USE OF CENTRIFUGAL COMPRESSORS

for

TRANSMISSION OF NATURAL GAS

By

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INTRODUCTION

It is contemplated in the preparation of this Thesis, entitled "Use of Centrifugal Compressors for Transmission of Natural Gas", that the historical background in the development of this type of equipment be stated in Part I, giving a description of the first such installation in 1931, and also the operating expense and economic aspects thereof.

During the period 1931 to 1945 considerable research was carried on by the blower manufacturers to perfect a seal for the shaft to withstand 700 or 800 psi, the accepted discharge pressure for large diameter pipe lines. This was accomplished and offered to the industry in 1946. The prime mover for the centrifugal compressors under consideration was the electric motor, but later in 1949 the gas turbine and steam turbine attracted considerable attention.

In order that the proper economic evaluation of the various types of prime movers to pump natural gas be made, a portion of this Thesis, Part II, is devoted to a determination of such pumping costs when using gas engines, gas turbines, steam turbines and electric motors.

Part III will be devoted to a description of the equipment, and economic results obtained by the installation of centrifugal compressors by the "Big Inch" pipeline system, now known as Texas Eastern Gas Transmission Company.

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PART I

History of the Development of the Centrifugal Compressor for Natural Gas Transmission

The use of natural gas for domestic and industrial purposes has replaced practically all other forms of fuel in the Southwest, with the exception of some rural areas, and is now becoming a major source of fuel supply in the East, where this form of fuel has been almost wholly depleted, leaving little dependable capacity available.

Many pipe lines are being projected and older lines are being "looped" in order to increase gas capacities to the East, primary source of supply of which lies in the states of the Southwest including Texas, Oklahoma, Kansas and Louisiana. These states contain within their boundaries almost three-fourths of all the gas reserves in the United States, but consume only about one-half of their own production, leaving almost 2 trillion cu ft per year available for export. With the delivered cost of coal and oil reaching all time highs since the close of World War II, the large industrials and gas suppliers are looking, with envious eyes, to the Southwest to obtain natural gas to replace former solid fuels, and or to augment their present sources of gaseous fuel supply.

For many years the gas transmission industry has followed a conventional pattern of installing reciprocating compressors, driven by gas engines in the booster stations, operating a compression ratios of 1.6 to 2.0, with stations being spaced from 80 to 120 miles apart. Apparently little thought was given to using other forms of power, closer station spacing, rotating machinery, or higher initial working pressures, although the oil industry recognized this problem more than 20 years ago.

It cannot be said, however, that the natural gas transmission industry has been dormant all these years, for as early as 1931 an experimental centrifugal unit was installed in Kansas on a main pipe line serving Kansas City, which originated in the Hugoton field of southwest Kansas. This unit consisted of a 3000-hp, 3600-rpm motor driving six stages of centrifugal blowers, all self-contained in a cylindrical housing with dimensions of about 25 ft in length and 5 ft in diameter, being designed for outdoor installation, mounted on a simple concrete pad. The suction gas was taken in at one end of the cylinder and passed around the stator coils of the motor supposedly for a cooling effect, and then passed successively through the six stages of the centrifugal blowers. The discharge was taken from the side of this cylindrical housing. Observations made in October, 1931, at the installation at El Dorado (See Fig. 1) indicated mechanical efficiency was approximately 75 per cent when delivering 88 million cu ft a day, when the suction pressure was 211 psia, 68°F, and discharge pressure was 292 psia, 130°F, and resultant R 1.38. The K W input was 1400, and with motor efficiency of 95% the bhp was 1770 and using Weymouth formula for the theoritical adiabatic hp at 1.38 compression ratio, it was determined to be 1320 hp.

Power factor of the motor at (1770/300 hp) 59% of full load was 77%. Ordinarily a 2 pole motor operating at full load would have a power factor of about 93%.

This compressor station was located several miles East of Ed Dorado, Kansas, 37 miles from the originating station, the line containing 9.8 miles of 18 inch line and 27.2 miles of 20 inch line or an equivalent 20 inch line 44.7 miles long. As originally designed the line when pumping through 58.6 miles of equivalent 20 inch pipe, would have handled 76,500,000 cfd with 292 psia pressure at the originating station. The gain, therefore, in capacity was (88/76.5) 15%. The alternative would have been to lay a "loop" of 34% of the distance (58.6 miles) or 20 miles of 20" line to cost, in 1931, $500,000. Ordinarily, a booster station of this type would be required only in the winter months (December, January, February and March), so that the principal expense of about 75% of the total annual charges could be attributed to the cost of power, which in this case would have been (1400 KW x 720 x 4 months x 1.0¢) $40,400 or a total of $53,800 per year to cover power, labor and fixed charges. Note that the fixed charges on the cost of 20 miles of "loops" would be about $50,000, or an offset practically. Generally speaking, a booster station of this type can be installed economically but a general program of "looping" is resorted to as the market for gas increases, so that the operating life of the booster station is only 2 or 3 years, since it is obvious that if capacity increases become necessary for longer periods than 4 months per year, then the "loops" or parallel pipes become more economical.

The cost of this experimental unit complete with transformers, switch gear, and piping, was about $120,000 or $40 per hp. The discharged gas was not after-cooled before passing into the pipe line, since the temperature of the discharged gas was only 130°F.

For some unaccountable reason, further field experimental work was not followed up with this unit, but later developments indicated "loops" were being installed to gain increased capacity, since some difficulty was encountered in the original design. Wet gas (unprocessed) passing around the motor coils caused deterioration and ultimate breakdown of some of the windings. The basic centrifugal principle, however, remained alive and most of the future research work was centered on developing a shaft seal to eliminate the compact cylindrical housing that contained both the motor and the centrifugal blowers. See Exhibit "A".

In the meantime, rotating machinery for pumping natural gas was being used in West Virginia for gas gathering lines. These units were of the rotary type with radial sliding vanes accomplishing compression. Working pressures of only three or four atmospheres were utilized. Electric motors were used as prime movers to be actuated with mercoid automatic pressure control. Motors as large as 100 hp, to handle some 2 or 3 million cu ft a day, were employed.

Practically all centrifugal blower manufacturers, except one, refrained from offering equipment to operate against the higher working pressures of 500 to 800 psi, ordinarily employed in main gas transmission lines. This one manufacturer, however, continued the development of the shaft seal and after considerable research was in a position to offer such equipment at the time the "Big Inch" pipe lines (24 in. and 20 in.) were offered for sale by the War Assets Administration in early 1946.

These pipe lines constructed during the war (World War II) were desired to deliver oil and/or oil products from Texas to the Eastern Seaboard, but were destined to be used for gas deliveries, at the close of hostilities in World War II. The oil industry had amassed a sufficient number of tankers during the latter phases of the war, so that the Big Inch lines, as oil carriers, would no longer be required especially in peace time. The physical manner in which these lines were constructed, the station spacing, and the availability of sufficient electric power every 50 miles, made the use of electric-driven centrifugal compressors a distinct possibility.

As stated previously, the gas transmission industry had always accepted line design compression ratios of about 1.8, along with station spacing of about 100 miles, as being the most economical arrangement for its purposes.

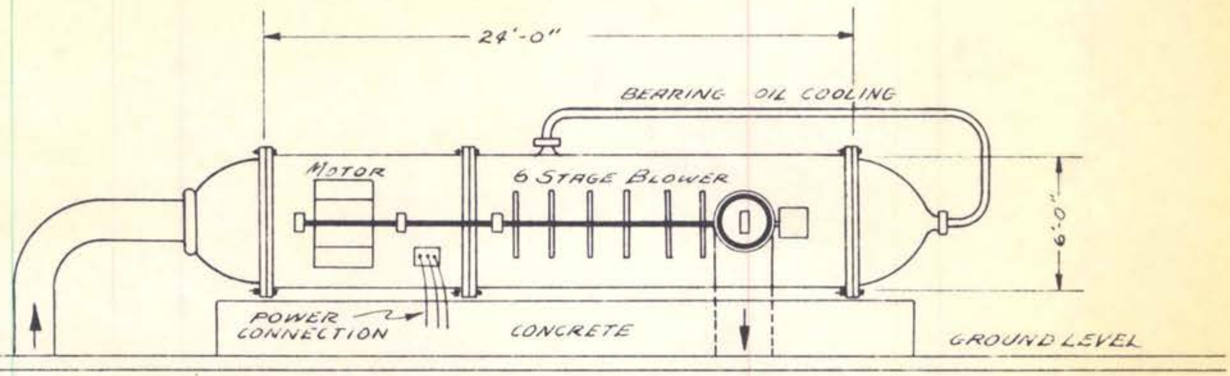
Subsequent studies of the economics of gas pumping indicated that closer station spacing with consequent lower compression ratios might prove more beneficial, in that great savings could be made in invested capital, since the required horsepower installed could be reduced in the order of 25 to 40 per cent.

Referring to Figure 2 it can be noted how this saving in total hp per mile can be reduced. Thus at 1.8 compression ratio (conventional) the theoretical adiabatic hp is 27.5 per million C F D, whereas at 1.25 compression ratio the theoretical adiabatic hp is only 10 hp per million C F D, required at 1/2 the original distance, or a saving of some 27% in total hp. This saving is important economically because each hp saved in equipment represents a capital investment of about $250. A 1000 mile pipe line, originally required 10 stations of 8000 hp each. Savings would approach $3,750,000, provided 20 stations were installed and these to operate at a much lower compression ratio such as 1.25 or 1.30.

Exhibit "A" is a photograph taken in 1931 of the original centrifugal station installed by Cities Service Gas Company at El Dorado, Kansas, in their Kansas City supply pipeline.

