533791	O-RING			
266	5533792	O-RING			
267	5533793	O-RING			
268	5533794	O-RING			
269	5533795	O-RING			
270	5533796	O-RING			</

U.S. DEPARTMENT OF ENERGY									
Referred to									
JAN 15 1975									
DATE	TO	FROM	SUBJECT	INITIALS	DATE	TO	FROM	SUBJECT	INITIALS
1/15/75	100-100000000	100-100000000	100-100000000	100-100000000	1/15/75	100-100000000	100-100000000	100-100000000	100-100000000
1	100-100000000	100-100000000	100-100000000	100-100000000	1	100-100000000	100-100000000	100-100000000	100-100000000
2	100-100000000	100-100000000	100-100000000	100-100000000	2	100-100000000	100-100000000	100-100000000	100-100000000
3	100-100000000	100-100000000	100-100000000	100-100000000	3	100-100000000	100-100000000	100-100000000	100-100000000
4	100-100000000	100-100000000	100-100000000	100-100000000	4	100-100000000	100-100000000	100-100000000	100-100000000
5	100-100000000	100-100000000	100-100000000	100-100000000	5	100-100000000	100-100000000	100-100000000	100-100000000
6	100-100000000	100-100000000	100-100000000	100-100000000	6	100-100000000	100-100000000	100-100000000	100-100000000

NOTES:

DETAIL B
12 PLACES
SCALE: 2:1

LOCKHEED MISSILES & SPACE COMPANY, INC.
8 THUNDERBOLT AVENUE, AEROSPACE RESEARCH LABORATORY
GLADSTONE, CALIFORNIA

ROTOR ASSY
COAL FEEDER

DATE	TIME	DATE	TIME
E 7077	5533545		
SCALE 1:1		SCALE 1:1	

NOTES:

1. ALL MACHINED SURFACES TO BE
OR NOTED ✓
2. INTERNAL SURFACE TO BE BORIDE
CASE HARDENED TO DEPTH OF .001 INCH
ON CAST CARBON STEEL SUBSTRATE
AS ALTERNATE MATERIAL.

SECTION A - A
SPRUE

SCALE 4/1

DIMENSION "A" & "B"

DASH	"A"	"B"	"C"
-1	.328	.231	.252
-3	.402	.283	.308
-5	.490	.345	.376
-7	.500	.390	.425

VIEW C

SCALE 20/1

REF: FOR A 2-020 PARKER 'O' RING

VIEW D

SCALE 20/1

REF: FOR D 2-DIG PARKER "O" RING

[illegible]

U.S. DEPARTMENT OF ENERGY

[illegible]

FILE ID 44-6714

NOTES:

1. ALL MACHINE SURFACES TO ¹²⁹✓
BE BETTER AND NOTED
2. BREAK ALL EDGES AND ROUGH CORNERS
3. MAY BE PURCHASED FROM HYPERSON
KENTVILLE, CA.

VIEW 8

VIEW (12 PLACES

[illegible]

6 MATERIAL: TO BE DETERMINED.

2. UNLESS OTHERWISE SPECIFIED:
SURFACE ROUGHNESS $\sqrt{}$ PER NAS329
DRAFT ANGLE 8.5° MAX.
FILLET RADII .12
EDGE AND CORNER RADII .07

9. RUBBER STAMP 4461897 AND REV LTR
TO WHICH PART IS RD.

9. ALL DIMENSIONS CENTERED ON DATUM

④ INTERNAL SURFACE TO BE BORIDE CASE HARDENED TO DEPTH OF .007 INCH ON CAST CARBON STEEL SUBSTRATE AS ALTERNATE MATERIAL.

Δ°	P	T	Y
19.00	1.500	1.181	906
18.00	1.720	1.509	900
17.00	1.910	1.876	890
16.00	1.951	1.952	889
15.00	2.201	2.109	878
14.00	2.469	2.165	872
13.00	2.795	2.677	862
9.00	5.251	5.581	789
8.00	5.599	5.597	790
7.00	6.676	5.651	785
6.00	6.051	6.320	811
5.00	6.721	6.408	801
4.00	6.817	6.620	817
3.00	7.266	7.256	781
2.00	7.722	7.770	770
1.00	8.206	8.205	767
0.00	8.721	8.721	008

[illegible]

1. ALL MACHINE SURFACES TO ¹²⁹✓
BE BETTER AND NOTED
2. BREAK ALL EDGES AND ROUGH CORNERS
3. HEAT TREAT TO ROCKWELL C-40

SECTION A-A
SCALE 9/1

[illegible]

FILE NO. 100-3341

1. ALL DOCUMENTS SUBMITTED TO BE
RECEIVED

SCALE 2/1

[illegible]

NOTES:

1. NAME FROM STITCHED STAINLESS STEEL VIN.
2. UNIT METALURGICAL CORP.
3. 1000 INDUSTRIAL BLVD.
4. FRAMINGTON, MASS. 01022
5. FRAMINGTON, MASS. 01022

DRILL NO.	1/2" DIA.
1	.250
2	.250
3	.250
4	.250
5	.250

U.S. DEPARTMENT OF ENERGY	
OFFICE OF NEUTRON PHYSICS	
RESEARCH AND DEVELOPMENT DIVISION	
PROJECT: FILTER SPRUE	
SUBJECT: FILTER SPRUE	
DRAWN: D. DANIEL	
CHECKED: D. DANIEL	
DATE: 10/1/77	
FILE NO: 800142-10

U.S. DEPARTMENT OF ENERGY		PROJECT 123	
SUBJECT: BEARING ASSEMBLY		DRAWING NO. 123456	
REVISION: 1		DATE: 12/15/78	
BY: J. D. SMITH		CHECKED: J. D. SMITH	
APPROVED: J. D. SMITH		SCALE: 1:1	
SHEET NO. 1		TOTAL SHEETS: 1	
PROJECT NO. 123		TASK NO. 456	
SUB-TASK NO. 789		DRAWING NO. 123456	
REVISION: 1		DATE: 12/15/78	
BY: J. D. SMITH		CHECKED: J. D. SMITH	
APPROVED: J. D. SMITH		SCALE: 1:1	
SHEET NO. 1		TOTAL SHEETS: 1	
PROJECT NO. 123		TASK NO. 456	
SUB-TASK NO. 789		DRAWING NO. 123456	

NOTES

1. REWORK SEAL DIVISION
6TH GLENN AVE
DREHLENG. ILL. 60090

1 O-RING

2 O-RING

3 O-RING

4 O-RING

5 O-RING

6 O-RING

1 SHAFT ASSY

QTY	ITEM	DESCRIPTION	REMARKS	DATE	BY
1	10-32X1/2	CAP SCREW	ALLEN OR EQ		
1	2-100	O-RING	PARKER	DUNA-H	
1	5539563	WATERING RING			
2	2-100	O-RING	PARKER	DUNA-H	
1	5539523	SHAFT			
1	-501	SHAFT ASSY			
QTY	ITEM	DESCRIPTION	REMARKS	DATE	BY
1	10-32X1/2	CAP SCREW	ALLEN OR EQ		
1	2-100	O-RING	PARKER	DUNA-H	
1	5539563	WATERING RING			
2	2-100	O-RING	PARKER	DUNA-H	
1	5539523	SHAFT			
1	-501	SHAFT ASSY			

U.S. DEPARTMENT OF ENERGY

**SHAFT ASSY
POWER INPUT
COAL FEEDER**

7277 5539543

FILE ID: 8001.52-14

FILE ID: W001,92-14

D-33

NOTES:

1. ALL MACHINE SURFACES TO $\sqrt{16}$ OR BETTER AND NOTED
2. BREAK ALL EDGES AND ROUGH CORNERS
3. TAP 10-24UNC-1B 7.000 DIA 2.175 DEEP 2 PLACES

