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CONTINOUS HIGH PRESSURE LUMP COAL FEEDER DESIGN STUDY

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CONTINUOUS HIGH PRESSURE LUMP COAL FEEDER DESIGN STUDY

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The purpose of this project was to try to develop a continuous lump coal dry feeder for a pressurized fluidized bed combustor. The approach was to adapt the commercially available Fuller-Kinyon pump to feed coal against a pressure differential of 100 psi or more. The pump was modified and tests performed at various pressure differentials, with differently pitched screws, various screw rotational speeds, and various seal lengths and configurations.

Successful operation of the modified Fuller-Kinyon pump was generally limited to pressure differentials of 60 psi or less. Although the results of this project are not conclusive, test data and observations were made that indicated that higher pressure differentials could be attained by further modifications of the test setup. In particular, it is recommended that further testing be performed after replacing the 40-horsepower pump motor presently in the test setup with a motor having a significantly higher power rating (thereby allowing pump operation with longer seals and at higher pressure differentials than those tested so far).

PURPOSE OF PROJECT

In the process of fluidized bed combustion of coal, if the fluidized bed combustor is operated suitably pressurized, overall power generating efficiency can be increased by expansion of the hot flue gases through a gas turbine in a combined cycle plant. However, in this case, there is a need for equipment to continuously and reliably feed large quantities of coal and dolomite into the pressurized combustor.

A screw-type feeder such as that manufactured by the Fuller Company for the low pressure differentials used in pneumatic conveying systems in the cement industry (the Fuller-Kinyon pump) appears to offer potential for this application provided the pump can be modified and upgraded to meet system and process requirements. Thus the general purpose of this project has been to establish the feasibility of using a screw-type feeder to feed lump coal and limestone continuously into an appropriately pressurized vessel. Specifically, the possibility of adapting the existing design of the Fuller-Kinyon pump to this application has been investigated.

DESCRIPTION OF STANDARD FULLER-KINYON PUMP

The Fuller-Kinyon pump is used extensively in pneumatic conveying systems for injecting materials into pressurized pneumatic conveying lines. In normal applications, the materials conveyed are dry, pulverized, and free-flowing, and have a fineness of at least 100% passing 50 mesh, 75% passing 100 mesh, 60% passing 200 mesh, and 45% passing 325 mesh screens. The Fuller-Kinyon pump has operated with such materials at pressure differentials as high as 50 psi. It is currently available in , standard sizes with capacities ranging from 8 to 200 tons per ho··· (in terms of portland cement).

The Fuller-Kinyon pump is shown in Figure 1 (cut-away view) and Figure 2 (general assembly drawing - exploded view). The construction and operation of the pump for pneumatic conveying system applications are briefly summarized in the following (items followed by numbers in parentheses are correspondingly called out in Figure 1).

> All parts are mounted on a cast iron base (1). The material to be conveyed enters the hopper (2) by gravity from the sources of supply, and is advanced through the barrel (3) by the impeller screw (4), the latter being directly driven through a flexible coupling connected to the driving motor. (The barrel is protected by renewable wear-resistant liners, and the screw flights are also protected with a special alloy to give a maximum of service.) As material advances through the barrel, it is compacted by the decreasing pitch of the impeller screw flights, and is further increased in density by the space or "seal" between the terminal flight of the impeller screw and face of the check valve disc (5). The exact density required is further controlled by an adjustment of the seal length by means of the screw jacks (6). (When there is no material in the hopper, the check valve prevents the flow of gas back into the barrel and screw.) The material then enters the check valve body or mixing chamber (7) wherein it is fluidized by compressed air introduced through a series of air jets (8) and from there enters the transport line.
>
> The pump impeller screw is positioned in a hollow shaft (9) which is supported by ball bearings (10 and 11) in a single bearing housing (12). The supporting shaft is rotated through an eccentric lock collar (13). There are no packing glands. The material in the hopper is sealed from the bearings when the pump is in operation and under pressure by means of a Graphitar air-cooled, fan-type seal ring (15), in the chamber (16). The seal ring is kept clear of material from the hopper by means of compressed air supplied through air piping (17) from the header (18). Finally, the hopper section has a catch basin on the bottom and a clean-out door (19) on each side.