FIGURE 1

SCHEMATIC OF FIRST CENTRIFUGAL PUMPING UNIT
INSTALLED BY CITIES SERVICE GAS CO.
EL DORADO, KANSAS



OPERATING DATA

OBSERVED OCT. 8, 1931

INTAKE 196 #GA. 68°F
DISCHARGE 277 #GA. 130°F
COMP'R. RATIO 1.38
THEO. ADIAB. H.P. 1320
K.W. INPUT 1400

BRAKE H.P. 1770
MOTOR EFF 95% 3580 R.P.M.
CAPACITY 88,000,000 C.F.D.
COMP'R. MECH. EFF 75%
POWER FACTOR 77%

LINE SYSTEM

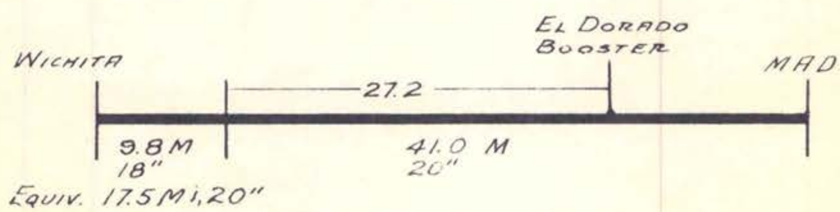
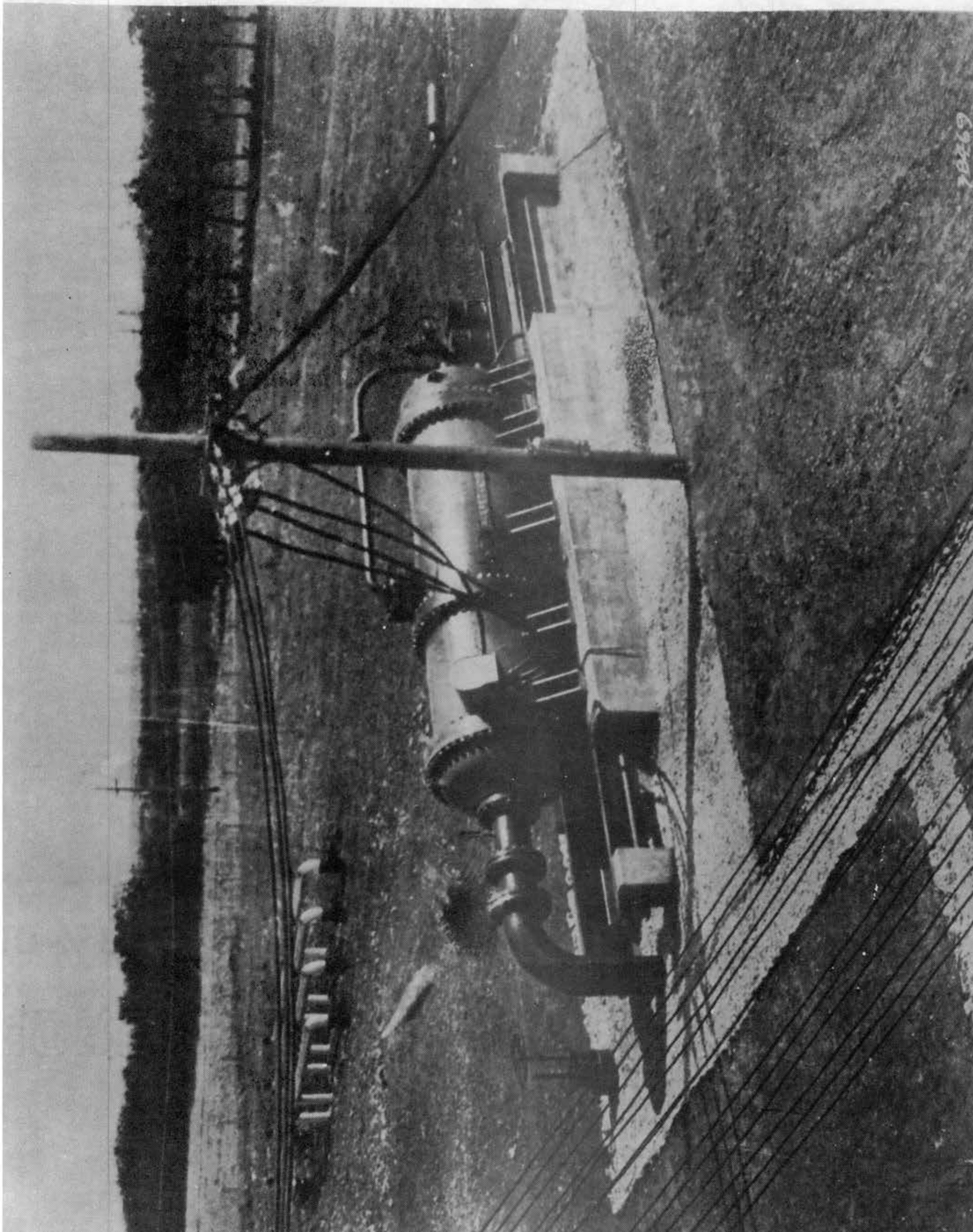


EXHIBIT "A"



3000 HP SIX STAGE CENTRIFUGAL COMPRESSOR

CITIES SERVICE GAS
EL DORADO, KAN.

PART II

1. Economic selection of pipe diameters and wall thicknesses for given quantitites of gas to be delivered in long pipe lines.
2. Economic evaluation of natural gas transmission costs using conventional gas engine driven reciprocating compressors, electric motor driven centrifugal compressors, gas turbine driven and steam turbine driven centrifugal compressors.

The results obtained from the first centrifugal compressor installation at El Dorado, Kansas, in 1931 were generally made known to the gas transmission industry, but the problem of sealing or packing off the shaft of the blower was considered an impossible solution for a great number of years. When consideration was given the higher line discharge pressures sought after, generally, in the order of 700 to 800 psig., only one manufacturer of blower equipment (Ingersol-Rand) continued research on the problem and finally, in 1946, evolved a solution by using the oil pressure system on the bearings. This method is now in general use. Since this firm had supplied the six-stage blowers for the installation at El Dorado, described in Part I, it naturally led to their continued research of satisfactory seal. Exhibit "B" illustrates the oil pump with separate motor drive to supply the shaft seal oil for the bearings on a compressor for the "Big Inch" stations. The oil supply to bearings is maintained slightly higher than the casing pressure of the gas discharge. The loss of the seal oil is very slight to the gas stream, amounting to only about 1.5 gallons per day per station of three units on the average.

The 3000 HP motor furnished in the El Dorado booster station was supplied by General Electric Company, and they too, continued to make economic studies having to do with the possibility of using the electric motor drive to operate the centrifugal compressors. Studies made by GE engineers in 1945, however, indicated that electric energy would have to be purchased for about 1/10¢ per KW hr to compete economically with the gas engine driven conventional reciprocating compressor. The matter of using high speed motors (3600 rpm) for centrifugal compressors, was in the same phase of skepticism at that time, as was the use of centrifugal pumps for oil pipe line service some 20 years previously. With the shaft seal problem being solved, there remained the problem of securing some prime mover of high rotative speed which could be installed at lower capital investment cost than the gas engine. The first selection made was that of the steam turbine but sufficient condensing water make-up or water of the proper quality for boiler make-up could not be obtained in most instances due to the remote location of the booster stations. About 1946 the "Big Inch" system was offered for sale by the Defense Plant Corporation, The specific location of the pipeline together with the availability of electric power in sufficient quantities every 50 miles along its route made it a feasible layout for the adaptation of the motor driven centrifugal compressor, especially since it was found more profitable to use lower compression ratios, compression ratios lower than the conventional accepted ratio of 1.8 to 2.0.

The terms of the sale imposed by the U. S. Government (DPC) stated that "in the event of an emergency, the pipe line had to be restored to the Government again within 60 days, capable of pumping oil"; this condition made imperative, that the motors, power lines and all electric switch-gear equipment be left intact and in first class operating condition. To meet these terms of sale the centrifugal pumps were merely set aside and replaced with similar pieces of equipment, i.e., the centrifugal compressor, which was designed to have the same shaft height, foundation bolts and flange couplings as the pumps.

The operating requirements of the centrifugal compressors was the same as for the pumps. Three compressors were couples in series with the total volume of gas passing through each single stage compressor. Sufficient flexibility could thus be obtained with the constant speed motors (3580 rpm) to satisfy the demands. This flexibility was accomplished by means of flow stationary vanes to give volume changes as demands made necessary, and also to improve mechanical efficiency.

The completed "Big Inch" system now owned by Texas Eastern Gas Transmission Company, as of March 1951, has installed 88,900 hp of gas engine driven reciprocating compressors, and 188,500 hp of electric motor driven centrifugal compressors. See Table 5, Part III, Pipe Line News, Bayonne, N. J., March 1951, E. A. Koenig, Clark Bros. Eng. Co.)

Thus the use of centrifugal compressors for gas transmission has been fully established, insofar as the actual performance of the equipment is concerned, however, in this case the circumstances practically demanded that motor driven centrifugal compressors be used in 16 stations out of a total of 26 for reasons primarily involving lower investment in new equipment, and secondarily meeting the emergency requirements of the U. S. Government.

There were many other reasons for using water driven centrifugal compressors, one of which is providing more gas for sale in the East, than would be available in case gas consuming equipment, for power supply, had been installed. Purchasing of electric power which has been generated by burning of coal by the utilities in their respective service territories in which the stations were located, consumed no gas from the pipeline.