ITEM	DESCRIPTION	QTY	UNIT	REMARKS
1	PLUG	1	PC	7.0 DIA HO
2	DRUM	1	PC	12.5 DIA HO
3	SHAFT, BRG	1	PC	2.75 DIA HO
4	SHAFT, SEAL	1	PC	2.75 DIA HO
5	SHAFT, SEAL	1	PC	2.75 DIA HO

U.S. DEPARTMENT OF ENERGY

POWER-INPUT SHAFT

COAL FEEDER

DATE: 10 MAR 70

DESIGNED BY: D. DANIEL

CHECKED BY: D. DANIEL

FILE NO: 5533523

FILE ID: 0001.72-19

NOTES:

1	172-1200-1	CAP SCREW			10
2	6433510	HOLD DOWN			17
3	172-1200-1	CAP SCREW			10
4	6433510	RETAINING RING	SMALLER		16
5	6433510	SLEEVE	BEARING		14
6	6433510	SEAL	BEARING		13
7	2-140	O-RING	PARSER	DUNA-4	12
8	6433510	BEARING			11
9	172-1200-1	SEAL ASSY	BEARING		10
10	172-1200-1	CAP SCREW			9
11	172-1200-1	CAP SCREW			8
12	2-040	O-RING	PARSER	DUNA-4	7
13	6433510	RETAINING RING			6
14	6433510	RETAINING RING			5
15	6433510	RETAINING RING			4
16	6433510	RETAINING RING			3
17	6433510	RETAINING RING			2

U.S. DEPARTMENT OF ENERGY

BEARING INSERT ASSY
POWER SIDE

DATE: 10/10/77
BY: D. DANIEL
CHECKED: 10/10/77
APPROVED: 10/10/77

FILE NO. 4001-12-10

D-34

D-35

NOTES

1. ALL MACHINE SURFACES TO BE $\sqrt{32}$ OR BETTER AND NOTED
2. LEFT HAND PART SHOWN WITH ODD DASH NUMBER.
RIGHT HAND PART, NEXT CONSECUTIVE EVEN DASH NUMBER
3. NOTE ROTATION OF SHAFT IN RIGHT AND LEFT HAND PARTS

DETAIL ITEM 2
SCALE 2/1

DETAIL C
SCALE 1/1

ITEM	QTY	DESCRIPTION	UNIT	QTY	DESCRIPTION	UNIT	QTY	DESCRIPTION	UNIT
1	1	NOZZLE	1/2" DIA	1	BEARING	1/2" DIA	1	BEARING	1/2" DIA
2	1	BEARING	1/2" DIA	1	BEARING	1/2" DIA	1	BEARING	1/2" DIA
3	1	BEARING	1/2" DIA	1	BEARING	1/2" DIA	1	BEARING	1/2" DIA

U.S. DEPARTMENT OF ENERGY

EXAMINER'S DATA PER

DESIGNER'S DATA PER

DATE: 10/1/68

BY: A. J. DANIEL

5533529

FILE NO: 4001-52-11

SECTION B-B

LAURENCE STEEL & PIPE COMPANY INC
A DIVISION OF LAURENCE STEEL & PIPE COMPANY
BIRMINGHAM, AL 35204

MAIN JOURNAL BEARING
COAL FEEDER

Item Code: 17077
Quantity: 5533523

2		1		
REVISION				
REV	BY	DESCRIPTION	DATE	APP.

[illegible]

FILE NO: 100-52-14

NOTES:

DETAIL A
SCALE: 9:1

TACK WELD (9) PLACES.
DO WELD NAT'L TO PROTRUDE
ABOVE END SURFACE OF
HOUSING. GRIND IF NECESSARY

[illegible]

PREPARED FOR
U.S. DEPARTMENT OF ENERGY

[illegible]

FILE NO: 44-38861-14

[illegible]

D-41

[illegible]

FILE ID: 44001.92-14

[illegible]

[illegible]

1. ALL MACHINE SURFACES TO ¹²⁹✓
BE BETTER AND NOTED
2. ALL PLATE MATERIAL ASTM A70.
3. DO NOT PAINT INTERNAL OR EXTERNAL
MACHINE FLANGES.

[illegible]

Appendix E
SUBSYSTEM DEFINITION STUDY FOR THE KINETIC EXTRUDER

Section 1
INTRODUCTION

A Kinetic Extruder (KE) pressurizing system consists of a number of subsystems in addition to the KE unit itself. These subsystems include a transport system for bringing the coal to and from the KE unit, a prime mover for supplying the required power to the KE, a lubrication and cooling system for the bearings and seals, a gas supply system for controlling the receiver pressure, supplying purge and fluidizing gases and providing control gas if fluidic turn down control is used, a rotor suction system to maintain the desired rotor inlet pressure, and an instrumentation and control system to provide automated operation of the feed system and integration into the overall plant operation. The purpose of this study is to define the systems requirements for a 7 tons-per-hour (TPH) scale Kinetic Extruder system by providing, for each subsystem, functional description, design requirements, envelope size, and interfaces. Data were gathered from equipment suppliers to determine subsystem availability. Specific sizes apply to a 7-TPH system, but functional descriptions and general requirements would also apply to other size units.

In the present study, a broad definition for the coal pressurizing system is assumed. The system interfaces are taken to be at the pulverizer outlet (at ground level), and at the reactor inlet. An overall view of the assumed system is shown in Fig. E-1 and a closer view of the KE unit and associated subsystems is given in Fig. E-2.

As shown in Fig. E-1, coal from the preparation plant, pulverized and dried at 200°F, is fed into the Coal Feeder System by an inert gas stream. The pulverized coal is first deposited into the receiving hopper of the pneumatic conveyor. From there, the coal is fed by starlock feeder into the batch tank, and then into a blow tank where a compressed airstream carries the pulverized coal through the conveying pipe to the top of the structure. The coal is then passed through a filter/receiver, where the gas is separated from the coal and vented through a dust collector system.

The coal then passes through a vibrating screen/scalper separating any tramp material. The screened coal is weighed before being gravity fed into a low pressure (LP) storage tank. A coal sampling device is installed between the scalper and the storage tank for checking the quality of coal. At the 7-TPH scale, the LP storage tank is designed to contain one hour capacity of coal, measuring it with a leveling device. The stored coal is fluidized by inert gas and fed into a stationary feed pipe and enters the KE unit rotor. The KE unit pumps the coal into the high pressure environment of the rotor housing. The coal is fed from the rotor vessel into a HP storage tank, before it is fed into the gasification reactors.

Based on the above overall configuration, a systems study was carried out to better define and trade off individual subsystems designs. It may be noted that most of the subsystems are commercially available systems or components. Except for the KE unit itself, original designs are not required.

Fig. E-1 Kinetic Extruder Pressurizing System Configuration

Fig. E-2 Kinetic Extruder Closeup View

Section 2
GUIDELINES AND ASSUMPTIONS

The preliminary assumptions used in this study are listed below:

- o Operating duty cycle -- 24 h/day, 7 days/week
- o Material Processed -- Bituminous coal
- o Coal Size -- 70 percent through 200-mesh
- o KE Data:
 - Capacity (TPH) 7
 - Pressure (psig) 400
 - Inert Gas GN₂
 - Bearing Type Journal
 - Sealing Mechanical Face Seal
- o Vessel Sizing -- Design to conform to ASME
 Unfired Pressure Vessel Code
- o Coal Feeder Plant Life (yr) -- 20
- o Equipment selection -- Will be based upon
 state-of-the-art
- o Prime mover -- High-pressure steam turbine,
 diesel, or electrical power
 drive

Section 3

COAL PRESSURIZING SYSTEM REQUIREMENTS

The overall system requirements for the 7-TPH Coal Feeder system are discussed in this section. The requirements were prepared using present coal handling procedures as a guide and requiring an ultimate coal handling capacity of 150 TPH or greater.