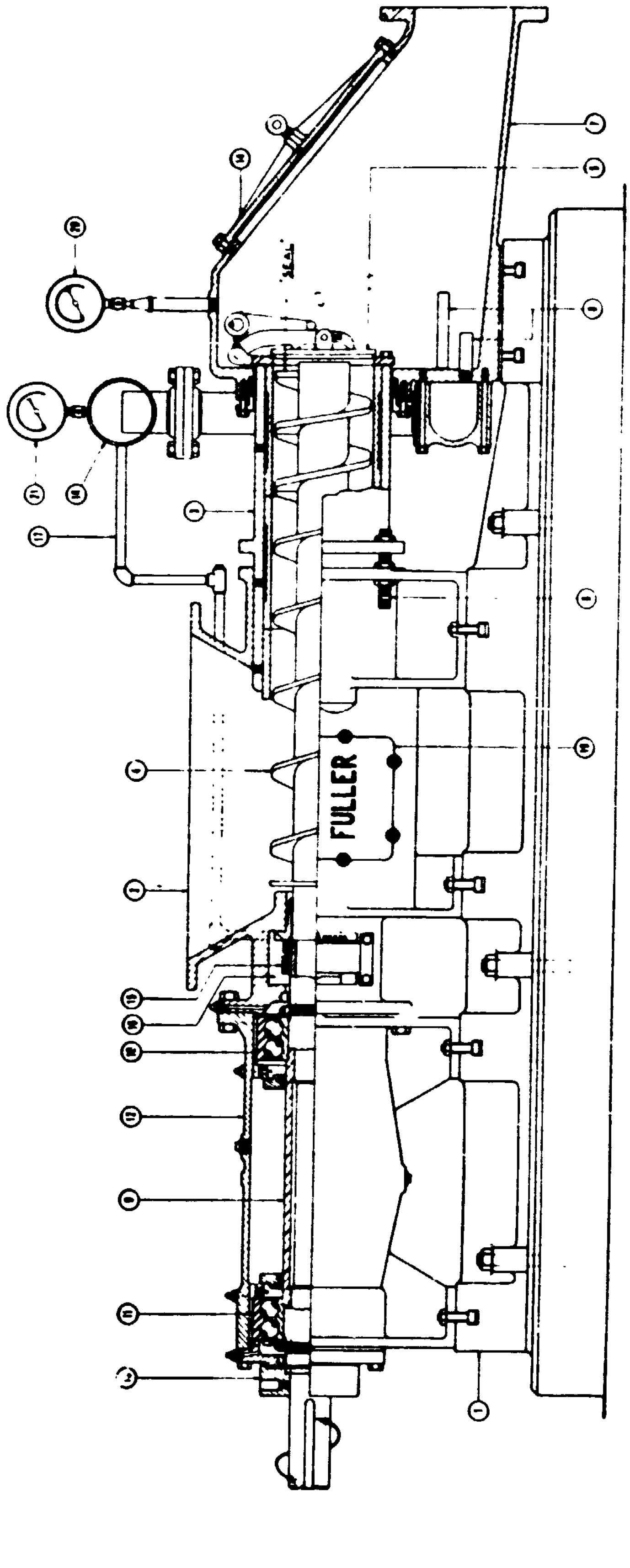

Figure 1 Fuller-Kinyon Pump

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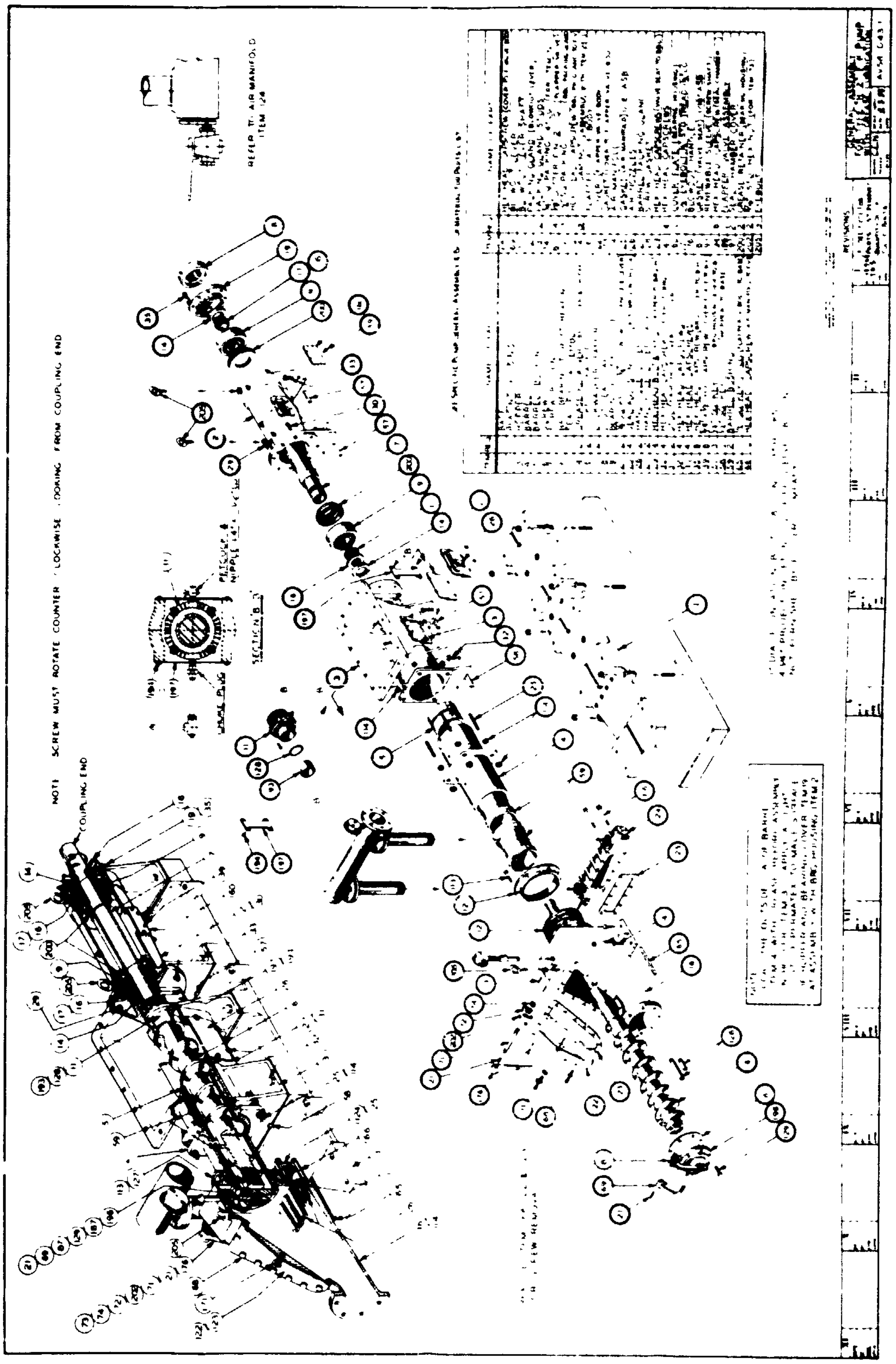


Figure 2 General Assembly for Type "H-2" Fuller-Kinyon Pump

DESCRIPTION OF THE DESIGN CONCEPT INVESTIGATED

The major modification to the Fuller-Kinyon pump investigated during this project involved: 1) removal of the check valve, the mixing chamber, and the air jets from the output end of the pump, and 2) extension of the barrel (in various configurations) beyond the terminating flight of the screw to provide additional length of extruded material for sealing purposes.

The object of this modification was to preserve the material transfer capability of the Fuller-Kinyon pump while at the same time obtaining, for materials much coarser than normally handled by the pump, a sufficient length of extruded material to provide, together with the material contained in the screw, an acceptable and reliable seal against pressure differentials much higher than those normally imposed on the pump. It appeared that with this modification other alterations in the existing design of the Fuller-Kinyon pump might not be required.

BACKGROUND WORK

Prior to testing of the design concept, considerable effort was expended: 1) analyzing the available background data on the Fuller-Kinyon pump to determine scaling relationships and the effects on pump performance of variation of such parameters as type of feed material, screw rotational speed, screw pitch, and screw compression ratio; 2) analyzing the mechanics associated with forced motion of a slug of granular material in a tube (as it relates to the concept of screw-type feeder operation involving downstream pressure sealing by a continuously extruded slug of feed material); and 3) making estimates of the various types of power consumption associated with the Fuller-Kinyon pump. (All of these analytical efforts are summarized in detail in Reference 1.)

Item 1 was aimed primarily at providing a basis for defining the tests to be conducted later in the project with a modified Fuller-Kinyon pump. This analysis led to the following conclusions. First, constant-pitch screws should be used in order to avoid possible material compression problems due to feed material non-uniformity and poor compaction characteristics. Second, values of the initial screw helix angle (the angle whose tangent is the ratio of the initial screw pitch to π times the screw outer diameter) should be chosen which appeared likely, on the basis of available data, to maximize pump volumetric efficiency for the feed materials of interest. Third, tests should be run over a significant range of obtainable screw rotational speed in order to establish an output-versus-rpm characteristic curve for the feed materials of interest. (This data would of course be of particular interest when considering applications requiring a variable feed rate capability.)

Item 2 was directed toward aiding in the design of seal configurations which could provide the required seal without generating excessive loads on the pump. Based on this analysis, it was concluded that it would be most advantageous to conduct tests with a seal enclosure first consisting of a tube of constant cross-section (having a length somewhat less than the critical length for jamming) and then perhaps two somewhat different divergent, truncated cones.

Item 3 was intended to provide a basis for estimating power consumption with the Fuller-Kinyon pump for the application under investigation. Estimates were made for the pump of the maximum rate of doing work: 1) against a pressure differential, 2) in imparting axial motion to the feed material, 3) in imparting rotational motion to the feed material, and 4) associated with feed material shear at the terminating flight of the screw. (No method was established to estimate the power consumed by friction occurring within the barrel of the pump.) These estimates indicated a power requirement for the 4-inch diameter Fuller-Kinyon pump (excluding the power consumed by friction occurring within the barrel of the pump) of up to about 30 horsepower for operation with an effective seal against a gas pressure of 100 psig.

Certain required materials properties tests were also performed. Some of these were required to complement the analytical results. For example, the analysis under item 2 required a determination of the length of feed material moving in a smooth pipe at which the phenomenon of jamming occurs. (This defines the maximum obtainable seal length for feed material in a smooth pipe.) Other tests were required to determine, for example, leak rate as a function of the length of feed material in a column and the pressure differential across the column. (These and other materials properties tests are summarized in detail in Reference 1.)

Three basic conclusions were made based on the results of these tests. First, only very limited compaction of the materials of interest (corresponding to a 12% reduction in volume) can occur before the onset of actual mechanical compression. Second, reasonably long material slug lengths can be moved in tubes of constant circular cross-section. For example, "free" movement of slugs of material up to a length-to-diameter ratio of approximately 7 can be expected for a mixture of typical moist coal and limestone (4:1 by weight), and "free" movement of considerably longer slugs of such material can be realized in slightly divergent truncated cones. Third, relatively long material slug lengths appear to be required to obtain sufficient sealing against the pressure differentials of interest on this project. (It should be noted here that, for the materials tested, slugs of the dry materials jammed at shorter length-to-diameter ratios but required less length to achieve a given leak rate than did slugs of the moist materials.)

With respect to the third conclusion, the actual slug length (of extruded feed material) required with a modified Fuller-Kinyon pump is of course highly dependent on the amount of feed material contained within (and the degree of "filling" in) the pump barrel and screw, the compaction of the feed material as it advances along the screw and in the seal, and the feed material velocity through the pump and seal. Thus, the effectiveness of the seal provided by a Fuller-Kinyon pump operating with an extruded feed material seal can only be accurately determined by testing with the pump itself.

FULLER-KINYON PUMP TESTS

In order to develop the design concept, establish its performance characteristics, and evaluate it in terms of feasibility, approximately 150 tests were performed during this project with a modified 4-inch diameter Fuller-Kinyon pump. The 4 in diameter Fuller-Kinyon pump, which is the smallest such pump marketed, was chosen primarily to satisfy restrictions in the laboratory with regard to power availability, receiver pressure vessel size (as determined by pump output and desired test time), available coal storage space for the test program, and equipment availability. In pneumatic conveying system applications, this pump has a rated capacity of up to 8 tons per hour for Type 1 cement. It was felt that test results obtained with this pump could be scaled to the 100-ton-per-hour range with reasonable accuracy.

Tests were conducted primarily with three essentially non-compressing screws. One was a 4-1/4" constant-pitch screw (referred to as a 4-1/4" x 4-1/4" pitch screw). The second was a 3-1/4" x 3" pitch screw which offered a significantly reduced screw helix angle in comparison with that of the first screw. (The pitch of all flights on this screw was 3-1/4 inches with the exception of a 3-inch pitch for the last flight.) The third was a 1-1/2" constant-pitch screw (referred to as a 1-1/2" x 1-1/2" pitch screw). Otherwise, the three screws were similar, having an overall length of the flights of 27-3/8 inches, a flight thickness of 1/2 inch, and a shaft diameter of 2-1/8 inches.