These considerations apparently satisfied the Federal Power Commission charged with the duty of obtaining the lowest rates possible for the public under the Natural Gas Act of 1938.

Since several gas booster stations are being installed by the Transcontinental Gas Transmission Company in Texas, using steam turbines as prime movers. The El Paso Gas Pipe Line is using gas turbines for several stations on their pipe line running from southwest Texas to California, all stations of which are equipped with centrifugal compressors. An economic study covering the use of the four types of prime movers is offered in the following portion of this thesis, Section II.

1. Deliverability of gas "Q" for standard pipe diameters.

Before offering the results of an economic study of the selection of prime movers it was considered desirable to discuss the ceonomic factors governing the selection of pipe diameters, wall thicknesses and station spacing. These factors generally tend to cover about 2/3 of the total cost of pumping gas when expressed in terms of cost in cents per standard MCF per 100 miles.

Figure 2 indicates values of HP per million CFD for various compression ratios: Curve (a) indicates the HP for isentropic 100% efficiency, 60°F., 14.4 psia conditions. Curve (b) indicates the HP required for reciprocating compressors with 82.4% mechanical efficiency, an average super-compressibility factor of .905, and a discharge pressure 750-850 psia. Curve (c) indicates the HP for centrifugal compressors operating at 3600 RPM, 82% mechanical efficiency - the other conditions the same as for (b). For the sake of simplicity curves "b" and "c" are combined.

Figure 3 shown curves developed to give capacity of 20" OD for pipe sizes 22" OD, 24" OD; 26" OD; 28" OD and 30" OD. These diameters were selected because they fall in the category of those diameters generally selected for long interstate pipe lines, when volumes of 250 to 500 million CF per day are considered as feasible and bankable propositions. Note that Figure 3 is prepared with "Q" as the abscissa and "F" as the ordinate based on the given conditions, and is accepted for general usage by the industry. Reference - "Gas Transportation System Calculations" by Benj. Miller, 1949. Base 14.4 psia, Sp. Gr. 0.6, viscosity 2.0×10^{-7} seconds per sq ft, Temp 500°F absolute (40°F), pipe line efficiency 100%, Turbulent Flow.

$$\text{Factor "F"} = \frac{(\text{Discharge Pressure})^2\ (\text{psia}) - (\text{Suction Pressure})^2\ (\text{psia})}{\text{Distance in Miles}}$$

Correction factors can be applied readily to values for "Q" as determined from the curves in Figure 3. Thus "Q" increases approximately as 2.65 power of diameter, i.e., a 1% increase in diameter will cause a 2.65% increase in "Q". Note that the 24" OD pipe curve is labeled 23.5", which is the actual inside diameter giving 8/32" wall thickness. The other curves are for inside diameters also.

The capacity of any size pipe is also affected by the flowing temperature of the gas, and "Q" decreases by .1% for each degree F. above 40°F. In this study the temperature is 60°F., thereby reducing the value of "Q" by 2%.

Capacity or deliverability is further decreased directly as the "pipe line efficiency" falls below 100%. The theoretical condition for a new smooth pipe being taken as 100%. In actual practice the "pipe line efficiency" is generally assumed to be 92% or sometimes slightly greater, but for simplicity in this study it is taken at 90% which is also the value expected after some years of operation.

Thus with 24" OD pipe, and 5/16" wall thickness, the internal diameter of the pipe would be decreased from 23.5" to 23.375 or a reduction of .6% and correction factor in this case would become 98.4% for determination of "Q".

Therefore, to deliver 300 million CFD through a 24" OD pipe with 5/16" wall thickness, the value of "Q" from the curve should be increased by dividing 300 M^2 CFD by the product of .90 (efficiency) x .98 (effect of a temperature of 60°F.) x .984 (.6% of 2.65 or 1.6%) diameter correction of .867 giving "Q" a value of 347 M^2 CFD. Using 35,000 psi stress for the pipe and the Barlow formula where working pressure P = 2 x psi stress x wall thickness - outside diameter of pipe 24", P becomes 914 psig for 10/32" pipe. Similarily for 9/32" wall thickness pipe, P becomes 815 psig for 24" OD pipe.

In case the base absolute pressure, is greater than 14.4 psia, then the flow will be decreased in the ratio of 14.4 to higher base delivery pressure. As an example, if the gas is to be delivered at 14.65 psia, then the deliverability will be 14.4/14.65 or 98.2%.

In case the specific gravity is increased from .60 to .65 an approximate correction factor of 3.5% decrease in "Q" would be noted, and conversely if the specific gravity is decreased from .65 to .6 an increase of approximately 3.5% in "Q" would apply.

Let us assume that certain bankers, through their respective engineers, have worked out contracts with gas companies and industrial plants, which call for the delivery of 300,000,000 CF per day of natural gas somewhere in the East, at a base pressure of 14.4 psia and that delivery of this gas requires several hundred miles of pipe line.

Calculations indicate that if the initial working pressure is maintained at about 800 psig, stations are spaced at 100 mile intervals and compression ratios of about 1.5 to 1.6 (conventional) are broadly considered, that 22", 24", or 26" pipe would fit the condition, since factor F from Figure 3 would be approximately 4,000.

Table 1 is now constructed primarily for different wall thickness of 24" OD pipe. Initial working pressures (maximum) are indicated, when maximum allowable stress of the steel pipe is 35,000 psi. The calculations are made for compression ratios of 1.2 to 1.5 to determine the pumping cost in "cents per 1000 cubic feet per 100 miles."

Figure 4 is now constructed from the data obtained from Table 1, with "Miles between stations" as abscissa and "Compression Ratio" as ordinate and for pipe wall thickness 8/32", 9/32", 10/32" and 12/32", for the initial condition of delivering 300,000,000 CF per day through 24" OD pipe line. Also shown in Figure 4 are the points and curves for values both of 22" OD and of 26" OD pipe. Figure 5 is drawn to indicate the "HP per Mile" required for the same pipe diameters at various "Compression Ratios."

TABLE I

(a) Pumping 300,000,000 CF per day at CR 1.2

OD Pipe Diam	Wall thick-ness	Max. Inj. psia	Mile bet. sta.	HP per mile	Thousands $ Investment/mile Pipe	Sta.	Tot.	Operat-ing $ Cost/mi.	¢/MCF/ 100 Miles
24	8/32	750	37.4	76.0	37.8	17.5	55.3	14,030	1.280
24	9/32	830	44.6	64.0	40.3	14.7	55.0	13,570	1.238
24	10/32	929	55.0	51.7	44.0	11.8	55.8	13,570	1.238
24	12/32	1115	77.5	36.8	50.3	8.7	59.0	13,330	1.215
22	10/32	1010	42.2	67.5	40.7	15.5	56.2	14,040	1,284
26	10/32	855	64.2	44.5	47.3	10.2	57.5	13,740	1.258

(b) Pumping 300,000,000 CF per day at CR 1.5

24	8.32	750	68.0	95.0	37.8	22.9	60.7	15,440	1.410
24	9/32	830	81.2	79.5	40.3	18.3	58.6	14,820	1.355
24	10/32	929	100.0	64.5	44.0	14.8	58.8	14,520	1.328
24	12/32	1115	141.5	45.4	50.3	10.4	60.7	14,480	1.310
22	10/32	1010	77.0	84.0	40.7	19.3	60.0	15,280	1.395
26	10/32	855	121.0	53.2	47.3	12.2	59.5	14,400	1.315

Table I is prepared by using the following values;

Pipe line costs: $3750 per mile for right-of-way, clearing, etc.

$ 375 per mile per inch of diameter for ditching, laying pipe, welding and backfill

$ 150 per ton for pipe. (24" - 8/32" 166 tons per mile)

Fixed charges: Pipe line depreciation 2.5%; station depreciation 3.5%; maintenance pipe lines and stations 1%; advalorem taxes 1.2%; administration 5%; profit 12.3%, so that earnings can be 6.5% allowed by Federal Power Commission, to be paid after Federal Income Tax of 47% is paid. Total pipe system 22% and for pump stations 23%.

Station Operating cost: $7.00 per HP per year for fuel gas, 10 CF per b.h.p. at 8¢ per 1000 CF.

$15.00 per HP per year for operating labor.

$230 per HP installed (115% of net HP required) assuming 6 units for net power, plus one stand-by unit.

The results computed from Table I are plotted in Figure 6, with "Compression Ratio" as abscissa, and "Cents per 1000 CF per 100 Miles" as ordinate, in order that the most economic pipe diameter and wall thickness can be selected for the condition of pumping 300,000,000 CF per day.

It can be also noted from Figure 6 that the costs for the thinner walled pipe are greater as the compression ratio becomes greater. Note that the 22" OD pipe costs are greater than the 24" OD costs, and further that the costs for 26" pipe are also greater than the 24" OD pipe costs, except that the 26" pipe costs become less after passing 1.4 CR, but 1.2 CR is cheaper for all diameters and wall thicknesses; therefore, 24" OD pipe with 10/32" wall (5/16") is selected as the best, in view of the fact that while the pumping costs are slightly greater (1.238¢ vs 1.215¢) than the 24" - 12/32" pipe, the investment per mile for the pipe line and stations is $3,110 greater for the 24" - 12/32" than for the 24" - 10/32" pipe. Of course this greater cost is included in the pumping costs of 1.215¢ per MCF per 100 miles, but the capital money required may be hard to obtain when extended to a long line, say 1000 miles , or which means $3,110,000 more for the 12/32" pipe than if 10/32" pipe had been used.