The applicable plant system requirements for the case at hand are as follows:

- o The coal feeder system will be designed to handle the processing of coal at the 7-TPH rate.
- o The plant will be designed to operate for 24 h/day and 7 days/week
- o The coal feed stock will be a nominal grind size of 70 percent passing through 200-mesh.
- o The KE will be capable of operating at a pressure of 400 psi and with temperatures of up to 300°F.
- o Equipment selected for all subsystems will be designed for a plant life of 20 yr, and selection will be based on state-of-the-art technology and/or on-the-shelf hardware.
- o Storage tanks will be designed in accordance with ASME Code for Unfired Pressure Vessel.
- o All electrical components will be of commercial quality and will conform to federal or IEEE specifications suitable for the lifespan of 5 yr without jeopardizing the intended service.
- o All electrical motor drives will be explosion proof.
- o Sequential logic and procedures for operational plant startup and shutdown will conform to commercial plant practices.
- o All control of the equipment will be monitored from a central plant control center.

- o Control valves and piping connections will be hydraulically or pneumatically actuated in accordance with standard commercial practices.
- o A suitable structural frame will be provided to support all the equipment with the necessary platforms to support equipment foundations.
- o Foundations for each component will be designed to withstand the structural and vibration loads imposed by the overall plant requirements based on standard commercial practices.

The Coal Feeder Plant conforms to the following general operating requirements:

- o Safety -- All equipment and components selected will be based on appropriate government and industry codes to cover all phases of safety; for plant personnel, safety guidelines will be in accordance with Williams/Steiger, October 1970 (OSHA).
- o Economy -- Operational economy will be achieved with low-cost equipment and construction, and with efficient operation.
- o Maintenance -- Provisions will be made for efficient maintenance during downtime; specific maintenance will be scheduled annually and periodically as necessary.
- o Maintenance handling equipment will be provided in accordance with (1) the weight of the largest component to be installed or removed, and (2) span and lifting capacity of the crane.
- o Working conditions -- Provisions will be made for optimal lighting, heating, ventilating, medical, and sanitary facilities; noise attenuation will be in accordance with the Federal Noise Control Act, 1972, and OSHA, 1970.
- o Environmental Condition -- As it relates to coal plants, will be in accordance with Williams/Steiger, October 1970 (OSHA), in controlling fly ash and other emissions.

Section 4

COAL FEEDER SUBSYSTEMS

This section presents the different subsystems requirements applied to a 7-TPH Coal Feeder Plant. The plant system configuration calls for the use of a single KE driven by a prime mover. The coal feeder system will employ either a mechanical or pneumatic transport method of conveying the pulverized coal to a height of up to 100 ft. The coal will be fed and accumulated by gravity flow to a low pressure (LP) or atmospheric storage tank before it is fed into the KE. In the KE, the dry pulverized coal will be pressurized and fed to a high pressure (HP) storage tank, from which it will be transported into the gasification reactors.

Fourteen elements evolved to support the Coal Feeder Plant as presented in the schematic layout and are listed below:

1. Lift System
2. Vibrating Screen Scalper
3. Weighing System
4. Feed Hopper
5. Kinetic Extruder (KE)
6. Prime Mover
7. Rotor Eye Suction System
8. Gas Systems
9. HP Storage Tank
10. Baghouse
11. Coolant and Lubrication System
12. Gas Generation System
13. Control and Instrumentation System
14. Structure

4.1 LIFT SYSTEM

Pulverized coal from the Preparation Plant is transported by the Lift System to a higher elevation. Coal is then fed into a vibrating screen feeder and then into the LP storage tank. The lift system consists of a coal hopper receiving section, elevator section, feeder section, and the elevator drive and controls. The two methods considered in elevating the coal to the desired height are (1) mechanical elevator, and (2) pneumatic conveyor.

4.1.1 Mechanical Elevator (Fig. E-3)

- o The elevator will be able to transport coal at 7 TPH.
- o Operation duty cycle will be 24 h/day, 7 days/week.
- o Elevation from load point to discharge point will be a minimum of 100 ft above ground level.
- o Coal size to be transported will be 70 percent through 200-mesh with 5-percent moisture content, at 200°F.
- o The mean effective bulk density of pulverized coal will be 25 lb/ft³.
- o Receiving section opening will be sized according to the elevator size used for Coal Preparation Processing Plant--(size/TBD by supplier).
- o Dimensions of the elevator TBD will be provided by the supplier.
- o The elevator section will be oriented vertically.
- o The end section will be provided with a uniform 7-TPH rate of feeding to the vibrating screen system.
- o The elevator drive will use an electric motor with variable speed, continuous duty, 460 V, 3 phase, 60 Hz 1800 rpm in accordance with the National Electric Code.

4.1.2 Mechanical Elevator Sizing Analysis

An analysis of the mechanical conveyor system for the development of a feeder/handling of coal is presented in this subsection. The conveyor system considered is a bucket elevator consisting of a steel knuckle chain to which are attached open receptacles equidistant from one another as shown in Fig. E-3.

Fig. E-3 Mechanical Bucket Elevator

The velocity is given in terms of the distance covered by the transported material per unit of time so that the corresponding number of buckets that has passed through for the same interval of time is represented by v/d . With Q representing the capacity of each bucket and the density of the transported material, the flowrate is expressed as

$$w = \frac{\rho v Q}{d} \quad (E.1)$$

where

- w = flow rate of material being transported (lb/min)
- ρ = density of material (lb/ft³)
- Q = volume capacity of a bucket (ft³)
- d = spacing between buckets (ft)
- v = velocity of material (ft/min)

Using Eq. (E.1), calculations for sizing the elevator equipment were undertaken with the selection of the appropriate components, based on the data extracted from suppliers' catalogs. The calculation will provide data on the following:

- o Motor drive size in horsepower (hp)
- o Chain pull (lb)
- o Bucket size and configuration (in. or ft)
- o Linear speed of operation (ft/min)

The sequence of operation of the mechanical elevator is shown in Fig. E-4 with dry pulverized coal deposited on a coal bin on top of a bucket feeder.

The coal bin will have a variable airlock valve to control the amount of material fed into the bucket feeder. The feeder carries the coal to the inlet opening of the bucket elevator unit where the receptacles are filled with coal material as they pass. Each bucket carries the pulverized coal vertically upward. At the top of the bucket elevator unit, the coal is unloaded to a weighing belt before being fed into a vibrating screen scalper of about 10

Fig. E-4 Mechanical Elevator Schematic

mesh. This protects the Kinetic Extruder unit from oversize coal or tramp material.

A preliminary calculation for the mechanical elevator is presented using Eq. (E.1), the material transport rate expression derived previously:

$$w = \frac{\rho v Q}{d}$$

To produce the required 7 TPH, the amount that the elevator has to carry is obtained from

$$w = 3.89 \text{ lb/s (7 TPH)}$$

Using this value with

$$\rho = 25 \text{ lb/ft}^3$$

and

$$d = 18 \text{ in. or 1.5 ft}$$

the bucket capacity and elevator speed relationship is given by Eq. (E.1)

$$Q = \frac{0.233}{v}$$

The choice of bucket capacity and spacing is made compatible with the commercially available elevator system.

A standard bucket elevator was chosen for this application. The sizing based on catalog values, is summarized in Table E-1. A 10-TPH requirement was assumed for conservative design.