The first two screws were slightly modified for the test program with the addition of a "terminal taper" at the output end of the screw to prevent the formation of a void region in the extruded feed material seal. For each screw, the terminal taper consisted of a cone which was welded to the output end of the screw shaft. Each cone was 4 inches in height and had a 2-1/8-inch diameter base.

Seal enclosure configurations tested included straight-pipe sections, two divergent truncated cones, and straight-pipe sections combined with these cones.

Tests were conducted at screw rotational speeds between 1160 rpm (the normal operating speed for the Fuller-Kinyon pump) and 570 rpm. Primarily for convenience, -1/4" coal was, in general, used as the feed material. However, some tests were also conducted with -1/4" coal mixed with -1/4", +1/32" limestone (20% by weight). Pressure differentials ranged from zero to 90 psi.

The Fuller-Kinyon pump laboratory test setup is best described by the schematic drawing shown in Figure 3 and the photographs of the test setup which follow (Figure 4). A close-up of one of the seal enclosures (test sections) installed in the test setup is shown in Figure 5.

General test procedure began with pneumatic conveying of the feed material to the storage bin suspended from a load cell above the modified 4-inch diameter Fuller-Kinyon pump. For each test, enough feed material was placed in the storage bin to maintain a sufficient feed material head above the pump for a continuous uniform supply to the screw throughout the test. At the beginning of a test, the 40-horsepower pump motor was started, and once the

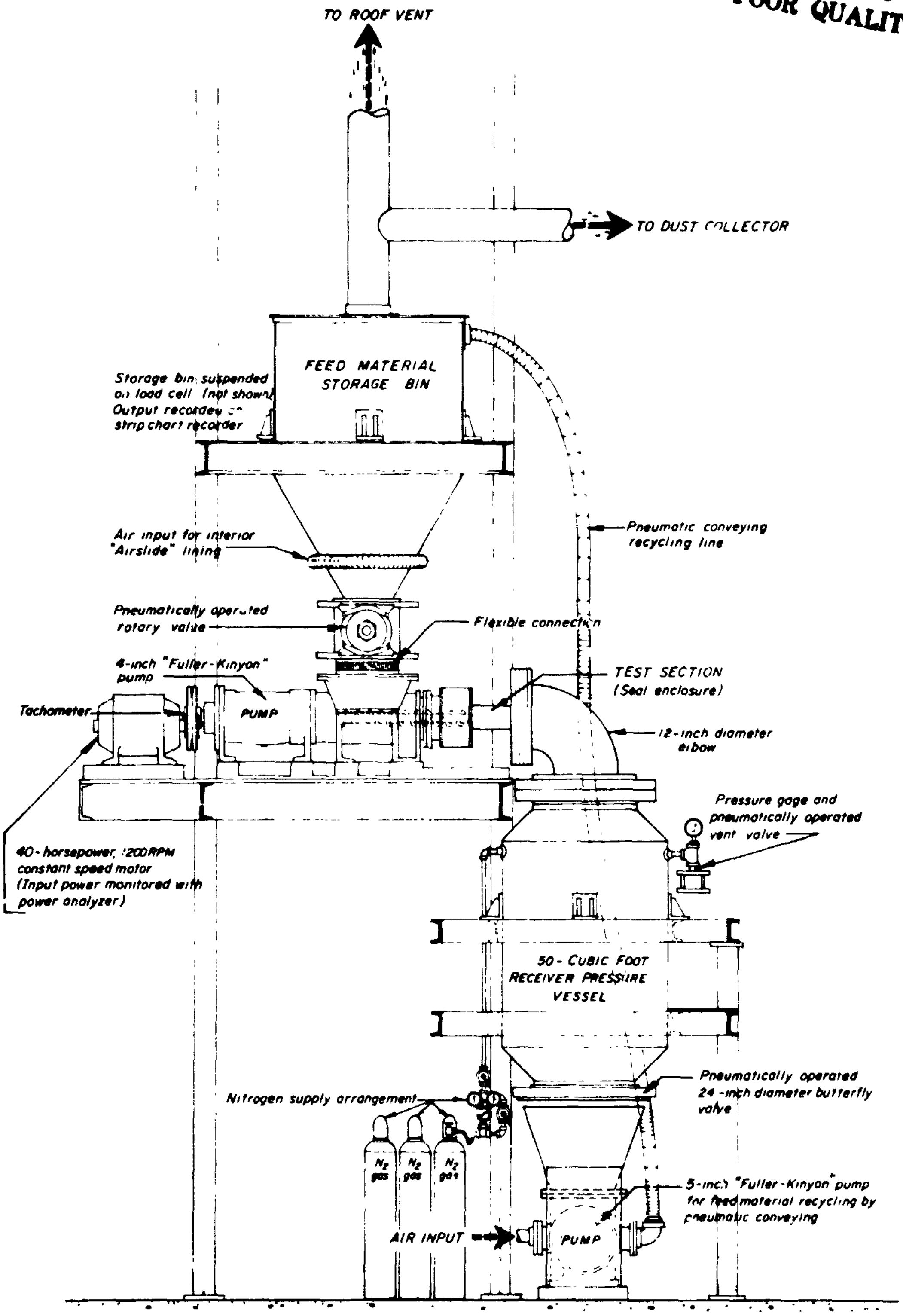


Figure 3 Fuller-Kinyon Pump Laboratory Test Setup Schematic

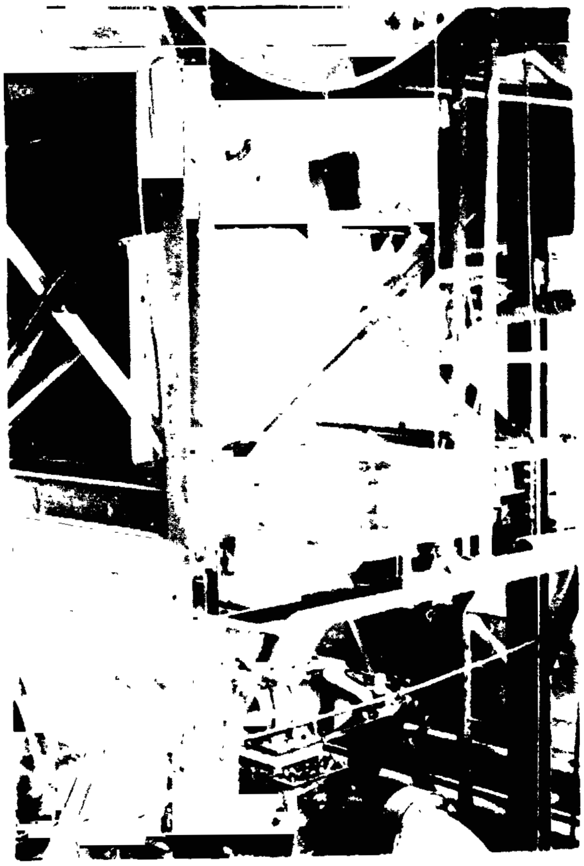
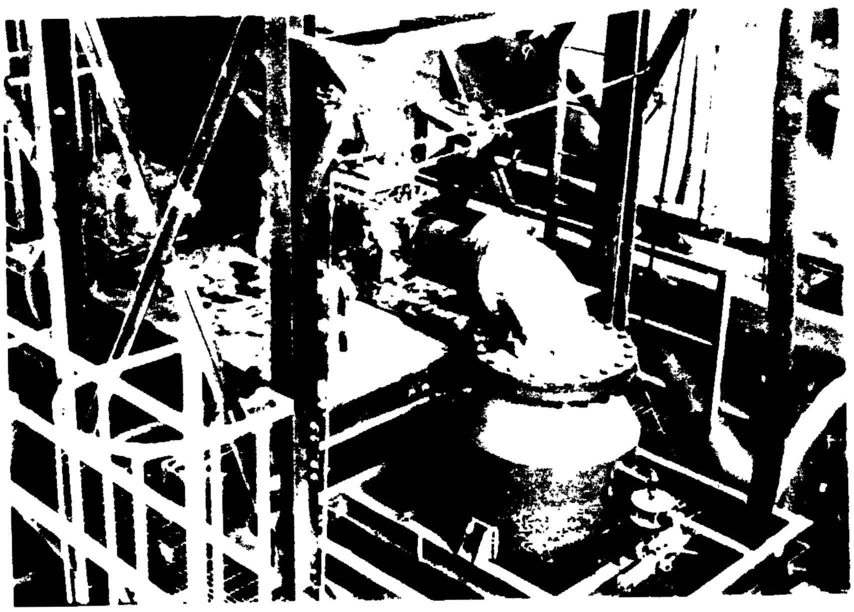