Another reason for selecting the pipe with the 10/32" wall thickness was that the industry generally is reluctant to consider initial pressures in excess of 1000 psi. Note that the 24" - 12/32" pipe requires 1,115 psi when pumping 300,000,000 CF through 77.5 miles with CR of 1.2.

If other capacities are specified, the same procedure can be followed, to determine the most economical diameter, station spacing and wall thickness of the pipe.

2. <u>Selection of most economic prime movers</u>.

The gas engine driven reciprocating compressor has been the accepted type of equipment for many years, and only recently (1946) has the centrifugal compressor been considered by the industry for initial pressures of 600 to 1000 psi at the pumping stations. A study of the various types of power supply involves a great many factors, and many of these factors are of an indeterminate nature, primarily they are those factors effecting the cost of the installed equipment; cost of the fuel, efficiency of the compressor unit; thermal efficiency of the power unit, and labor and maintenance in station operation. The question of standby equipment is also important, and the solution of this problem seems to rest with each individual design engineer.

Generally speaking the most quoted costs are those of the conventional gas engine - reciprocating compressor station. In 1946 a cost of $175 per HP was generally accepted, but some contracts recently indicated a cost of $250 per HP would be more appropriate; however, these prices included a cooling tower for the compressed gas with ratios of compression of the order of 1.6 to 2.0. Since cooling towers are not required when the compression ratio is less than about 1.38 and 130°F., (Reference Figure 1.) it is assumed that the cost today of the engine station will approximate $200 per HP when the station spacing involves compression ratios of 1.2 to 1.3. Generally a standby unit is included in the design when the station contains 3 to 6 engine units for maximum output. Again the matter of mechanical and thermal efficiency is subject to discussion, so again for the purpose of this study an average is struck between the claims of various design engineers, as 37% for the gas engine at full load (6900 BTU per BHP), and 82.4% mechanical and compression efficiency of the reciprocating compressor.

In the case of centrifugal compressors the mechanical and compressor efficiency was taken at 82% for single stage units and for units in series. For the purpose of this study the BHP required for reciprocating and centrifugal compressors is assumed equal. (Reference Figure 2.) The thermal efficiency of the steam turbine and gas turbine with regenerators is taken at 25%. The cost of the gas turbine for unit sizes of about 2000 hp is taken at $190 per HP (Exhibit "C" - Oil and Gas Journal, Page 102, August 9, 1951). Under the assumption that the cost of regenerative equipment for the gas turbine would equal the cost of condensing equipment for the steam station, there would still remain the cost of boilers and cooling tower to increase the cost of the steam station about $35.00 per HP over the gas turbine station. The electric station (without transformers) contains merely the motors, switchgear (across-the-line starting) and centrifugal compressors, and a fair estimate of cost in comparison to the cost of the other types above mentioned would be about $150 per HP.

In 1943 the 4500 HP stations for the "Big Inch" system for oil pumping, cost $75 per HP with practically the same style of equipment installed. Explosion proof motors, however, are contemplated for gas pumping. The centrifugal single stage compressor simulated the centrifugal pump in outward appearance.

The cost of fuel is taken at 25¢ per MCF, since that is to be the assumed wholesale value in the East, or delivery point. However, the Federal Power Commission permits a charge against operations of only 8¢ per MCF, the gas field price compressed to line pressure at the first station or origin. It would appear, that to be realistic in comparing the three fuel burning types of prime movers, i.e.: gas engine, gas turbine and steam turbine with the electric motor which consumes no fuel from the pipe line, that the sale price of the delivered gas should be charged against the fuel burning stations. The cost of natural gas in the East has to be competitive with coal and oil. Electric power generation by utilities would pay perhaps 20¢ per 1000 CF of gas making up 50% of sales of the pipeline delivery would pay 30¢ per 1000 CF of gas equivalent ($7.20 coal), so that the average price of 25¢ per 1000 CF seems equitable for wholesale or gate delivery at East coast cities.

Fixed charges for the pumping stations are taken at 23% since the allowable depreciation factor of 3.5% is determined by Federal Power Commission. For the pipe line itself this depreciation factor is 2.5% for a total of 22%. Other items making up the total fixed charges have been mentioned previously in Part II, 1. (Determination of economic pipe sizes and wall thicknesses).

The matter of determination of station operating costs is predicated on the economic choice of pipe diameter and wall thickness, to pump 300,000,000 CF per day through 24" pipe with 10/32" wall thickness, with station spacing of 55 miles. (Reference Table 1). This station spacing is also selected as being most practical, in the event the stations are to be used as intermediate boosters for an approximate increase in pipeline capacity as contemplated for the future.

Table 2 is offered to show investment in stations, and operating expense for the condition of most economic pipe diameter and wall thickness selected in Part II, Section 1 for delivery of 300,000,000 CF per day, a hypothetical case as a profitable investors proposition.

TABLE 2

300 M^2 CFD, Discharge 929 psia, Ratio 1.2, 55 miles

24" OD 10/32", 23.375 ID, 60°F., .60 sp. gr., K 1.25,

base 14.4 psia, BHP 2850.

	Gas Eng Recip Compr	St Tur Centr Compr	Gas tur with reg. Centr Com	Elec Mtr Centr Compr
Equipment	4-1000 HP	2-1650 HP	2-1850 HP	2-1500 HP
RPM	-	5000	5000	3600
Therm Eff %	37	25	25	x 94
Cost per HP	$ 200	$ 225	$ 190	$ 150
Mech & Compr Eff	82.4	82	82	82
BHP 100% Cap	2850	2850	2850	Input 3040
Installed Cost	$800,000	$745,000	$700,000	$450,000
x Mech Eff				

Annual Operating Costs - 100% Capacity

Fixed chg:23%	$184,000	$171,000	$161,000	$103,500
Labor:				
9 Oper's/Mo.	$ 38,400	$ 38,400		
5 Oper's/Mo.			$ 21,800	$ 21,800
Fuel:				
6.9 CF/BHP	$ 43,400			
10.2 CF/BHP		$ 64,200	$ 64,200	
Elec Power:				
2270 KW @ .8¢/KWH				$159,000 x$139,000
Total	$265,800	$273,600	$247,000	$284,300
			xx($254,000)	x($264,300)

x If power cost is .7¢ per KWH
xx No regenerator (Thermal Eff 17%)

Cost per 1000 CF per 100 miles, cents

	Gas Eng Recip Compr	St Tur Centr Compr	Gas tur with reg. Centr Com	Elec Mtr Centr Compr
Pumping:	.441	.454	.411 xx(.422)	.472 xxx(.442)
Pipe Line: $44,000/mi	.902	.903	.903	.903
Total - cents	1.344	1.357	1.314 xx(1.325)	1.375 xxx(1.345)

xx No regenerator
xxx .7¢ per KWH Power Cost

From the above results, it is obvious that the gas trubine-centrifugal compressor shows the lowest operating costs, provided the capacity factor remains at 100% or the delivery is constant at 300,000,000 CF per day. This high capacity factor generally is not maintained, and Figure 7 is offered to show typical monthly deliveries encountered in actual operations. (Reference July 1947, Petroleum Engineer.)

Note that the 20" line has an average capacity factor of 95.2% with underground storage facilities to absorb the excess gas when apparent 100% pumping is attempted. Practically the same situation exists with the 16" OD line with storage facilities. The thruput of a line serving a large city in Texas is also shown, and note that without storage facilities the capacity factor is only 51%.

In the operation of long gas pipelines, the most economical method of pumping is to maintain high discharge pressures at all the stations starting with the maximum design pressure at the field supply originating station, and for reduced volumes, the suction or inlet pressure is allowed to rise starting with the final or delivery station, and following back eventually to the first station next to the field station.

The effect of maintaining constant discharge pressures and permitting the suction or inlet pressure to rise as deliveries are decreased, is shown in Figure 8. The constant speed electric motor, of course, meets this condition, but the speed can be lowered by the steam turbine and gas turbine for changes in volume. Note that at a 90% capacity factor the total power required is 1990 BHP for constant speed operation (Motors and Engines.), whereas if the speed is lowered to 90% of rated speed for 90% capacity the power required is 2100 BHP for the steam and gas turbine stations.

In one case the power requirement falls off as the 3.5 power of capacity factor (2850 BHP to 1990 BHP) or 31%, whereas if the speed of the centrifugal compressors is lowered the power required falls off only as the 3.0 power of capacity factor. This would seem to point out that speed variation of the steam and gas turbine is not particularly an essential element in the selection of that type of equipment, especially when only slight changes in capacity become necessary.

From Figure 7, (20" Line, 200,000,000 CFD), it can be noted that the average power requirements are 84.5% of the maximum design power requirements when the capacity factor averages 95.2%.

Tables 3 and 4 are now set up, patterned after Table 2, for capacity factors of 90 and 95% to determine overall pumping costs for the four types of prime movers, at capacity factors lower than 100%, with same pipe line spacing (55 miles), and same gas specifications used in Table 2, and maximum capacity of 300,000,000 CF per Day (100%).