Table E-1

BUCKET ELEVATOR DESIGN CHART

Chain Mounted, Style D Buckets, Steel Knuckle Chains

Elevator Number	Maximum Size Pieces		Capacity				Max. Centers				Buckets		Chain		Horsepower For Various Wts. of Materials (3)							
			Tons Per Hour Materials Weighing (2)				in Feet for Materials Weighing				Style D 10 Ga.		Chain Num- ber	Speed in Feet Per Min.	For Max. Ctrs.				Deduct Per Foot			
	Uni- form Size	10% of Whole (1)	Lbs. Per Cu. Ft.				Lbs. Per Cu. Ft.				Size	Spac- ing			Lbs. Per Cu. Ft.				Lbs. Per Cu. Ft.			
			25	50	75	100	25	50	75	100					25	50	75	100	25	50	75	100
	VC-85	¾	2½	10	20	30	40	100	100	100	100	8x5x 7¾			8	6102BM	160	2.24	3.65	5.05	6.46	.013
VC-126	1	3	15	30	45	60	100	100	100	100	12x6x11¾	12	6110MB	160	2.95	5.01	7.07	9.13	.019	.039	.060	.079
VC-128	1½	3	25	50	80	105	100	100	100	100	12x8x11¾	12	6110MB	160	4.69	8.37	12.06	15.74	.035	.071	.107	.143
VC-148	1¾	4½	30	60	92	125	100	100	100	95	14x8x11¾	12	6110MB	160	5.36	9.67	13.97	17.43	.041	.083	.125	.167
VC-168	2	5	35	70	105	140	100	100	100	85	16x8x11¾	12	6110MB	160	6.04	10.97	15.91	17.79	.047	.095	.143	.191

Elevator Number	Head Shaft				Boot (7)		Casing	Shipping Weight Lbs. (5)			
	Diam. Shaft	Brg. Size (6)	Diam. Spkt.	R.P.M.	Diam. Shaft	Diam. Spkt.	Number	Head & Boot Sections		Intermediate Sections	
								Machinery (4)	Casing	Chain & Bkts. Per Ft.	Casing Per Ft.
VC-85	3⅝	3⅝	24¼	25	2⅝	18	4416	1000	1695	37	75
VC-126	3⅝	3⅝	25	24.6	2⅝	17.5	4416	1060	1685	40	75
VC-128	4⅝	3⅝	25	24.6	2⅝	17.5	5022	1230	2150	46	90
VC-148	4⅝	3⅝	25	24.6	2⅝	17.5	5022	1260	2145	49	90
VC-168	4⅝	3⅝	25	24.6	2⅝	17.5	5022	1300	2155	53	90

(1) Mixed with fines.

(2) Buckets filled to approx. 75% of catalog rating.

(3) Horsepower at head shaft (based on buckets 100% full).

(4) Weight of machinery includes chain and buckets required for head and boot sections.

(5) Total weight of elevator = (weight of head & boot sections and machinery) + (weight of intermediate casing and machinery) × (total height of intermediate sections).

(6) Head shaft & screw take-up shaft furnished with roller bearings.

(7) Gravity take-up can be furnished when specified.

All dimensions in inches unless noted.

Table E-1 (cont.)

Style D Continuous

A-2021-D1

Size of Buckets			Capacity Δ Cubic Feet		Dimensions Inches		Weight Lbs. 10 Ga. Steel
Length In.	Projection In.	Depth In.	At Line X-X	At Line Y-Y	A	B	
8	5	7 $\frac{1}{4}$	.075	.045	4 $\frac{1}{16}$	3 $\frac{3}{8}$	7.0
12	6	11 $\frac{1}{4}$	.165	.095	7 $\frac{1}{16}$	5 $\frac{1}{8}$	12.5
12	8	11 $\frac{1}{4}$	.295	.170	7 $\frac{1}{16}$	5 $\frac{1}{8}$	15.5
14	8	11 $\frac{1}{4}$	.345	.200	7 $\frac{1}{16}$	5 $\frac{1}{8}$	17.0
16	8	11 $\frac{1}{4}$	.395	.230	7 $\frac{1}{16}$	5 $\frac{1}{8}$	19.0

CHAIN TABLE

Chain No.	Pitch In Inches	Average Ultimate Strength Lbs.	Rated Working Value Lbs.	Wt. per Ft. Plain Chain Lbs.	Attachment Wt. per Ft.		Dimension in Inches		
					No.	Lbs.	Pin Dia.	Side Bar	Knuckle Dia.
6102-BM	4.00	44,000	6,500	6.9	K-2	9.0	$\frac{3}{8}$	1 $\frac{1}{2}$ x $\frac{3}{8}$	1
6110-MB	6.00	44,000	6,300	6.4	K-2	8.8	$\frac{3}{8}$	1 $\frac{1}{2}$ x $\frac{3}{8}$	1 $\frac{1}{4}$
6856-M	6.00	100,000	14,000	16.5	K-24	21.5	1	2 $\frac{1}{2}$ x $\frac{1}{2}$	1 $\frac{3}{4}$
6869	6.00	200,000	21,800	34.0	K-44	68.0	1 $\frac{1}{4}$	4 x $\frac{3}{8}$ °	2 $\frac{3}{8}$
6859	9.00	87,000	12,250	13.6	G-9	20.0*	1	2 $\frac{1}{2}$ x $\frac{1}{2}$	1 $\frac{3}{4}$
6322	12.00	46,000	7,250	7.1	G-16A	8.2*	$\frac{3}{4}$	2 x $\frac{3}{8}$	1 $\frac{1}{2}$

* Attachment one side only, all other attachments both sides.

° Outer side bars equal 3 inches.

Referring to Table E-1, for vertical continuous bucket elevators, a 10 TPH capacity for a 25 lb/ft³ material was chosen. Since a transporting distance of 100 ft vertical is required, then

$$\begin{aligned} \text{hp} &= \text{(required at head shaft)} = (2.24) \pm h (0.013) \\ h &= \text{length beyond (+) or short (-) of 100 ft} \end{aligned}$$

Therefore,

$$\text{hp} = 2.24$$

In sizing the speed reducer, we assume an overload service factor equal to 1.5

$$\text{shaft hp} = 2.24 \times 1.5 = 3.36$$

Assume a belt drive efficiency of 85 percent; then the motor size required is

$$\text{hp} = \frac{3.36}{0.85} = 3.95$$

A 5-hp motor is selected to drive the speed reducer of the elevator.

4.1.3 Pneumatic Conveyor

A pneumatic conveyor transports dry, free-flowing, suspended granular material inside a pipe or a duct using high velocity gas-stream, as shown in Fig. E-5.

The sequence of operation of the pneumatic conveyor shown in Fig. E-5, is as follows:

The coal bin is equipped with an airlock valve system to control the amount of material fed into the receiver vessel. The coal is transferred in batches. When the vessel is filled, the airlock valve is closed and the

Fig. E-5 Pneumatic Conveyor Schematic

pneumatic air from the air blower is fed into the vessel to aerate the pulverized coal and convey it to the top receiver-vessel. The coal is then transported into a vibrating screen feeder.

Requirements are as follows:

- o Elevation from load point (ground level) to discharge point will be a minimum of 100 ft
- o The receiving-section opening will be sized (TBD) according to the conveyor used for Coal Preparation Processing Plant.
- o The conveyor section will transport the coal using air.
- o Selection of conveyor will be based on cost and reliability.
- o Dimensions of the conveyor TBD will be provided by supplier.
- o The conveyor tube of schedule 40 carbon steel will be oriented vertically, with diameter TBD by supplier.
- o The end section will be provided with a uniform 7-TPH rate of feeding to the vibrating screen system.
- o The conveyor drive will require an air flow of up to 500 scfm.
- o The conveyor system will be enclosed, and will have a pressure drop of 6 to 8 in. Hg.
- o The conveyor will be self-cleaning and highly efficient in handling pulverized, dry, free-flowing coal at a nominal speed of 4000 to 5000 ft/min.
- o All pneumatic connections and pipings installations will be in accordance with standard commercial plant practices.
- o The conveyor blower will be provided with a positive displacement blower assembly operating between 1400 and 1800 rpm, and 6 to 8 in. Hg vacuum; the blower will be belt-driven with an electric motor.