Figure 4 Fuller-Kinyon Pump Laboratory Test Setup:

a) Feed Material Storage Bin and Pneumatically Operated Rotary Valve (opposite)
b) Rotary Valve, 4-Inch Fuller-Kinyon Pump and Motor, Test Section, and Top of Receiver Pressure Vessel (below)

(Figure 4a)



(Figure 4b)

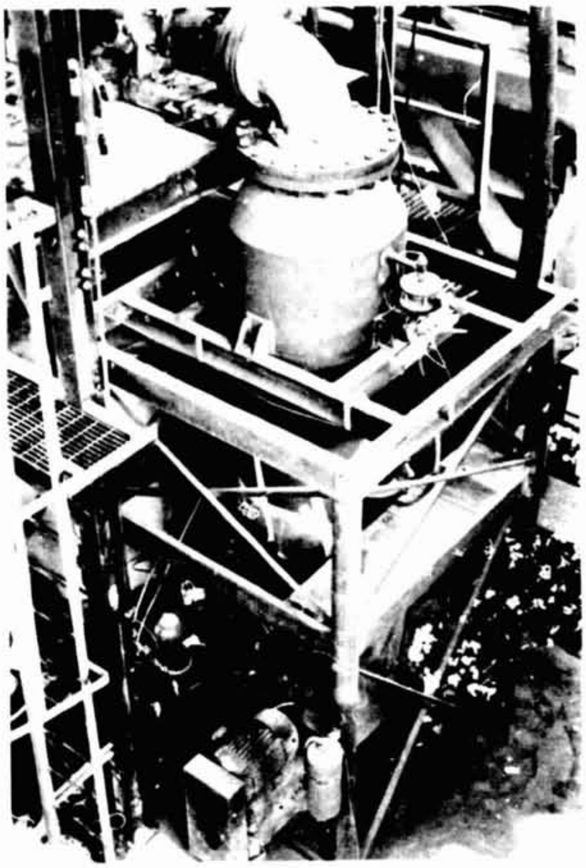


Figure 4 Fuller-Kinyon Pump Laboratory Test Setup:

c) Receiver Pressure Vessel and Compact Fuller-Kinyon Pump for Feed Material Recycling by Pneumatic Conveying

(Figure 4c)

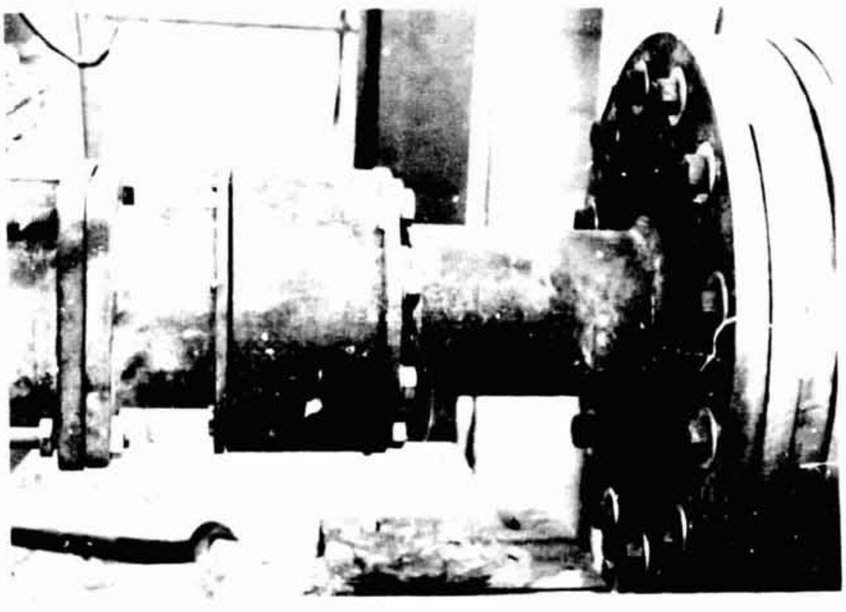


Figure 5 Divergent Truncated Cone Test Section Number 1 Installed

selected rotational speed had been reached, the rotary valve isolating the feed material storage bin from the pump was then opened, thus allowing feed material to enter the pump and test section (seal enclosure). Feed material which had been extruded through the system would then accumulate in the receiver vessel.

For tests with non-zero pressure differentials, the receiver vessel was maintained closed and was supplied with a sufficient quantity of inert gas at the start of testing to pressurize the vessel to the desired level. In general, pressurization was performed as quickly as possible after establishment of a steady rate of feed material transfer through the system. (Significant additional pressurization was seldom required to maintain the desired pressure level, with the exception, of course, of those tests during which the seal was not successfully maintained.)

Upon completion of a test and shutdown of the pump, the receiver vessel was vented, if required, and the accumulated feed material was removed through a large butterfly valve at the bottom of the vessel (the feed material would either be scrapped or recycled). At this time, samples were often taken to determine the feed material size distribution and/or moisture content.

In the event of seal loss and rapid depressurization of the receiver vessel (blow-back) during tests with pressure differentials, venting of the system occurred through the pump and feed material storage bin to a dust collector, and for short periods, to a vent in the roof of the laboratory. The dust collector was also used during pneumatic conveying of the feed material to the storage bin prior to testing.

The feed material transfer rate (feed rate) through the system was determined by continuously recording the output from the load cell on a strip chart recorder. The input power to the 40-horsepower pump motor was monitored by observing readings on a power analyzer, while the screw rotational speed was monitored with a tachometer. (Tachometer readings were displayed in digital form on a meter at floor level.) The pressure differential across the pump and seal was monitored by observing readings on a pressure gage mounted on the receiver vessel.

PRELIMINARY TESTS

Preliminary tests with zero gas pressure differential across the pump were conducted with both the 4-1/4" x 4-1/4" pitch screw and the 3-1/4" x 3" pitch screw operating at each of two rotational speeds (1160 rpm and 570 rpm). These tests were conducted with various lengths of smooth straight pipe downstream of the pump to form the seal enclosure for the extruded feed material. Seal lengths tested ranged up to the maximum values which could be obtained with the test setup.

The results of the preliminary tests led to the following observations (a detailed presentation of results can be found in Reference 1):

1) For relatively "new" -1/4" coal and the range of moisture contents observed (less than 5%), the maximum feasible seal length is about 10 inches for both screws and both screw rotational speeds. (This length is considerably shorter than what one might expect on the basis of the results of the materials properties tests.)
2) For both screws and both screw rotational speeds, pump power consumption is relatively similar and in general, varies only moderately with seal length for lengths less than 10 inches (at this length, pump power consumption generally tends to rise rapidly beyond the 40-horsepower rating of the pump motor, necessitating shutdown).
3) The coal feed rate (weight per time) appears to be basically a function of feed material density. Increased moisture content (again in the range of observed values) is associated with decreased feed material density and decreased feed rates. To a lesser extent, coal recycling is associated with increased feed material density (through the production of finer size distribution) and increased feed rates. It should be noted here that some of the effect of moisture content on feed rate may be attributable to bridging of the coal in the feed hopper of the Fuller-Kinyon pump and resultant restriction in the supply of feed material to the pump itself.
4) For the <u>same</u> feed material, there appears to be little difference between the two screws in terms of the feed rate to be expected at the two screw rotational speeds and various seal lengths.

5) For the same feed material, the feed rate at various seal lengths is somewhat higher for both screws at a screw rotational speed of 1160 rpm.

TESTS WITH PRESSURE DIFFERENTIALS*

Since the data from the preliminary tests provided little if any differentiation between the two screws tested, the 3-1/4" x 3" pitch screw was chosen for tests with gas pressure differentials across the pump (at least initially), primarily because its smaller pitch and larger number of screw flights should enhance whatever pressure sealing effects occur within the screw itself.

Initial tests were conducted at screw rotational speeds of 1160 rpm and 570 rpm with extruded material seals formed in various lengths (9 inches or less) of straight 4-inch diameter pipe downstream of the 4-inch Fuller-Kinyon pump. These tests were conducted with a coal/limestone mixture and were limited by higher than expected pump power consumption to pressure differentials less than 15 psi.