The results of calculations in Tables 3 and 4 for the four types of prime movers are plotted in Figure 9. It can be noted from this plot that the "Electric Motor" centrifugal compressor becomes about equal to the "Gas Enginer - Reciprocating Compressor" and "Steam Turbine - Centrifugal Compressor" operation at approximately 95% capacity factor, whereas the "Gas Turbine - Centrifugal Compressor" operation is lower until the capacity factor reaches 85%. On the other hand if electric power can be purchased at an average price of .7¢ per KWH. then the "Electric Motor" operation practically equals the "Gas Turbine" operation at 93% capacity factor. Power for the Texas-Eastern Gas Transmission line, with 188,500 HP connected in 16 stations costs an average of .725¢ per KWH, which would tend to make it practically competitive with all types of prime movers available at the

TABLE 3

Annual Operating Costs - 90% Capacity Factor

	Gas Eng Recip Compr	Stm Tur Centr Compr	Gas Tur with reg. Centr Com	Elec Mtr Centr Compr
Fixed Charges	$184,000	$171,000	$161,000	$103,500
Labor	38,400	38,400	21,800	21,800
Fuel: 1990 BHP 8 cf/BHP	34,800			
11.7 cf/BHP		51,100	51,100	
Elec Power: 1615 KW, .92 eff .85¢/KWH				121,000
Total Pump Cost	$257,200	$260,500	$233,900	$246,300
Cost per 1000 CF per 100 miles, cents				
Pumping: 270 M^2CFD	.473	.476	.431	.453
Pipe Line	.903	.903	.903	.903
Total 270 M^2 CFD	1.376	1.379	1.334	1.356

TABLE 4

Annual Operating Costs - 95% Capacity Factor

	Gas Eng Recip Compr	Stm Tur Centr Compr	Gas Tur with reg. Centr Com	Elec Mtr Cent Compr
Fixed Charges	$184,000	$171,000	$161,000	$103,500
Labor:	38,400	38,400	21,800	21,800
Fuel: 2380 HP				
7.4 CF per BHP	38,900			
11.0 CF per BHP		57,600	57,600	
Electric Power: 1920 KW, .93 eff .825¢/KWH				139,200
Total Pumping Cost	$261,300	$267,000	$240,000	$264,500

Cost per 1000 CF per 100 miles, cents

Pumping: 285 M^2 CFD	.456	.464	.419	.460
Pipe Lines	.903	.903	.903	.903
Total	1.359	1.367	1.322	1.363

time the electric motors were selected in 1947. The "Gas Turbine" was not on the market at that time.

In the calculations shown in Tables 3 and 4, fuel rates have been adjusted to compensate for the lighter loads on the engines, that is, when decreased loading occurs in passing from full load to partial loads. Similarily the cost of electric power is permitted to rise when the demand in KW is lowered for decreased capacities, since power rates are generally designed to take into account the greater unit turbo-generator cost for smaller units on a competitive or "traffic bearing" basis.

The effect of charging the gas consuming prime movers, i.e.: engines, steam and gas turbines with lower gas costs, such as generally used in the gas transmission, cost accounting, of 8¢ per MCF, would tend to change the results demonstrated in Figure 9, and to push these curves to the left, making the Electric" operation a great deal more expensive under all conditions.

It would appear, however, that in order to properly portray the relative advantage of the simple "Electric" operation, the delivered sale value of the gas at the terminus should be used to make comparison, namely, 25¢ per MCF.

The electric utilities, in the main, obtain their power from a coal economy, and, therefore, do not use up the reserves of natural gas, simply to propel the gas to the Eastern customers.

In a long transmission line, requiring a great number of pumping stations, the gas consumed by the gas consuming units becomes worthy of consideration, since the gas consumed per station (Table 2 "Steam" or "Gas Turbine") amounts to "$64,200/25¢) 257,000 MCF per year, or 468,000,000 CF per 100 miles. In other words, if the pipe line were 1000 miles long, the gas consuming equipment would use 4,680,000,000 CF or 4.26% of the line capacities. Certainly some manner of crediting the electric operation should be evolved, in order to make the comparison equitable, and this has been done herein. No attempt has been made to take into account the gradual diminishing amount of horse power required by the pumping stations as gas is consumed at each pump station, leaving a somewhat lesser amount to be propelled to destination. In one sense it may be said that the line actually uses as fuel 1.56 days capacity per 100 miles each year.

The experience of the Texas-Eastern Gas Transmission Company, indicated that a long transmission line cannot be totally electrified, and that reciprocating compressors are required in the line, interspersed about every fourth station location to maintain proper pressures in the event of a line break. The centrifugal compressor becomes unstable at low volumes, and in attempting to pack the line after a break, the extremely high compression ratios encountered would tend to overload the installed centrifugal compressor power equipment, whereas with the reciprocating units all operating in parallel the packing could be accomplished readily, by adding units as it became necessary.

In summary it may be said that in the event the capacity factors or annual load factors of proposed pipelines can be maintained between 95 and 100%, it would suggest that the gas engine reciprocating compressor be continued to be used for station operation, for the regenerative gas turbine has not proved itself, to date, to guarantee power continuity and at low maintenance. Most operators would insist on installing a standby unit, and then the fixed charges would tend to increase the ultimate pumping costs beyond the conventional gas engine costs.

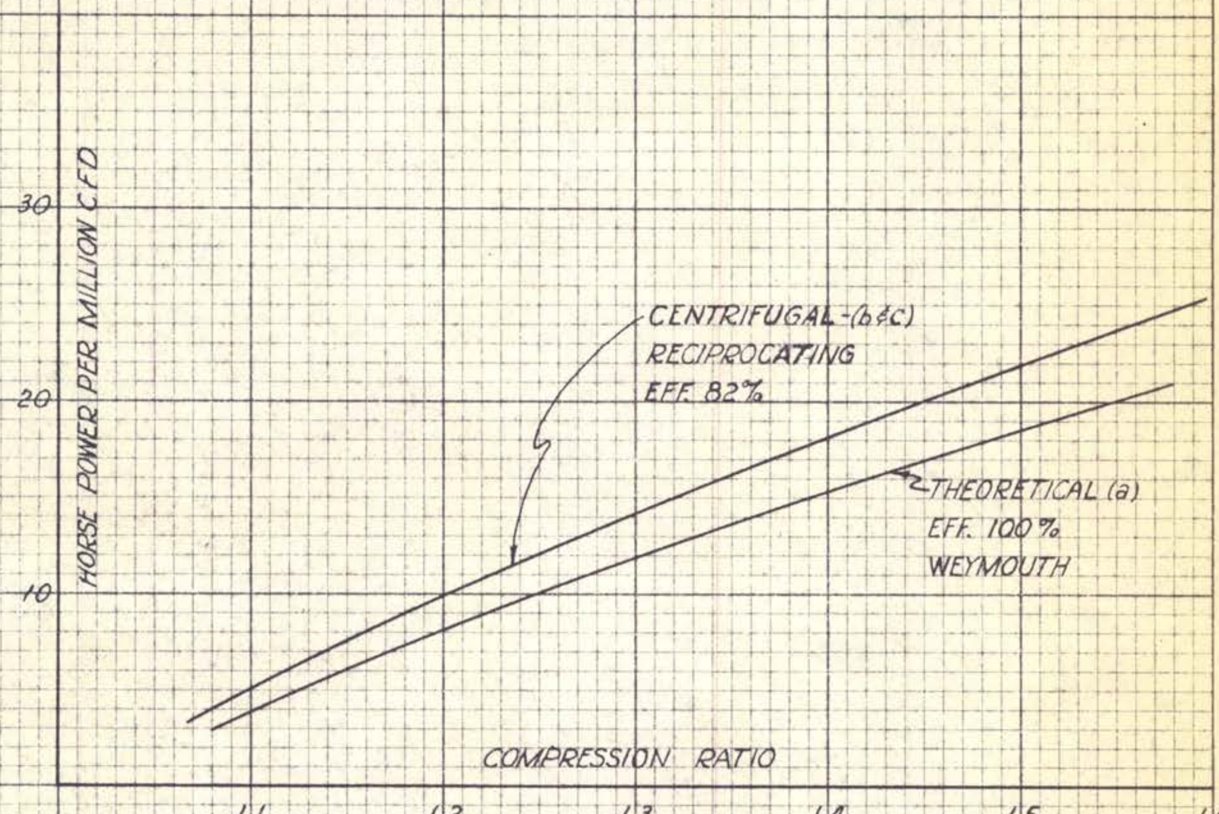
On the other hand, if power can be purchased for about .7¢ per KWH as an average along the line, then "Electric - Centrifugal" operation would be the best economic selection. In general the gas transportation industry would not insist on consideration of standby motors in the electrified stations, due to the experience gained in operation of the Texas-Eastern pipe line. During 1942-1943 with 14,000 hours operation of the Big Inch System, the outages due to electrical difficulties amounted to only 14 hours or .1% of elapsed time.