4.1.4 Lift System Selection

A short tradeoff study was conducted comparing the merits of the mechanical elevator with various types of pneumatic conveyors. Advantages and disadvantages of the two types of systems are summarized in Table E-2. A

Table E-2

COMPARISON OF MECHANICAL AND PNEUMATIC CONVEYOR

Elevator System	Advantages	Disadvantages
BUCKET TYPE MECHANICAL ELEVATOR	<ul style="list-style-type: none"> (1) On the shelf-equipment items (2) Proven technology (3) Low initial cost (4) Nominal safety and good environmental control achievable 	<ul style="list-style-type: none"> (1) Periodic mechanical wear and parts breakdown (2) Fixed installation (3) Limited flow rate control (4) Requires large installation space allocation (5) High maintenance cost
PNEUMATIC CONVEYOR	<ul style="list-style-type: none"> (1) Flexible installation with minimum space requirement (2) Mechanical moving parts minimized (3) Less maintenance work (4) Wider degree of material flow rate control (5) Good pollution control in closed loop system (6) Low cost 	<ul style="list-style-type: none"> (1) Strict seal and leak-proof joints requirements (2) Additional filtering and disentrainment equipment required (3) Open loop system unattractive to clean air environmental requirements (4) Closed loop system expensive (5) Requires high air pressure line installation (6) High initial cost

pneumatic positive pressure batch conveyor was selected for the following reasons:

- o It is an enclosed system, and thus eliminates wastage and spillage of coal which would otherwise be exposed to mechanical parts.
- o Since the system has a minimum of parts, the overall cost of maintenance is relatively low, which, when compared to the mechanical type, offsets the relatively high initial cost.
- o It requires minimum space for installation.

4.2 VIBRATING SCREEN SCALPER SYSTEM

The vibrating screen system shown in Fig. E-6 is a scalper which prevents any oversized coal or tramp material which could plug the KE sprues from being delivered to the LP storage tank. Refuse coal is directed back to the pulverizer unit.

The vibrating screen system consists of a close pan, screen feeder, ducting, vibrating mechanism, and electrically controlled motor drive.

Requirements are as follows:

- o The capacity of the vibrating screen (Fig. E-6) will be 7 TPH.
- o The operation duty cycle will be 24 h/day, 7 days/week.
- o The vibrating screen will be located either at ground level or elevated up to 130 ft. Final determination will be based on supplier recommendations as to advantages in terms of structural requirements, weight, vibration, and effect of weighing.
- o Pulverized coal size to be fed will be 70 percent through 200-mesh.
- o The mean bulk density for pulverized coal will be 25 lb/ft³.
- o Coarse coal will be ducted by gravity to the Coal Preparation Plant.
- o The screen size will be 10-mesh scalper.
- o The vibrating mechanism will provide the required vibration to screen and feed the coal to the storage hopper.

Fig. E-6 Vibrating Screen Feeder System

- o The feeders will be driven by an electric motor drive having TBD hp, 220/440 V, 3 phase 1850 rpm in accordance with the National Electric Code.
- o The screen feeder will be provided with a magnetic pulley capable of extracting tramp ferrous materials.

4.3 WEIGHING SYSTEM

The weighing system will determine the weight of the pulverized coal being fed to the LP storage tank during coal motion feeding.

The weighing system (Fig. E-7) consists of strain gage, load cells, tachometer generator, solid state electronics integrator totalizer, capacity rate meter, feeder rectifier, set point controller, converter weight recorder, remote control, and readout.

Requirements are as follows:

- o The capacity will be 7 TPH.
- o The weighing system will be able to determine weight of pulverized coal during continuous operation.
- o Coal size will be 70 percent through 200-mesh.
- o The mean bulk density for pulverized coal will be 25 lb/ft³.
- o The load cells will measure the weight of the coal in motion and relay the data to the recorder.
- o Additional load cells, if necessary, will be provided to increase reliability.
- o The weighbelt system is a constant-speed belt feeder with a rotary-inlet gravimetric feed, electrically driven and actuated by a pneumatic scale beam linked to the low inertia weighbelts, as shown in Fig. E-7. Accuracy is at least 1 percent of full scale.
- o A double weighing system is currently being considered: (1) a weigh-belt system located between the elevator and vibration screen system, and (2) an optional weigh system attached to the HP storage tank.

Fig. E-7 Weighing Systems (Schematic)

- o Both systems will use vibration isolation and slip joints for expansion and contraction of the major components.
- o Control components will be isolated from the production zone and dust can be air purged.
- o The rotary inlet will be a low-pressure air lock using a gravity flow design, suitable for pulverized coal.
- o The rotary inlet/air lock will be made to control input to the weighbelt and to maintain a vertical position to accommodate stop-and-restart.
- o Sensors will be provided to maintain constant speed on the belt and the vertical gate control; alarm switches will be provided as a safety precaution at each hazardous location.
- o The inlet and outlet hoppers will meet the interface opening requirements of the elevator and LP storage tank units.
- o Belt will be selected from material that is extra sensitive to high loading and will be able to withstand 250°F without expanding.

4.4 FEED HOPPER

The feed hopper (Fig. E-8) receives and stores the pulverized coal fed by the vibrating screen feeder system. The feed hopper consists of a cylindrical and conical shell, lid, support ring, access openings, and piping. The system consists also of a shutoff valve and necessary connections, inert gas fluidizing, purging points, and other instrumentation points. The LP storage tank is zone fluidized around the outlet to the KE unit feed pipe.

Requirements are as follows:

- o The capacity of LP storage tank will be determined by user requirements.
- o The operation duty cycle will be 24 h/day, seven days/week.
- o The load point will have a 100-ft minimum elevation, and the discharge point at 65-ft elevation.
- o Coal size stored will be 70 percent through 200 mesh.
- o Mean bulk pulverized coal density will be 25 lb/ft³.

Fig. E-8 Feed Hopper

- o The receiving hopper opening will be sized to conform to the vibrating screen feeder chute opening.
- o The tank will be designed for 50-psig internal operating pressure in accordance with ASME Unfired Pressure Vessel Code.
- o A shutoff valve will be attached at the bottom pipe outlet.
- o The tank will be constructed of a carbon steel cylindrical shell with a 30-deg conical bottom section and zone fluidized at the discharge; the discharge pipe diameter for the 7-TPH unit is 2 in.
- o Level indicator(s) and means of controlling the start or stop of storage tank feeding and discharge equipment will be provided.
- o Sample access ports at the side of the tanks will be provided at an elevation accessible to operating personnel, as required.
- o Structural storage tank linings of stainless steel 400 will be considered as abrasion-resistant to protect the primary tank structure.
- o Linings will not be computed as contributing to the strength of the tank wall.
- o Blowout panels will be provided in case of tank overpressurization and will be installed in accordance with OSHA Safety codes.

4.5 KINETIC EXTRUDER (KE) SYSTEM

The KE System employs centrifugal force generated by the rotation of a high-speed rotor designed to compact and pump pulverized coal into a high-pressure region. The KE rotor is illustrated by the cutaway drawing in Fig. E-9, and the overall subsystem is shown in Fig. E-10.

The system consists of high speed rotor, receiver vessel, shaft, journal bearings, pressure housing, seals, shutoff valves and pipings, and base support structure.

Requirements are as follows:

- o The KE will be able to deliver 7 TPH against a back pressure of 400 psig.

Fig. E-9 Lockheed Kinetic Extruder Rotor

Fig. E-10 Lockheed Kinetic Extruder

- o Operation duty will be 24 h/day, 7 days/week.
- o Coal size to be transported will be 70 percent through 200-mesh.
- o Coal will be fed into the rotor using the pneumatic transport method.
- o The feed pipeline diameter is 2 in.
- o For transporting pulverized coal into the KE unit, pipeline material will be schedule 40 carbon steel.
- o Pipeline will have minimum number of bends.
- o Expansion joints will be provided to absorb the thermal/shock movement during normal operation.
- o Expansion joints will be provided with replaceable liners made from stainless steel 400.