Divergent truncated cone test section number 1 was then designed and fabricated for use in place of the straight pipe as a seal enclosure (see Figure 6 and 7). Tests with this test section were limited by pump power consumption to pressure differentials less than 10 psi for a screw rotational speed of 1160 rpm. (Tests were performed with both the coal/limestone mixture and also -1/4" coal.) However, the comparatively high feed rates observed at zero pressure differential indicated that the test section itself introduced very low resistance to the feed material extrusion process.

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* A detailed presentation of results can be found in Reference 1, with the exception of the results for the 1-1/2" x 1-1/2" pitch screw, which are presented in Reference 2.

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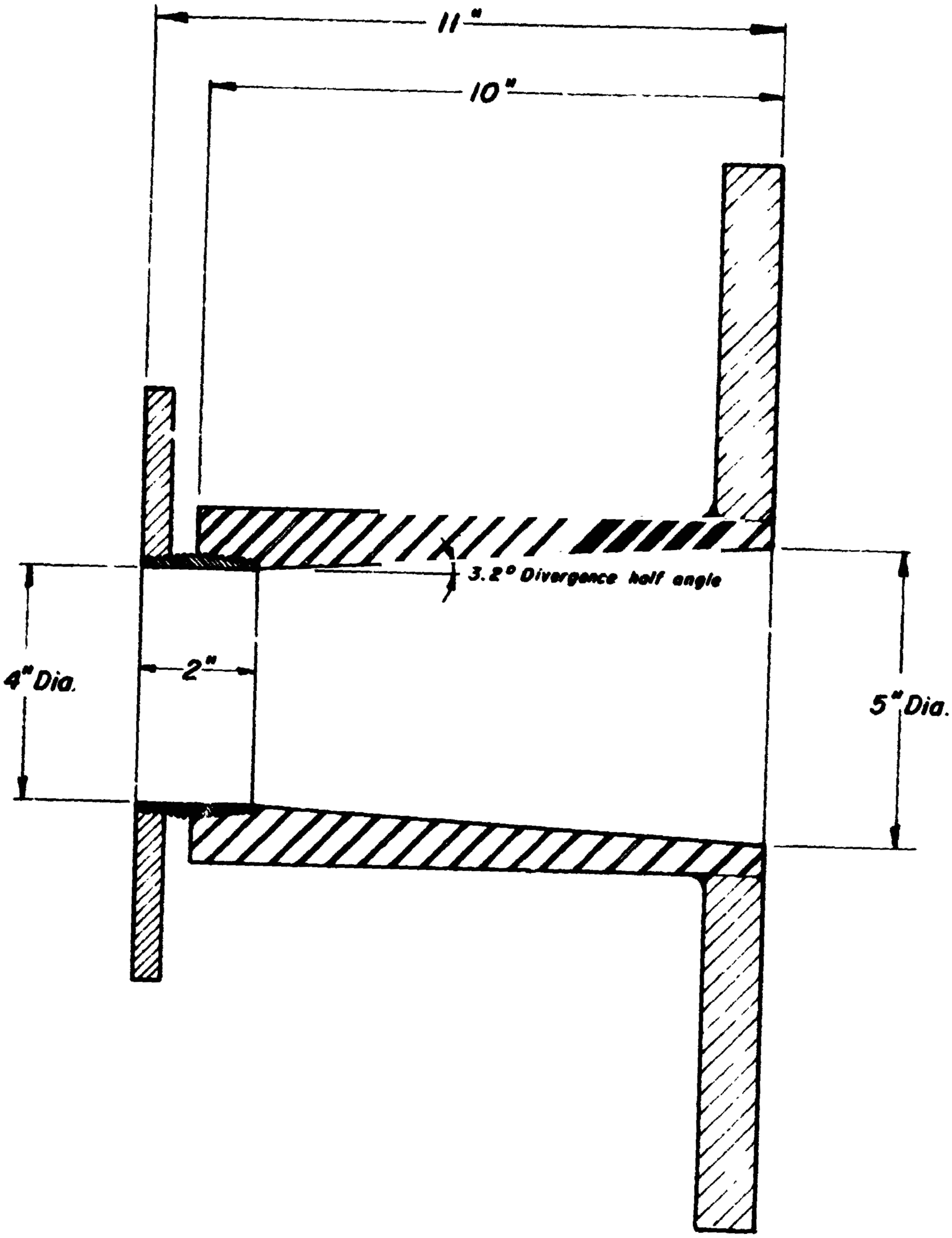

Figure 6   Divergent Truncated Cone Test Section Number 1

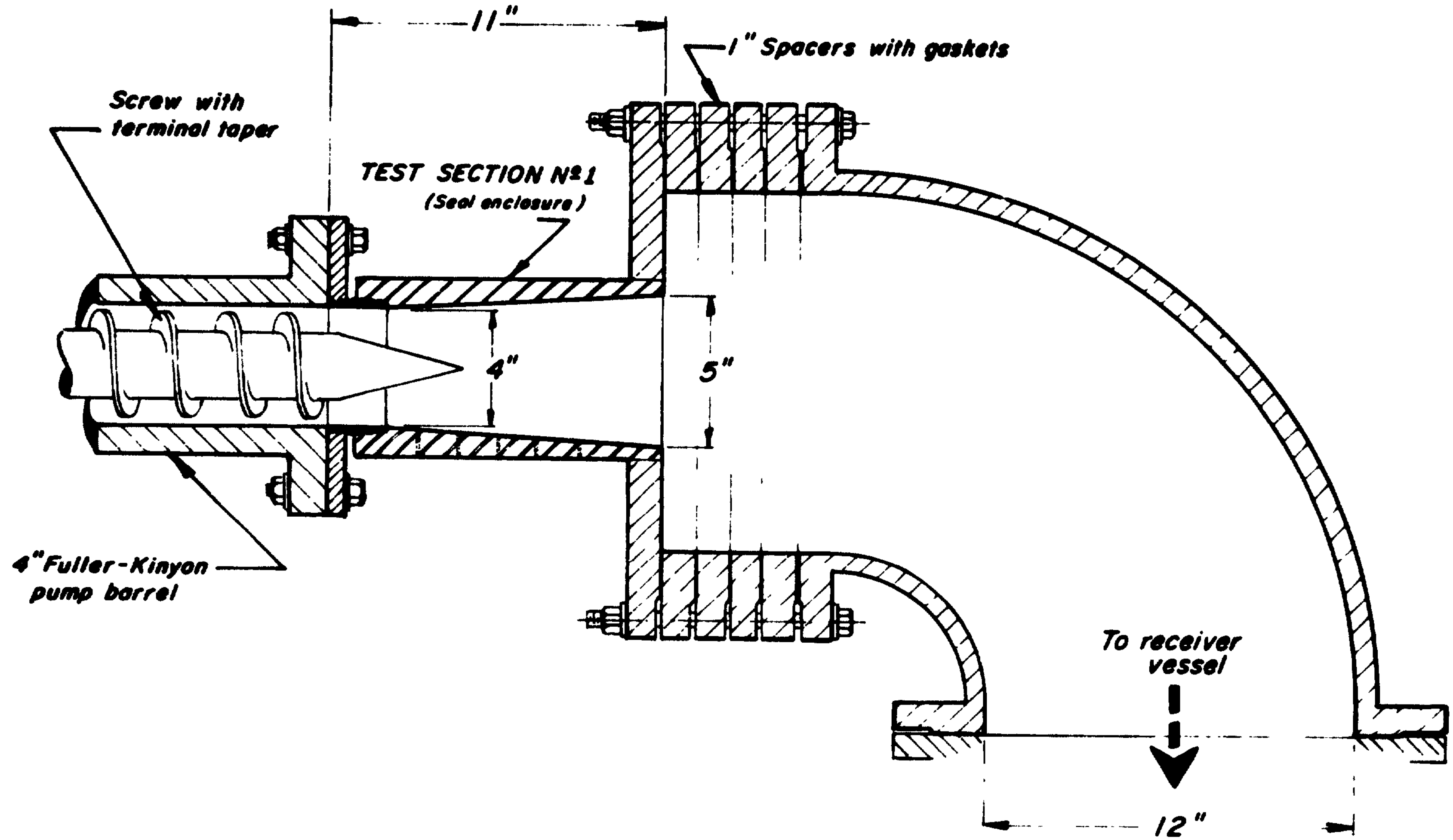

Figure 7 Installed Configuration - Divergent Truncated Cone Test Section Number 1

Further tests (again with test section number 1) were conducted at a screw rotational speed of 570 rpm. These tests were limited by pump power consumption to pressure differentials less than 40 psi in one sequence, 35 psi in another, and somewhat less than 70 psi in a third (in this case blow-back through the pump became a frequent occurrence). Higher than expected pump power consumption was once again observed during these tests together with a dramatic decline in feed rate with increasing pressure differential.