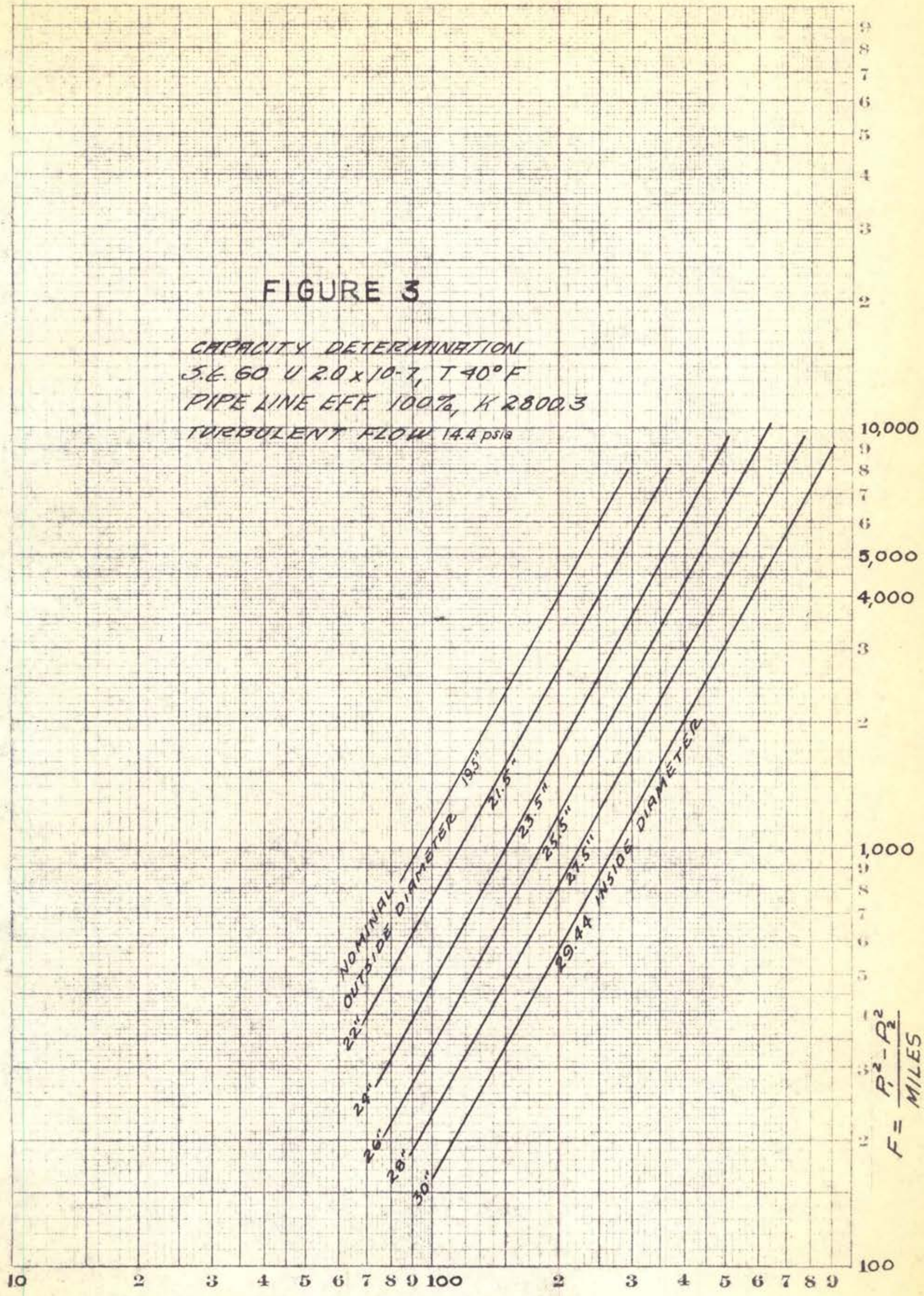
It is also interesting to calculate pumping costs with the simple cycle gas turbine (no regeneration) since it is believed the operators of gas pumping stations would not insist on consideration of standby equipment, due to exceptional good operating experience gained by electric utilities. In this case the installed cost of a two unit station woudl be $600,000 or $162 per HP, but the thermal efficiency would be 17% (Exhibit "C"). Note from Figure 9 that the pumping costs of the simple cycle (one shaft) gas turbine, pass and become greater than electric operation at 96% capacity factor of the pipe line, with power costing .7¢ per KWH.

The slight difference of costs between the two types of gas turbines would indicate that a choice of the simple cycle machine might prove more satisfactory, provided electric power cannot be arranged for, or if its overall costs exceed .7¢ to .75¢ per KWH.

FIGURE 2

H.P.-COMPR. RATIO
750 PSIG, SG.60, 60°F
.9 COMP. FACTOR, 14.4 PSIA





"Q" MILLIONS STD. CU. FT. PER DAY

"GAS TRANSMISSION"
BENJ. MILLER

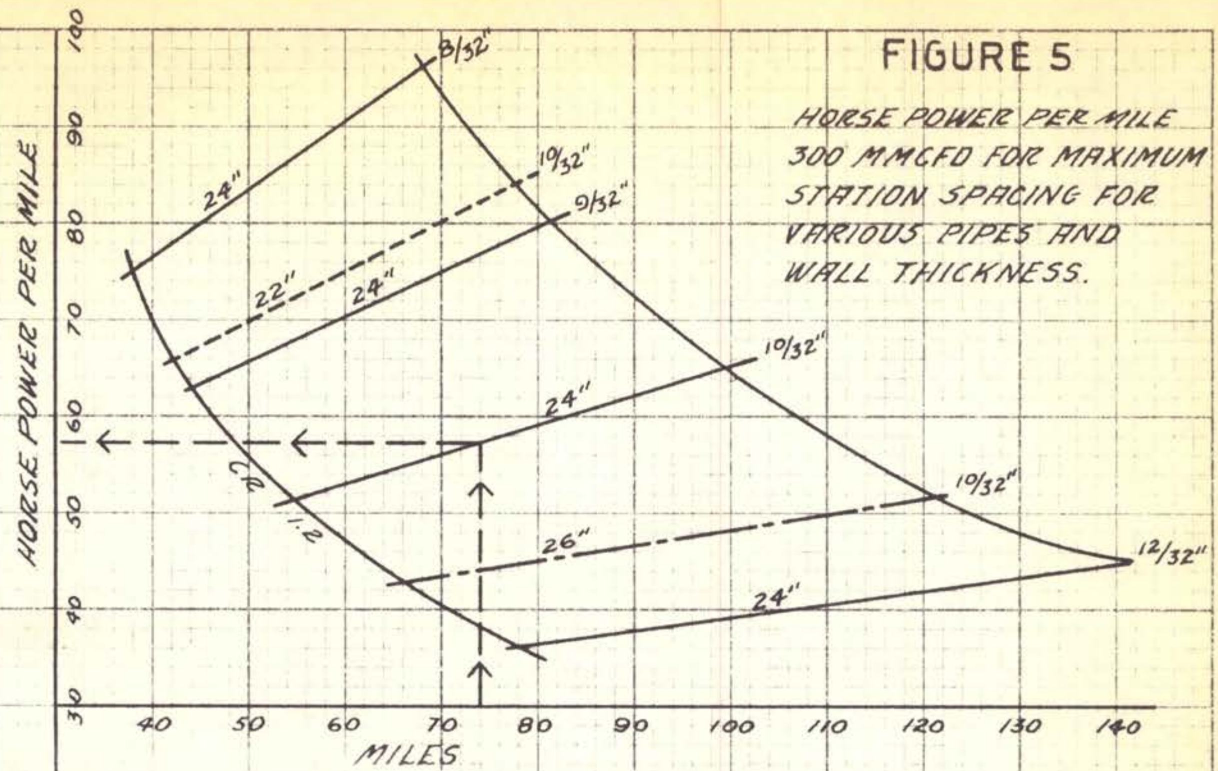


FIGURE 5
HORSE POWER PER MILE
300 MMCFD FOR MAXIMUM
STATION SPACING FOR
VARIOUS PIPES AND
WALL THICKNESS.
HORSE POWER PER MILE
MILES
8/32"
10/32"
9/32"
24"
22"
24"
10/32"
24"
CR
1.2
10/32"
26"
12/32"
24"
30
40
50
60
70
80
90
100
110
120
130
140