4.5.1 KE Design Data

The following data were extracted from LMSC computations and experimental pilot plant data:

o Capacity (TPH)	7
o Rotor diameter (in.)	35
o Rotor speed (rpm)	3,300
o Turndown ratio	2:1
o Pressure (psig)	
- Inlet	Atmospheric
- Outlet	400
o Pressurization gas	Inert gas
o Minimum design permeability (10^{-12} ft)	3.0
o Bearings type	Journal
o Sealing	Mechanical contact seal
o Rotor power requirement (hp)	200
o Temperature pulverized coal ($^{\circ}$ F)	
- Input	200
- Output	300

o Sprue parameters:

No. of sprues or channels	18 in single row
- Length (in.)	10
- Inlet radial position (in.)	6
- Outlet diameter, (in.)	0.50
- Area ratio	30

o Control Orifice

- Length (in.)	1.5
- Outlet diameter (in.)	0.13

Startup of the Kinetic Extruder, that is, the establishment of the moving coal plugs, must take place at zero backpressure. To bring a Kinetic Extruder unit online, as shown in Fig. E-10, the coal flow is started by opening valve A while the rotor housing is at ambient pressure and isolated from the high-pressure reactor by the closed valve B. Once the coal plugs are established, the unit is pressurized to the reactor level and valve B is opened. To shut the feeder down, the procedure is reversed. Valve B is closed, the housing is depressurized, and then the coal flow through the rotor is shut off by closing valve A.

Since the unit is feeding coal during pressurization, the rotor enclosure vessel must be sized so that the coal accumulation built up during this transient may be accommodated. The size of the hopper portion of the unit thus depends on the pressurization rates that can be achieved in a stable manner.

Theoretically, only a finite pressurization rate can be tolerated by the dynamics of the coal plugs. The tolerable rate is quite high from a practical standpoint. As long as the time scale for pressurization is much greater than the time constant of the sprue flow, t_s , the sprue pressure distribution will not differ significantly from the steady-state profile. The sprue flow time constant is given by

$$t_s = \frac{L_s}{U_s}$$

where

L_s = sprue length

U_s = average coal velocity

A typical value would be $t_s \approx 1/2$ s. Thus, a pressurization time of the order of, say, $10 t_s$, or 5 s, should be comfortably tolerated.

As an experimental check, some pressurization tests were conducted using the Model No. 3 rotor mounted in the horizontal axis housing. Using the present gas supply line sizes, the highest receiver vessel pressurization rate that could be obtained was: 0 to 200 psig in 20 s (600 psi/min). No sprue flow interruptions or other untoward effects were observed during the pressure transients. This would be expected considering the sprue flow time constant. The tests also showed that typically only 10 to 20 s elapsed between opening the coal feed line valve and the establishment of a pressure sealing coal plug.

To conclude, based on startup transient considerations, the housing hopper section of the feeder need be able to accommodate only a 1/2 to 1 min coal delivery.

Thus, a 7-TPH unit would require a hopper of about 9 ft³ capacity.

4.6 PRIME MOVER DRIVE POWER

Both steam turbine and electrical drive power were considered as prime movers for the KE unit.

4.6.1 Turbine/Drive System

The turbine drive system (Fig. E-11) provides the power required for the KE using dry saturated or superheated steam.

The system consists of the turbine expander, high-speed coupling, base support structure, turbine controls, valves, and piping connectors.

General specifications for a single-stage turbine, which comfortably exceeds the requirements for a 7 TPH KE unit, are given below.

Maximum initial gage pressure (psi/bar)	700/48
Maximum initial temperature ($^{\circ}\text{F}/^{\circ}\text{C}$)	750/399
Maximum exhaust pressures (gage, psi/bar)	vac-100/6.9
Speed Range (rpm)	1000/7315
Wheel pitch diameter (in./mm)	14/360
Number of stages (impulse type)	1
Number of rows of rotating blades	2
Inlet sizes (ANSI, in.)	3
Inlet location (facing governor)	right
Exhaust size (ANSI, in.)	6
Exhaust location (L.H. Standard)	R.H. optional
Approximate range of capacities (hp/kW)	750/560
Approximate shipping weight (lb/kg)	870/400
Dimensions (in.)	
Length	41
Width	31
Height	24

Fig. E-11 Turbine Drive System

The turbine unit will also be provided with the following accessories:

- o NEMA governor system
- o Speed range--direct acting governor
- o Pressure-lubricated bearings (TBD)
- o Insulation and jacketing
- o Tachometer (vibrating reed and electronic)
- o Remote controlled dual steam hydraulic valves
- o Remote control tripping hydraulics
- o Bearing temperature thermocouples
- o Self-leveling, fully equalized, double acting thrust bearing
- o Trip and throttle valves
- o Coupling and coupling guard
- o Steam gages and gage board
- o Base support frame or sled

4.6.2 Electrical Drive

The electrical drive system (Fig. E-12) provides an alternate prime mover to drive the KE through a gear train unit. The system consists of the electrical motor, gear box, high speed coupling, base support, electrical controls, switch control panel, cabling, and connectors.

Requirements are as follows:

o Electrical motor type	Induction Motor, Variable Speed
o Motor rated horsepower	250
o Motor speed (rpm)	3,600
o Voltage	480
o Phase	3
o Cycle	60
o Service factor	1.4
o Motor efficiency	TBD
o Starter type	NEMA enclosed, dust proof, reduced voltage to start switch to full voltage

Fig. E-12 Electric Drive

- o The motor will be explosion proof in accordance with the IEEE or National Electric Code and NEMA Standards.
- o A heavy-duty gear box coupled between the electrical motor and the K unit will be mounted on a base support frame with the output shaft inline with the K rotor shaft, and the input shaft in line with the electrical motor.

4.7 ROTOR EYE SUCTION SYSTEM

The purpose of this system, shown in Fig. E-13, is to maintain a negative pressure at the eye of the rotor by drawing off excess gases. Some coal dust is carried out along with the gases and provision for cleanup and recycle of this coal must be included.

This suction system includes the following:

1. A bag-type dust collector system equipped with a suitable blower or other device able to maintain a 10-in. Hg vacuum in the dust collector; the flowrate through the blower will be in the range of 10 to 50 scfm for the 7-TPH unit.
2. A suitable rotary valve so that the dust collector can be emptied of solids without interrupting the vacuum.

Fig. E-13 Rotor Eye Suction System

A blow tank and pneumatic transfer system (or equivalent) for dustless recycle of any accumulated coal to the main feed hopper may be added if needed.

Based on surveys of vendors, standard commercially available vacuum systems meet the requirements for the suction subsystem. Both roots blower type pumps or centrifugal vacuum blowers can easily generate adequate suction pressures

Finally, it may be noted that in commercial applications, one suction system would normally service multiple Kinetic Extruder units.

4.8 GAS SUPPLY SYSTEMS

4.8.1 High Pressure Gas System

The Kinetic Extruder basically pumps only coal; the gas flow through the sprues is always zero or nearly so. However, gas must be continuously supplied to pressurize the void space between the coal particles. The actual gas flow rate required is mainly dependent on the average bulk density of the coal being removed through the accumulator outlet. Table E-3 shows the gas requirements as a function of this outflow density for the 7 TPH unit. A portion of this gas is injected into the control orifices as "control gas," and the remainder is injected directly into the rotor housing as "pressurization gas."

Table E-3
PRESSURIZING GAS REQUIREMENTS FOR THE 7-TPH UNIT

Reactor Feed Line

Bulk Density (lb/ft ³)	Porosity	Coal Bulk Flow (CFM)	Gas Volumetric Flow (ACFM)	N ₂ Mass Flow ^(a) (lb/h)
20	0.78	11.7	9.1	900
30	0.67	7.8	5.2	520
40	0.56	5.8	3.3	320

(a) Accumulator pressure: 415 psia

Accumulator temperature: 200°F

From the standpoint of the Kinetic Extruder operation, no control beyond simple pressure regulation is required of the pressurization gas. However, it is anticipated that the coal outflow from the rotor housing into the reactor may be regulated by controlling the housing pressure. Complete definition of the system thus requires knowledge of the reactor injection system characteristics.