After these tests, the test setup was partially disassembled for inspection. Significant wear was noted on the terminating flight of the 3-1/4" x 3" pitch screw but not in the barrel liner to the pump or in the divergent truncated cone test section. (The screw wear noted is shown in Figure 8.) This wear was not entirely surprising in light of the fact that the flights of this screw did not have the normal hard-facing. While such wear had not been found on the 4-1/4" x 4-1/4" pitch screw (also without hard-facing), it had also not been used for tests with pressure differentials.

It looked as if the worn profile of the terminating flight of the screw could have led to significant wedging of feed material against the barrel liner and wall of the test section and also could have been conducive to blow-back. For this reason, it was decided to continue testing with the 4-1/4" x 4-1/4" pitch screw while at the same time rebuilding and hard-facing the 3-1/4" x 3" pitch screw for further testing later on.

Ten more tests were then conducted, 4 tests with the 4-1/4" x 4-1/4" pitch screw and 6 tests with the 3-1/4" x 3" pitch screw after it had been rebuilt and hard-faced. Results were very disappointing: tests with both screws were limited to pressure differentials of approximately 30 psi or less, pump power consumption was observed to rise rapidly and almost linearly with increasing pressure differential, and significant heating of the pump housing (barrel) and test section occurred in the vicinity of the terminating flight of each screw. This heating was not significantly affected when, for some of these tests, the barrel was extended to obtain an additional one-inch straight-pipe length between the terminating screw flight and the entrance to the divergent truncated cone test section. Inspection of the 4-1/4" x 4-1/4" pitch screw after testing indicated the beginning of the type of wear previously observed for the 3-1/4" x 3" pitch screw. The latter screw showed no wear after testing, presumably as a result of the hard-facing.

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Figure 8 Sequence of Observations Made After Test Number 79 of Wear on Terminating Flight of 3-1/4" x 3" Pitch Screw

(Note: When viewed from tip, screw rotation is clockwise in this sequence.)

The conclusion of all this was basically that the worn-down version of the 3-1/4" x 3" pitch screw had apparently permitted successful tests at higher pressure differentials. A terminating flight similar to that of a wood screw apparently led to performance superior to that obtained with the standard design.

Because of the evidence of significant loading being transmitted to the terminating flight of the screw during tests involving pressure differentials, two approaches were then taken to alleviate the axial loading transmitted to the screw as a result of seal enclosure wall friction. The first approach involved conducting a number of tests with a modified version of the divergent truncated cone test section used previously. Holes were tapped into the wall of the test section in order to permit jetting of nitrogen into the test section at various positions along its length during testing (see Figure 9). It was thought that the effective cone length could be reduced through downstream gas "lubrication" at the wall and partial fluidization of the extruded feed material. Tests with this arrangement were marginally successful in that some increase was noted in the pressure differentials which could be handled by the system. However, feed rates were observed to decrease significantly with these increasing pressure differentials.

The second approach involved testing with a new divergent truncated cone test section (test section number 2) having a cone angle of about 24° in contrast with the 6° angle of the previous test section. Test section number 2 is shown in Figures 10 and 11.

Results were basically similar to those for the previous test section (test section number 1) in the sense that full pump power consumption was reached at a pressure differential of about 35 psi (again for the 3-1/4" x 3" pitch screw operating at a rotational speed of 570 rpm). Once again a very linear increase in power consumption was noted with increasing pressure differential. The most interesting result was the observation that, for pressure differentials between 35 psi and 40 psi, pump power consumption <u>dropped</u> by about 50% with no apparent change in material flow rate or smoothness of operation. Blow-back occurred on both of these tests at a pressure differential slightly above 40 psi.

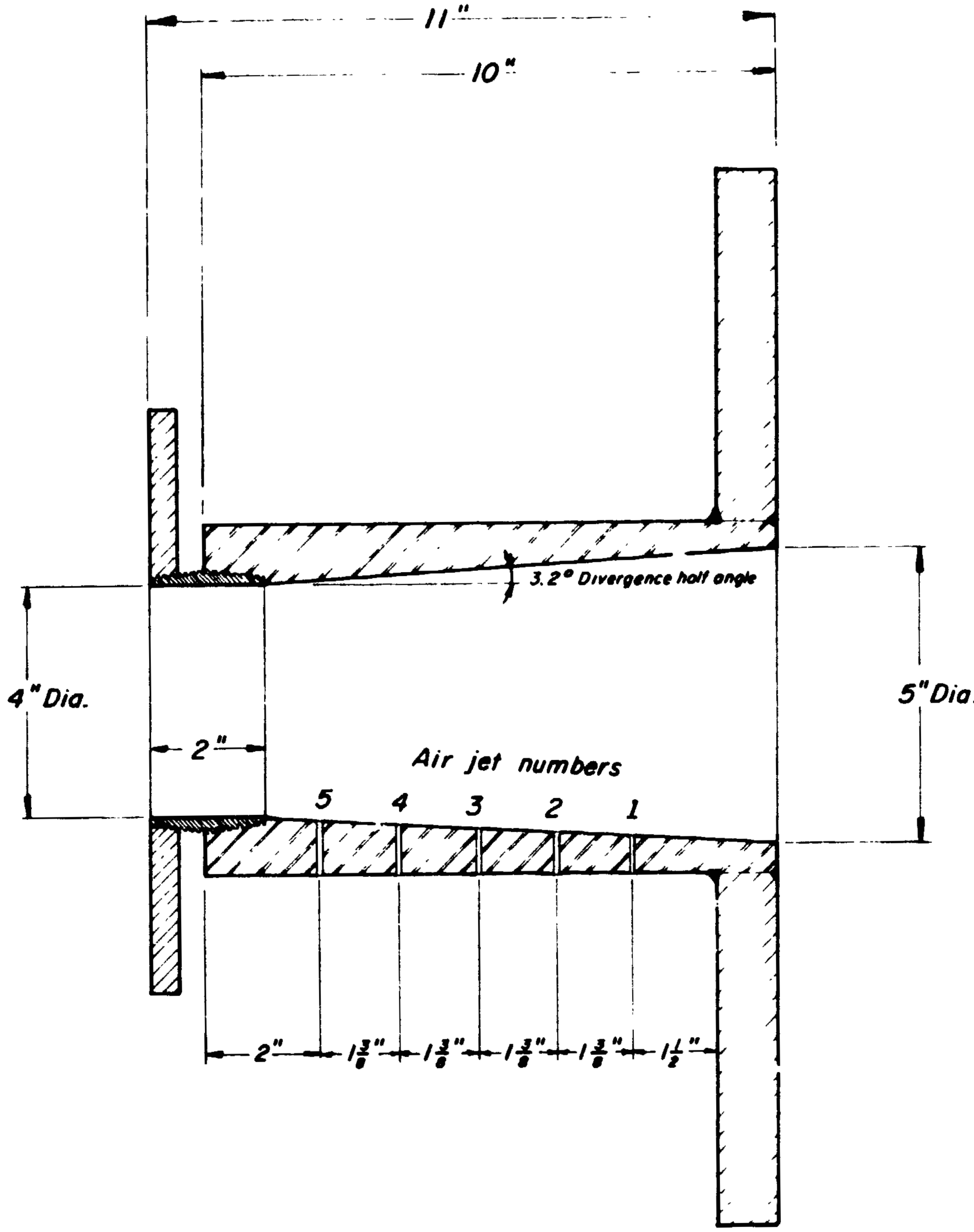

Figure 9 Divergent Truncated Cone Test Section Number 1 - Air Jet Locations for Test Numbers 90-106