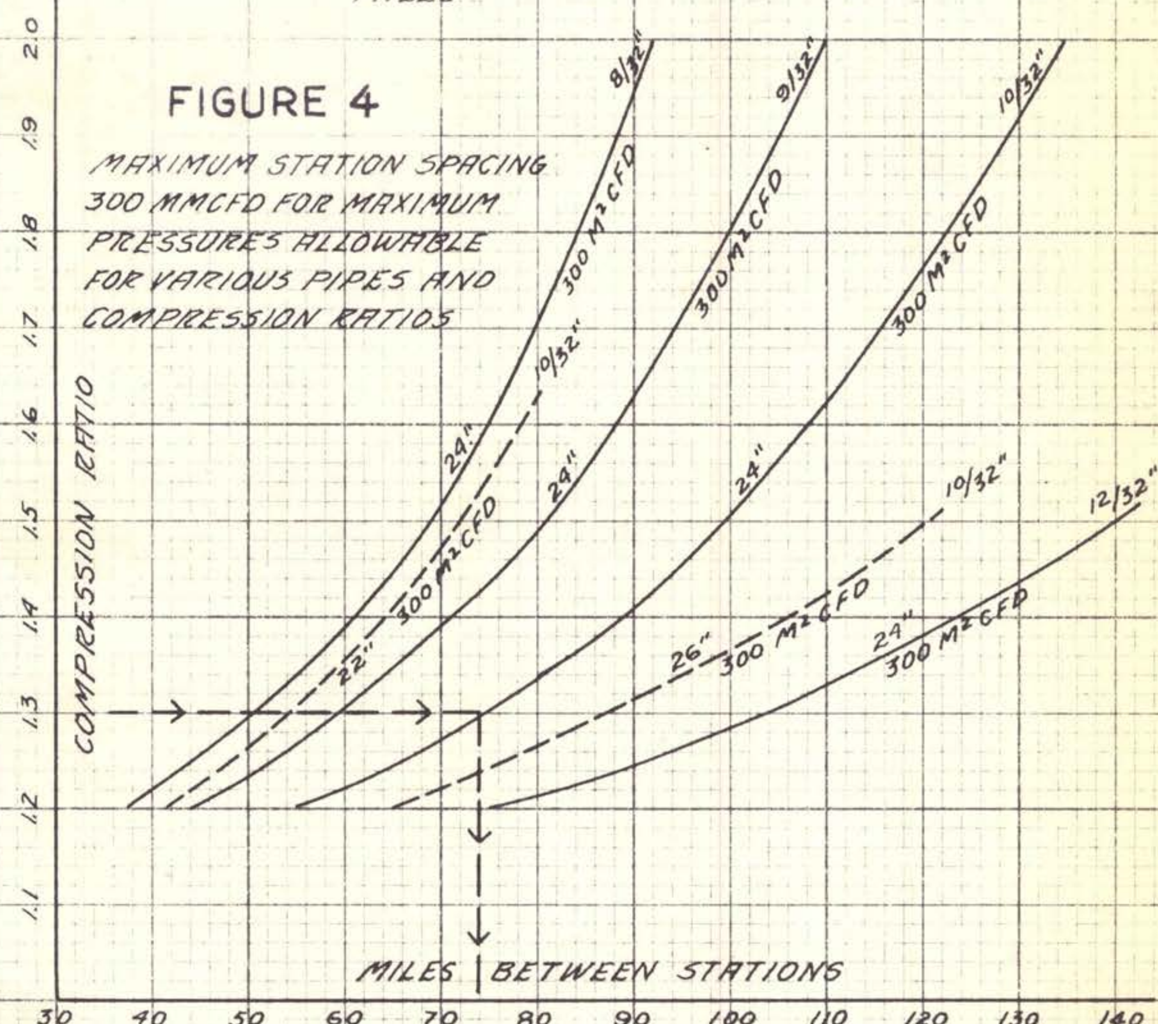


FIGURE 4
MAXIMUM STATION SPACING
300 MMCFD FOR MAXIMUM
PRESSURES ALLOWABLE
FOR VARIOUS PIPES AND
COMPRESSION RATIOS
COMPRESSION RATIO
MILES BETWEEN STATIONS
8/32"
300 M2CFD
24"
9/32"
300 M2CFD
24"
10/32"
300 M2CFD
24"
10/32"
300 M2CFD
22"
10/32"
300 M2CFD
26"
12/32"
300 M2CFD
24"
1.1
1.2
1.3
1.4
1.5
1.6
1.7
1.8
1.9
2.0
30
40
50
60
70
80
90
100
110
120
130
140