The control gas pressure is varied independently as a means of controlling the coal delivery rate of the Kinetic Extruder. The coal pumping rate through the rotor is a function of both the rotor speed and the gas pressure difference across the control orifices.

The mass flow may be expressed as:

$$m = \left[A\omega^2 + B(P_1 - P_2) \right]^{1/2}$$

where

- P_1 = control orifice pressure
- P_2 = rotor housing pressure
- ω = rotor speed
- A and B = nominal constants for a given rotor

A schematic of the high-pressure gas system, including instrumentation sensors, is presented in Fig. E-14.

4.8.2 Low-Pressure Gas System

A low-pressure gas supply (approximately 100 psig) is required for feed hopper fluidization, purge gas for the coal feed lines and suction lines as needed, and also to provide a source of rotor eye carrier gas. The rotor eye carrier gas stream is a small flow injected into the eye of the rotor near the suction line inlets. Its purpose is to provide a minimum gas flow in the suction lines at all times. This ensures a relatively high velocity and low coal density in these lines, eliminating any possibility of their plugging. The carrier gas flow is needed only when there is little or no gas back leakage through the sprues. The flow is therefore regulated based on the rotor eye pressure.

Figure E-15 shows the low-pressure gas system schematic including instrumentation sensors and purges.

Fig. E-14 High-Pressure Gas System

Fig. E-15 Low-Pressure Gas System

4.9 HIGH PRESSURE (HP) STORAGE TANK

The HP Storage Tank (Fig. E-16) receives and stores the coal fed by the KE system when the coal is ready to be transported to the gasification reactors.

The HP storage tank consists of the cylindrical and conical shell, lid, structural support ring, access opening, and blowout ports. The system consists of a control shutoff valve and the necessary piping connections for inert gas injection (to transport the coal), purging points, and other instrumentation points necessary to monitor the level of coal inside the storage tank.

Requirements are as follows:

- o The storage tank will have a capacity depending on process requirements
- o The operation duty cycle will be 24 h/day, 7 days/week.
- o Elevation of load point minimum 50 ft and discharge point 10 ft above ground level.
- o Coal size to be transported will be 70 percent through 200-mesh.
- o The tank opening will have a TBD diameter pipe connected to the outlet of the KE vessel.
- o The tank will be designed for 400 psig internal pressure in accordance with ASME Unfired-Pressure Vessel Code.
- o The tank will be insulated to maintain the coal temperature of 300°F
- o Provision for attaching the control valve at the bottom pipe outlet with the use standard flanges will be made.
- o The cylindrical tank will be constructed of stainless steel with material thickness TBD and with a 30-deg conical bottom section to maintain smooth flow of the fluidized coal.
- o The tank discharge port will be zone-fluidized or use other means to ensure proper flow into the reactor feed lines.
- o The level indicator will provide a means of controlling the start or stop of storage tank feeding and its discharge to the reactors.
- o Sample ports at the side of the tanks will be provided at an elevation accessible to operating personnel.

Fig. E-16 HP Storage Tank

- o Structural storage tank linings of stainless steel 400 will be abrasion resistant to protect the primary tank structure.
- o Linings will not be computed as contributing to the strength of the tank wall.
- o Blowout ports will be provided for overpressurization of the tank and will be installed in accordance with the OSHA Safety Codes.
- o Necessary interface attachment for weighing the overall tank will be provided if required.
- o The exterior will be corrosion resistant.

The stress generated for the HP storage tank is mainly due to the pressure of the fluidized coal. The main concern of the tank design, then, is the hoop stress present in the tank structure.

Table E-4 summarizes the tank thicknesses and their corresponding weights.

Table E-4
SUMMARY OF TANK WALL THICKNESS AND WEIGHT
Material - AISI 304, (SS)

	Tank Diameter (in.)	Recommended Wall Thickness (in.)	Tank Weight (lb)
Low Pressure Tank	120	0.25	3,560
High Pressure Tank	96	1.5	21,347

4.10 BAGHOUSE SYSTEM

The baghouse system (Fig. E-17) cleans up the carrier gas used to convey coal into the feed hopper. In the event that the KE unit housing must be depressurized, the discharge also must go through this baghouse for cleanup. Materials collected by the baghouse system are either removed from the unit,

Fig. E-17 Baghouse System

in a manner similar to ash handling, or transported back to the feeder. The dust collectors are especially designed for vacuum applications, either for a mechanical or a pneumatic conveying method.

The system consists of duct, accumulator tank, filter, exhaust blower, electric motor drive, valve connections and tank support structure.

Requirements are as follows:

- o The capacity of the baghouse system will be up to 500 scfm
- o The operation duty cycle will be 24 h/day, 7 days/week
- o The system will be located close to the LP storage tank, and the supplier will make the final determination as to advantages in terms of functional requirements for the removal and recycling of the coal dust, weight, vibration, and effect of weighing
- o Coal dust will be ducted by gravity to the disposal area or recycled to the vibrating screen feeder
- o The filter size will be based on the recommendation of supplier
- o Dust collector tank will be enclosed with a gas ventilator
- o The baghouse unit will be mounted on a beam support structure and will be provided with shock isolation units
- o Exhaust blower will be driven by an electric motor having TBD hp, 460 V, 3 phase 60-Hz 1800 rpm in accordance with the National Electric Code

4.11 COOLANT AND LUBE SYSTEM

The main journal bearing and seal coolant and lubrication system design approach is based on the use of treated water as the fluid, primarily because of its superior cooling quality. The major components of the system are a piston pump and a pilot-operated back-pressure regulator. At a speed of 700 rpm, the pump delivers 7 gpm at 500 psia. The pilot-operated regulator has an inherent cracking pressure of 50 psi which is constant and independent of pilot pressure. During operation, the pressure in the fluid contained by the bearing and seal will rise because of resistance to flow through the bearing.

The pressure increases to 50 psi and is maintained at this level above pilot pressure. Pilot pressure is coal delivery pressure.

Two pump systems will be in parallel for system redundancy and increased system reliability. Filters will be the dual-switchable type for filter change without system shutdown. The feed tube bearing and seal system is identical in design to the main system. Pilot pressure to the regulator is coal entrance pressure instead of coal delivery pressure.

This system (Fig. E-18) provides the necessary fluid pressure and flow required for the operation and cooling of the hydrostatic journal bearings in the KE.

The system consists of hydraulic reservoirs, pump/drive and controls, back-pressure control regulator check valve, piping, and connectors.

Requirements are as follows:

- o The operations duty cycle will be 24 h/day, 7 days/week
- o The hydraulic system will operate at 450 psig maximum pressure
- o All valves and controls will be in accordance with the Hydraulic Institute Standards, 13th edition
- o The pump filter will be high pressure 10 to 25 m filter at the reservoir
- o A 15-gallon capacity reservoir will be required to support the system
- o Hydraulic pumps will be required to provide the flow of 7 gpm to the KE bearings
- o The safety relief valve will be installed in-line to protect the system overpressurization
- o The lubricant will be 60/40 mixture H₂O/prestone
- o All connectors and piping will be in accordance with the Hydraulic Institute Standards, 13th edition

Fig. E-18 Coolant and Lube System

4.12 INERT GAS GENERATION SYSTEM

The gas generation system (Fig. E-19) provides the high and low pressure inert gas streams used by the KE.

The system consists of an inert gas generator, reservoir, pump drive and controls, accumulator, relief valve, primary and secondary compressor, dessicant dryer, separators, heat exchanger, prefilter, and air receiver.