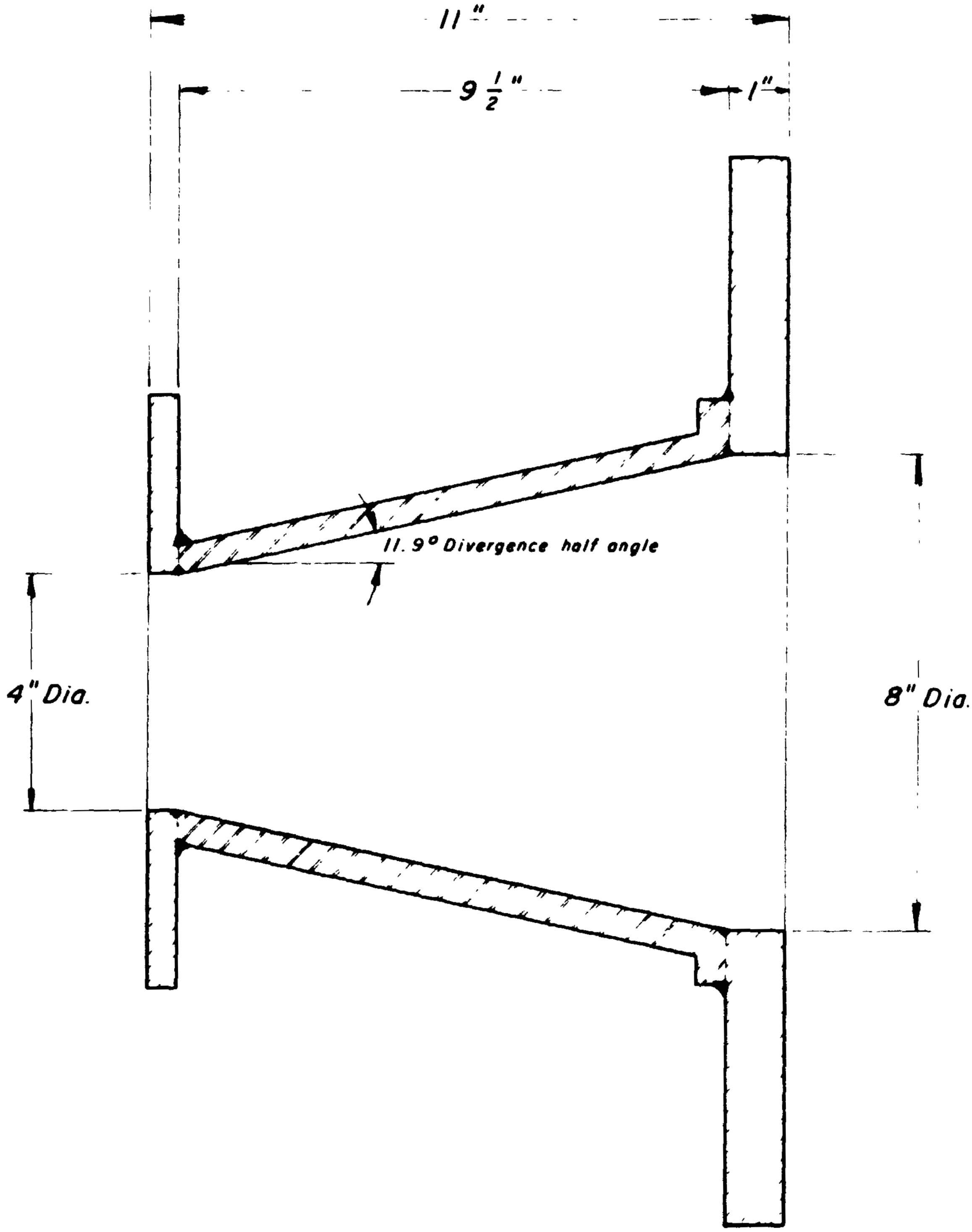

Figure 10 Divergent Truncated Cone Test Section Number 2

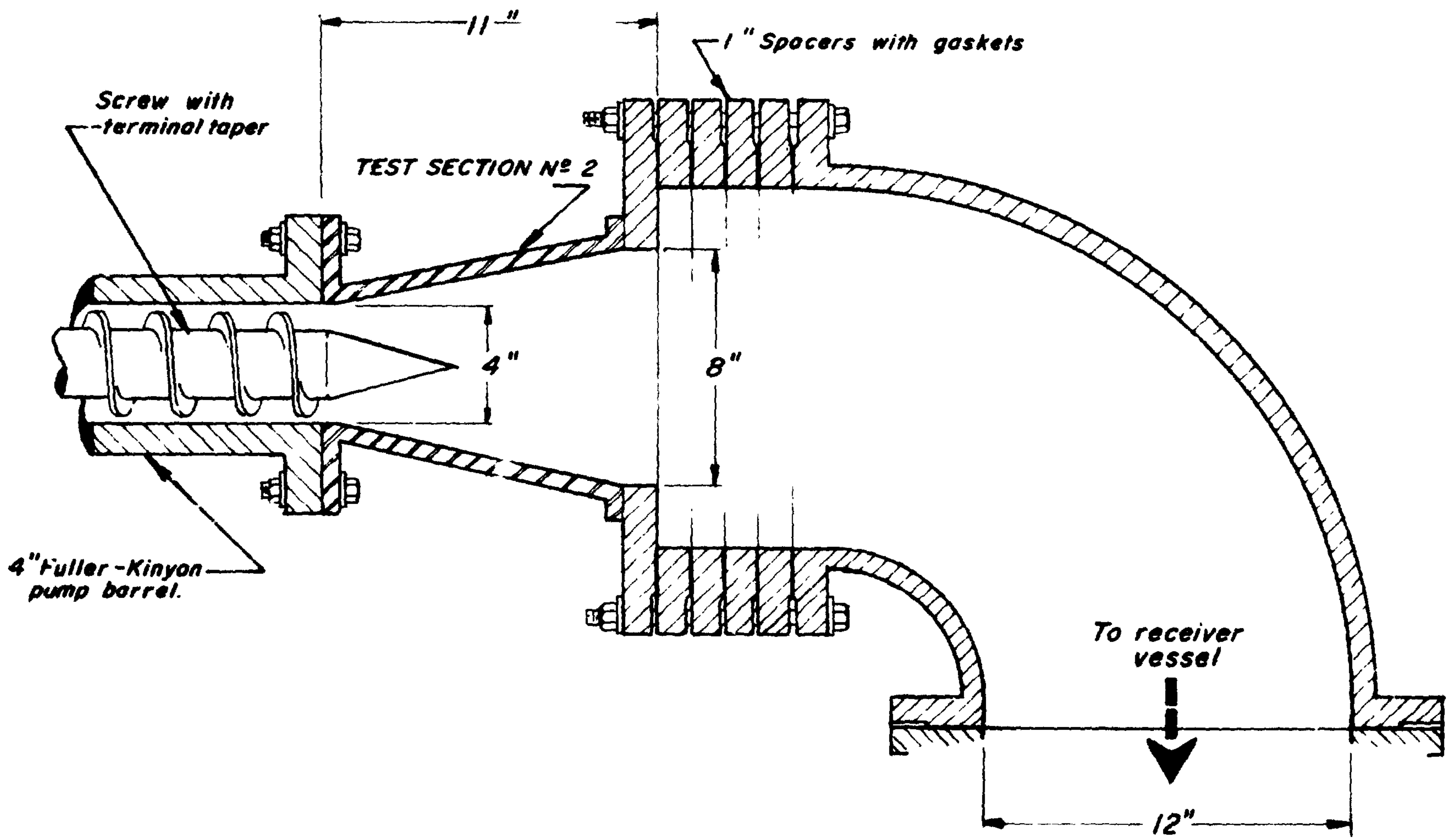

Figure 11 Installed Configuration - Divergent Truncated Cone Test Section Number 2

The observed drop in power consumption appeared to indicate that leakage back through the screw had led to fluidization of the feed material in the hopper and the pump barrel, thus reducing power consumpt, n in this region by reducing material wall friction. The fact that blow-back was observed at a pressure differential slightly above 40 psi tended to support the possibility that a precariously balanced fluidization situation could have existed at slightly reduced pressure differentials.

At this point the tests were stopped for evaluation of the results obtained, prior to further testing. Primarily in an effort to reduce power requirements and at the same time increase the effective seal which could be obtained, two major changes were decided upon. First the pump's normally smooth barrel liner was replaced with a fluted barrel liner (illustrated in Figure 12). It was theorized that this fluted barrel liner would help to constrain the feed material from rotating while enhancing its axial movement, thus increasing the efficiency of the pump. Second, a significantly smaller pitch screw was fabricated, namely a 1-1/2" x 1-1/2" constant-pitch screw. While the displacement rate for this screw would be significantly less than that of the screws previously tested, increased displacement efficiency was expected due to the reduced helix angle for this screw. Furthermore, the potential for blow-back was expected to be reduced due to the increased number of flights for this screw.

The combination of this screw and the fluted barrel liner was used throughout the remainder of the tests. These tests were also conducted with newly obtained coal having much higher moisture contents than the coal used for previous tests. Moisture contents generally varied from 9% to 13%, in comparison with previous moisture contents which were less than 5%.

Tests were first performed with both divergent truncated cone test sections and with straight-pipe sections of various lengths. In comparison with the results obtained previously, these tests yielded relatively similar results (in all respects) for the two divergent truncated cone test sections and somewhat improved results for straight-pipe seals. With respect to the latter, successful operation with a seal length of 10 inches was achieved up to pressure differentials of about 70 psig, at which blow-back became a limiting factor (close to maximum pump power was also required at this pressure differential).

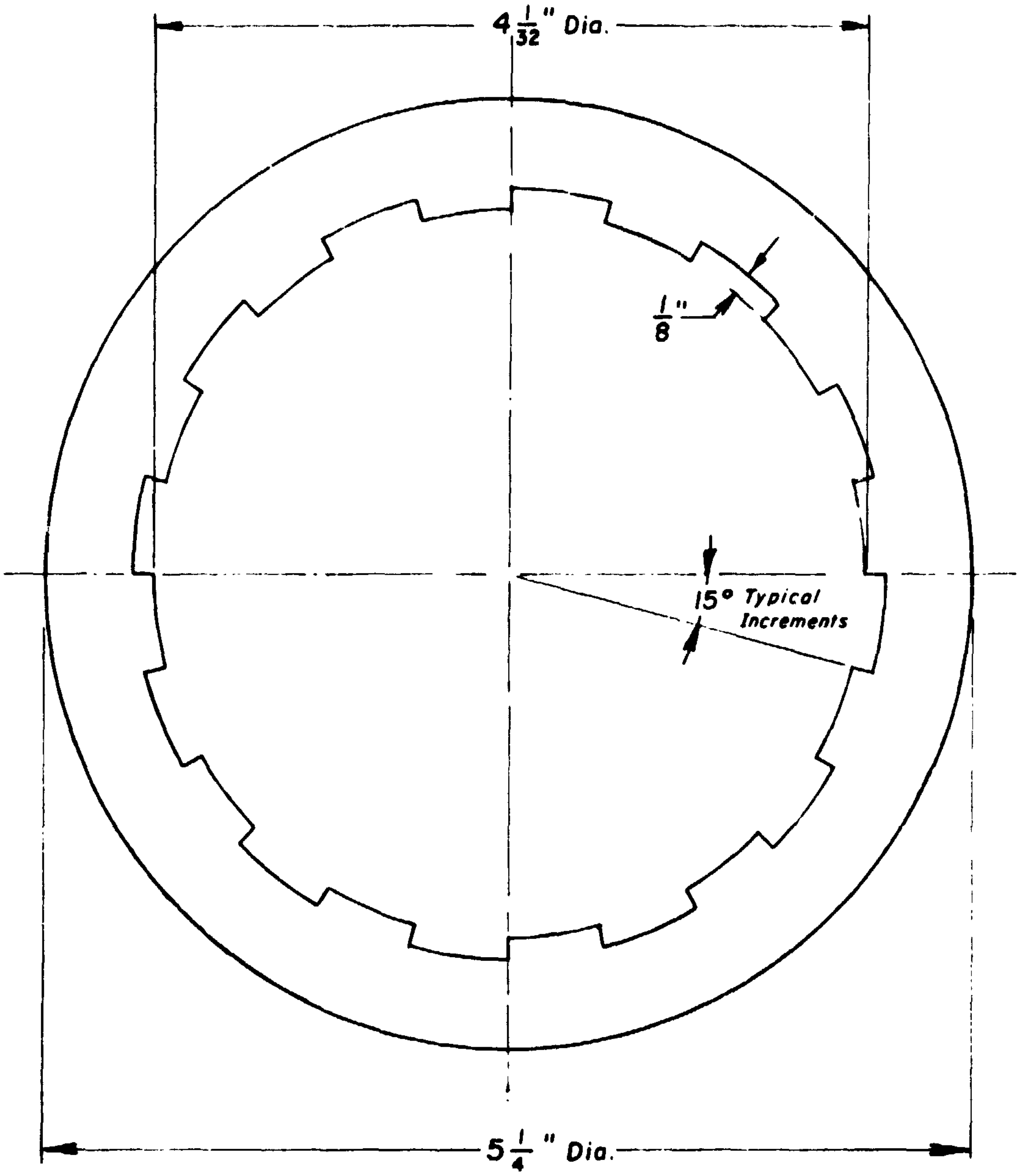

Figure 12 Fluted Barrel Liner

In an effort to extend the seal length beyond 10 inches, final tests were then conducted with seal enclosures consisting of combinations of various lengths of straight pipe and one or the other of the divergent truncated cones. (Feed material would travel through the straight pipe into one of the cones into the receiver vessel.) The majority of these tests were performed with divergent truncated cone test section number 1 and lengths of straight pipe varying from 6 to 7-1/2 to 9 inches. (All of these tests were performed at a screw rotational speed of 570 rpm.)

For the 6 and 7-1/2-inch lengths, tests were limited by blow-back to pressure differentials of approximately 40 psig and 55 psig, respectively. (Pump power requirements, while observed to increase with increasing pressure differential, remained relatively low.) For the 9-inch length (the last test performed), successful pump operation occurred for pressure differentials up to 90 psi, at which the feed rate through the pump began to surge somewhat, indicating that blow-back was most likely imminent. Once again, pump power requirements, while observed to increase with increasing pressure differential, remained relatively low, reaching 33 horsepower at 90 psi. In addition to this result, the feed rate for the pump remained relatively constant throughout the course of this test (while pressure differentials were increased) and at a value close to that expected for very low pressure differentials.

This last result was considered most important since it appeared to indicate that the substantial overall seal length utilized (19 inches) had made it possible to counteract the phenomenon of decreasing feed rate with increasing pressure differential noted during previous testing. However, since the 90-psi pressure differential reached also appeared to be close to producing blow-back with this seal and close to requiring pump power in excess of that available, it also appeared that further tests at pressure differentials of this order and higher would have to be conducted with somewhat longer seals and greater pump power than that currently available. It was also felt that this would be an even stronger requirement for tests with drier coal, such as that used for tests prior to these most recent tests. As a result, testing was stopped at this point.

CONCLUSIONS AND RECOMMENDATIONS

The major results of this project can be summarized as follows:

1) Successful operation of a modified 4-inch diameter Fuller-Kinyon pump was limited, with a few exceptions, to pressure differentials across the pump of 60 psi or less as a result of pump power requirements which exceeded the 40-horsepower rating of the pump motor in the test setup. Furthermore, operation with straight-pipe seal lengths was in general limited to seal lengths of 10 inches or less for the same reason.
2) If the significant drop in pump power consumption observed just prior to blow-back during some of the tests can be attributed to partial fluidization of the feed material in the pump barrel, then it is likely that a significant portion of the normally observed power consumption also occurred in the barrel.
3) The power required by the modified Fuller-Kinyon pump appeared to increase linearly with increasing pressure differential across the pump.
4) Power requirements were found to be somewhat reduced for a worn version of one of the screws tested.
5) In many cases, unaccountably similar results were obtained for significantly different types of seal configurations, significantly different screws, and significantly different screw rotational speeds. According to Reference 3, explanation of this may be related to rejection of feed material from the pick-up flights of the screw in the pump hopper (due to centrifugal forces) and also to gas infiltration into the feed material in the hopper (due to leakage back through the pump barrel when operating against a gas pressure differential).
6) Increasing the pressure differential across the modified Fuller-Kinyon pump led to a significant and apparently linear decrease in the feed rate for the pump and, for the limited seal lengths which could be tested, an increasing likelihood of blow-back through the system. Explanation of this may also be related to the gas infiltration problem cited previously.

7) Positive modification of many of the above results appears to be possible with longer seals, particularly longer "combination" seals (straight-pipe section combined with a divergent truncated cone).

Most of these results indicate that insufficient power was provided by the motor used to drive the modified Fuller-Kinyon pump (in particular this limited the seal lengths which could be used and therefore the pressure differentials which could be handled). Interestingly, a substantial increase in the power consumed by the system would not violate the guidelines established at the beginning of the project, namely that overall feed system power consumption should be less than 1% of the power eventually generated from the feed material. If we assume for the moment that feed rates of the order of 300 lb/min of -1/4" coal can be realized with the modified 4-inch diameter Fuller-Kinyon pump, then according to the guideline, the overall feed system based on this pump should operate on something less than 240 horsepower (based on power generation of 1 kwhr per pound of coal). Thus, it would be entirely reasonable to increase the power available to operate the modified 4-inch Fuller-Kinyon pump.

It therefore is recommended that further testing be performed with the substantial test setup which is presently available, in conjunction with replacing the 40-horsepower pump motor presently in the test setup with a motor having a significantly higher power rating. (This should in turn allow pump operation with longer seals and at higher pressure differentials than those tested so far.) It is also recommended that if this further testing is successful, additional testing be performed to determine the potential reduction in power requirements which might occur as a result of alterations in screw design and the design of the pump barrel.

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