Requirements are as follows:

- o The operations will be 24 h/day, 7 days/week
- o Gas pressurization will consist of a self-sufficient system which provides the fluid power to regulate the movement of coal
- o The required system capacity is 500-900 lb/h
- o The system will generate, compress, and dry inert gas at 500 psig
- o The compressor will be provided with the required motor, intercooler, and after-cooler units
- o Provide desiccant dryer from design pressure of 500 psig
- o All components will be designed and fabricated in accordance with ASME Code, Section 8

4.13 CONTROL AND INSTRUMENTATION SYSTEM

The Kinetic Extruder will be controlled by a relatively simple set of systems. Five basic automatic control loops are involved:

1. Control gas system control loop – adjusts coal flow rate to assure desired coal level in the accumulator section
2. Rotor speed control loop – maintains proper speed in relation to delivery pressure
3. Rotor housing pressure control loop – maintains set delivery pressure
4. Automatic safety shutoff control loop – isolates pump from reactor in the event of a failure

Fig. E-19 Inert Gas Generator System

5. Rotor eye carrier gas control loop – regulates suction line carrier gas injected into rotor eye

Control will be mainly by analog controllers with set points adjusted by the control processor, or manually as a backup.

In addition to the control instrumentation, instrumentation for diagnostic purposes may be needed. An automatic data acquisition system is to be used for this purpose.

The type of computer system selected as the control and data acquisition processor should include a microprocessor with adequate memory, an input/output module, an operations console with a CRT and keyboard, a keyboard printer, and software packages for control and data display.

The requirements for the five control loops and the data acquisition system are sketched out in subsequent sections.

All controls and instrumentation outputs should be located on a control panel some distance from the machine itself. Other manually activated controls, such as for the coal feedline valves and various purge valves, will be located at the control panel.

All instrumentation taps are to be equipped with continuous purges and isolation valves.

4.13.1 Control Gas System Control Loop

This system controls the coal pump mass throughput by controlling the differential pressure, P_c , across the "control orifices" inside the rotor. Here $P_c = P_1 - P_2$ where P_1 = control gas injection point pressure, P_2 = rotor housing pressure. Figure E-20 shows a simplified schematic of the control loop.

Fig. E-20 Schematic of Control Gas Control Loop

At least two types of control of ΔP_c are required:

1. Set and maintain ΔP_c .
2. Adjust ΔP_c to maintain a constant coal level in the accumulator section. However, do not exceed set limits on ΔP_c . The level sensor will provide a signal proportional to level.

4.13.2 Speed Control Loop

Rotor speed is controlled by adjusting the throttle of the engine prime mover. Required rotor speed is mainly a function of the pressure in the rotor housing and should be varied as this pressure changes. Figure A-21 illustrates the speed control loop. Two basic control modes are required:

1. Set and maintain constant speed.
2. Adjust speed in proportion to pressure, within high and low limits.

4.13.3 Rotor Housing Pressure Control

Pressure in the rotor housing is basically controlled by the inflow of pressurization gas into the housing. This should be sufficient under normal circumstances. However, should the coal outflow cease, the pressure in the housing would continue to rise due to the coal influx even with the gas supply turned off. Therefore, provision must be made to valve gas out of the housing if the pressure rises too far above the set point. A schematic of the control loop is shown in Fig. E-22.

4.13.4 Safety Shutoff System Control

The KE feeds a coal gasification reactor which contains high-temperature gases. It is imperative to prevent high-temperature gases from backflowing from the reactor into the KE housing. The KE will nominally operate somewhat above reactor pressure. Should this pressure differential approach zero, the KE must be automatically isolated from the reactor and shut down. Figure E-23 illustrates the type of system required.

Fig. E-21 Rotor Speed Control Loop

Fig. E-22 Rotor Housing Pressure Control Loop

Fig. E-23 Safety Shutoff System

If backflow or imminent backflow in the reactor feed line is indicated either by a D/P cell or a thermocouple in the feedline, several actions are to be taken:

1. The reactor feed ballvalve "B" (see Fig. E-8) is closed. Should the ball valve fail to close fast enough, a backup cartridge-actuated valve "C" is fired.
2. The KE housing is then depressurized by venting to a dust collector.
3. After the housing is depressurized, the coal-feed to the KE is terminated by closing valve "A."

Several alternate failure modes also should initiate the shutoff sequence. Specifically:

- o Loss of power
- o Rise of rotor eye pressure above a set point, indicating "blow-back" through the rotor
- o Loss of pressurization gas supply pressure
- o Failure of suction system
- o Coolant and lube system failure
- o Excessive vibration

4.13.5 Rotor Eye Carrier Gas Control Loop

The rotor eye carrier gas stream is regulated by the control loop shown schematically in Fig. E-24. This regulation is based on the measured rotor eye pressure. Should the rotor eye pressure rise, indicating back leakage, the carrier gas flow rate is reduced.

4.13.6 Data Acquisition System

The control computer system may also be used for data gathering and analysis. Data to be measured are mostly gas pressures, gas flow rates, and temperatures. The following is a list of parameters to be measured to completely characterize the performance of the Kinetic Extruder.

Fig. E-24 Rotor Eye Carrier Gas Loop

- o Pressures:
 - Coal feed hopper
 - Rotor eye
 - Delivery
 - Reactor
 - Control gas and differential
 - Suction system
 - Rotor eye carrier gas
 - Lubrication system
- o Flow Rates:
 - Control gas
 - Pressurization gas
 - Suction system
 - Rotor eye carrier gas
 - High- and low-pressure lubrication system

- o Temperatures:
 - Feed hopper
 - Coal delivery
 - Lubrication system
- o Other:
 - Coal level
 - Rotor speed
 - Sprue flow sensor output
 - Rotor vibration
 - Shaft torque

All data are to be displayed in real time at the panel mostly on the computer CRT, and also recorded for later data reduction. The extent of software development for data reduction and graphics is to be determined.

4.14 STRUCTURES

The structural system provides the housing and rigid support necessary for stacking the major equipment and subsystem components of the Coal Feeder.

Requirements are as follows:

- o The structure will be capable of housing all the components of the 7 TPH coal feeder plant and will be within the envelope size of 50 ft x 50 ft x 130 ft height.
- o The structure will be designed to support not only the static loads of the components, but also the moving loads created by the rotating machinery.
- o The plant support-structure including platforms, railings, stair foundations, etc., will be designed and fabricated in accordance with the AISC structural and building code.
- o All equipment will be installed in accordance with the current commercial practices.
- o Structural loads will be provided at a later date.

- o Interface requirements for the coal feeder plant structures will be provided at a later date.
- o Site information for the plant as per customer requirements will be provided at a later date.
- o Rigid-frame construction will be used so that beam-to-column connectors have sufficient rigidity.
- o Dead loads assumed in the design will consist of the weight of steel structures and of all components that are permanently fastened to or supported by them.
- o Structural members subjected to both axial compression or tension and bending stresses will be proportioned to satisfy the AISC codes.
- o All connections and connected members will have the capacity to carry the full gravity loads and to resist the wind moments.

Section 5

SUMMARY AND CONCLUSIONS

This study has served to define subsystems for a complete coal pressurizing system based on the Kinetic Extruder. The most important conclusion is that, beyond the Kinetic Extruder Unit itself, all of the required auxiliary subsystems are standard commercially available equipment. No development work on the subsystems appears to be necessary, although durability and reliability in the proposed use should be verified.

As discussed in the Introduction, a broad view of the pressurizing system was taken in this study. Many of the subsystems (e.g., storage tanks, baghouse, inert gas generator, lift system) will be subject to special user requirements and may not form part of a commercial Kinetic Extruder package.

Our present view of the minimum subsystem group which should be included with the Kinetic Extruder unit is the following:

- o Feed hopper
- o Drive power
- o Coolant and lubrication
- o Rotor eye suction
- o Gas systems
- o Control and instrumentation

All of these subsystems' requirements are heavily influenced by the KE unit design, and thus they form a logical group.