FIGURE 6

PUMPING COSTS FOR 24" PIPE
AND RELATION TO 22" & 26" PIPE
REF. TABLE 1

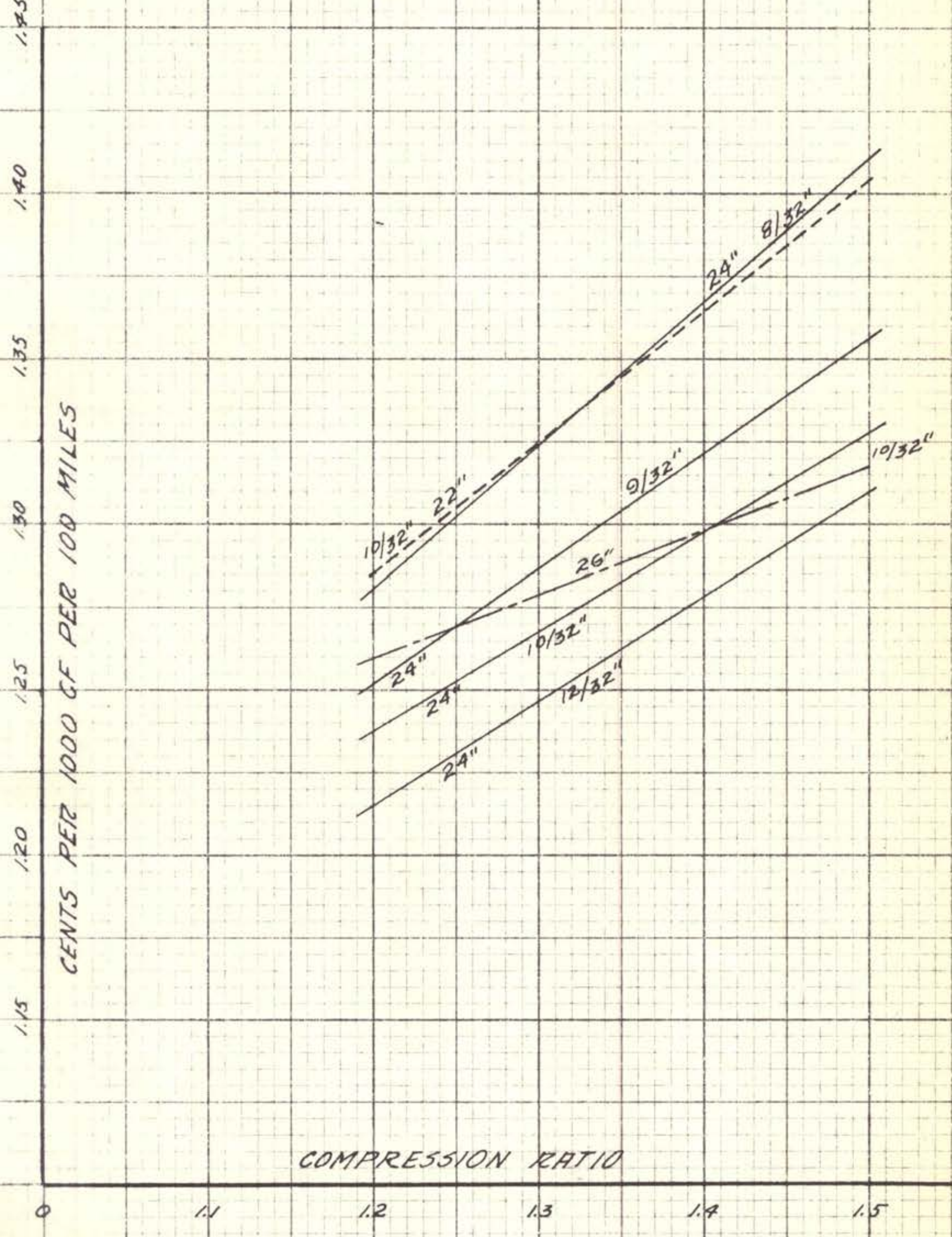


FIGURE 7

TYPICAL CAPACITY CURVES OF GAS TRANSMISSION LINES

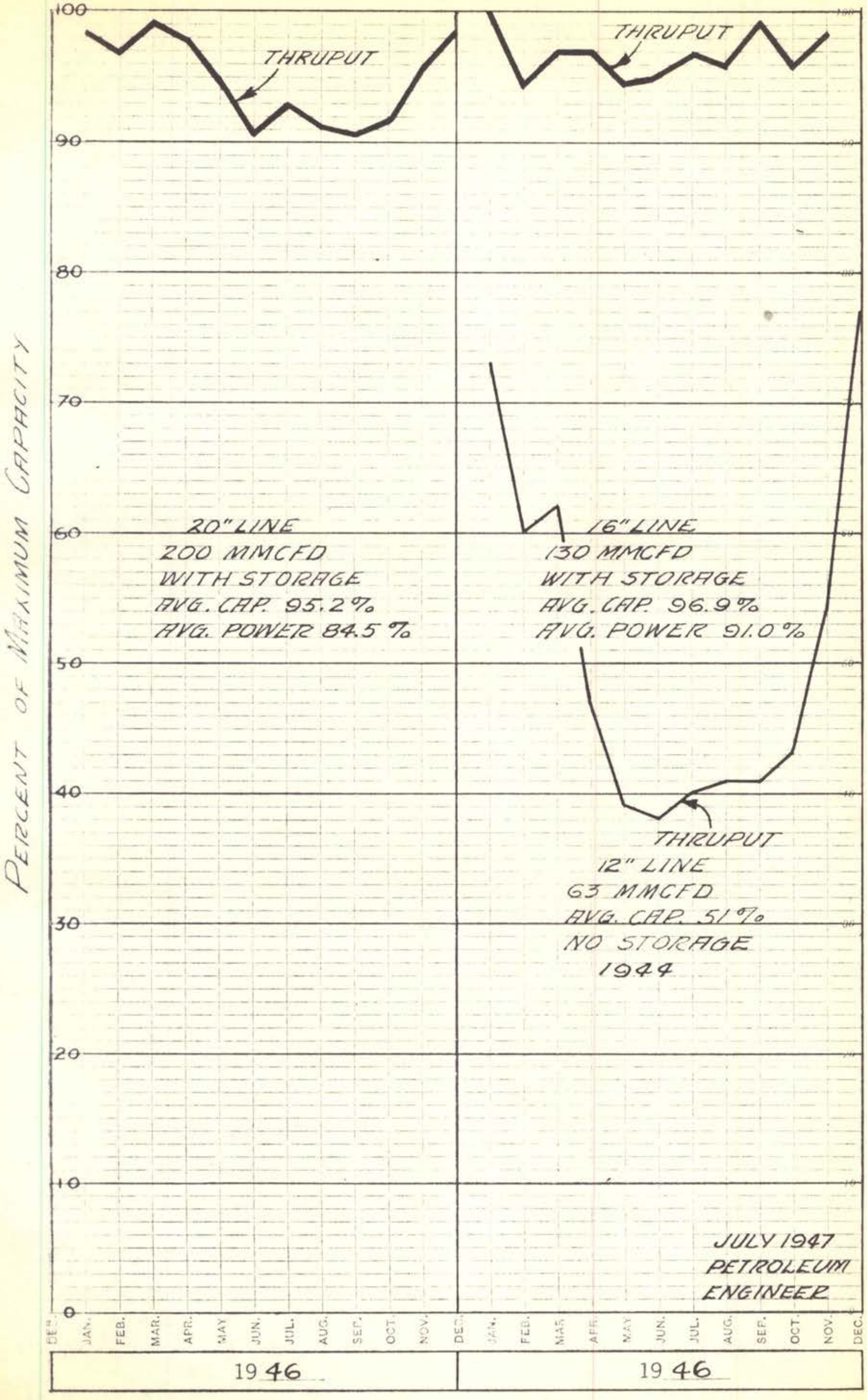


FIGURE 8

HORSE POWER FOR REDUCED
CAPACITIES, MAX. 300 M^2 CFD
929 PSIA DISCHARGE CONSTANT
H.P. = PERCENT CAPACITY $^{3.5}$

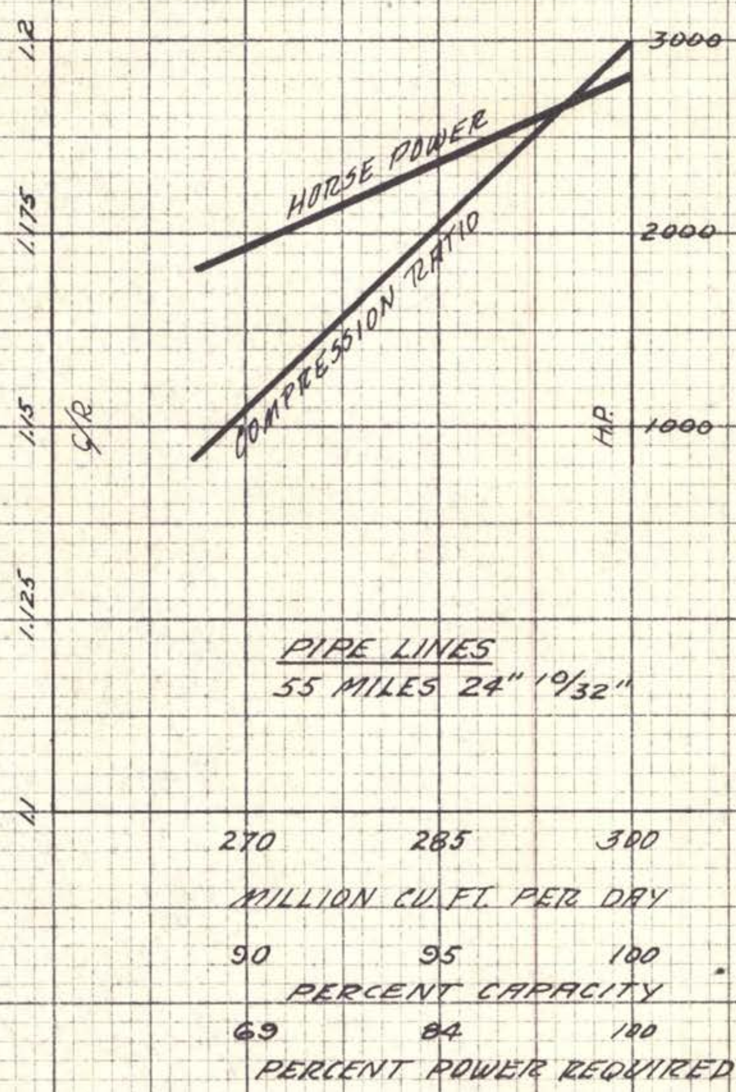


FIGURE 9

PUMPING COSTS WITH FOUR TYPES OF STATIONS
1. GAS ENGINE - RECIPROCATING COMPRESSORS
2. STEAM TURBINE - CENTRIFUGAL COMPRESSORS
3. GAS TURBINE - CENTRIFUGAL COMRESSORS
4. ELECTRIC MOTOR - CENTRIFUGAL COMPRESSORS

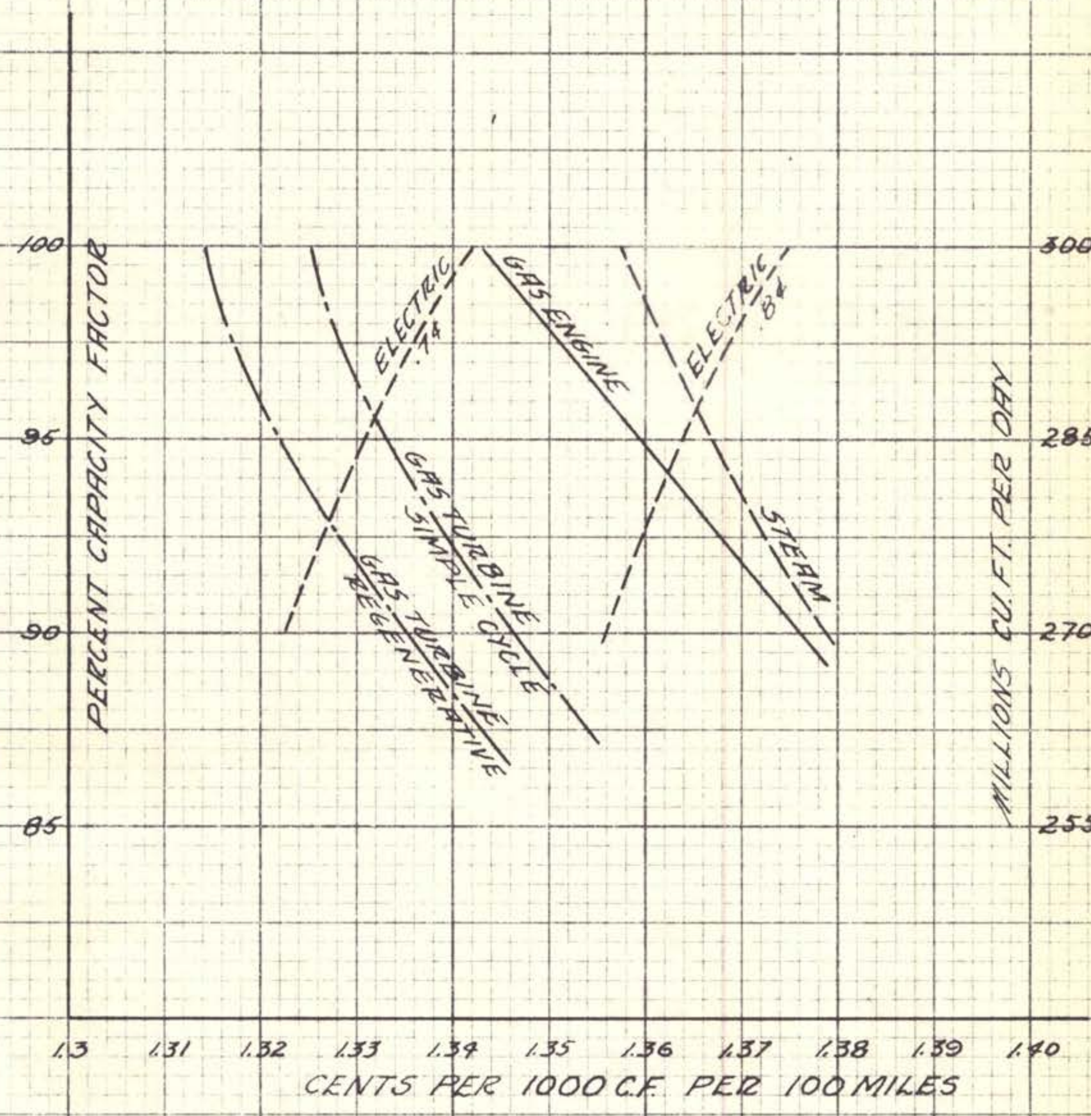
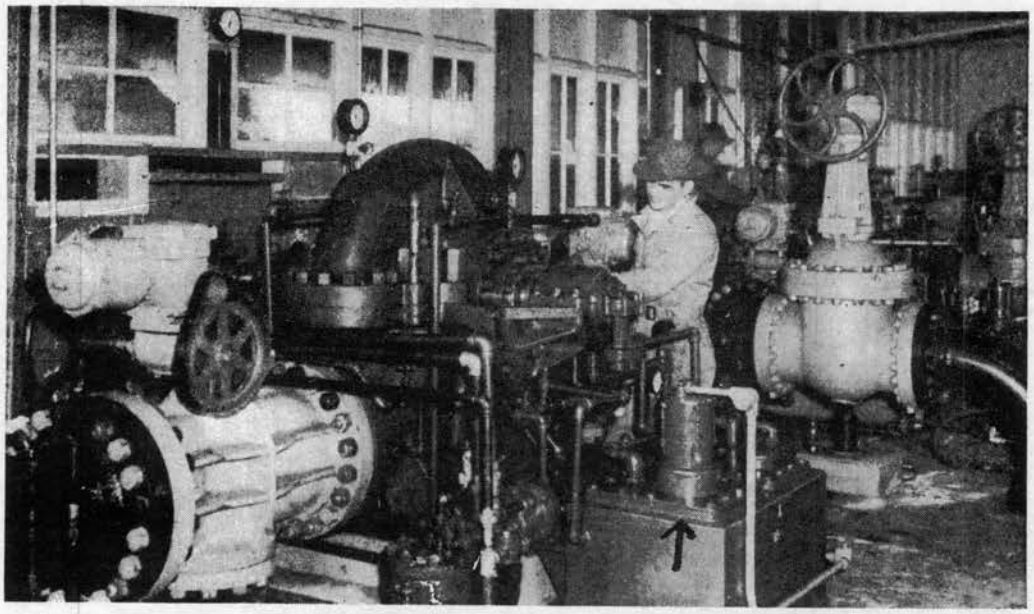
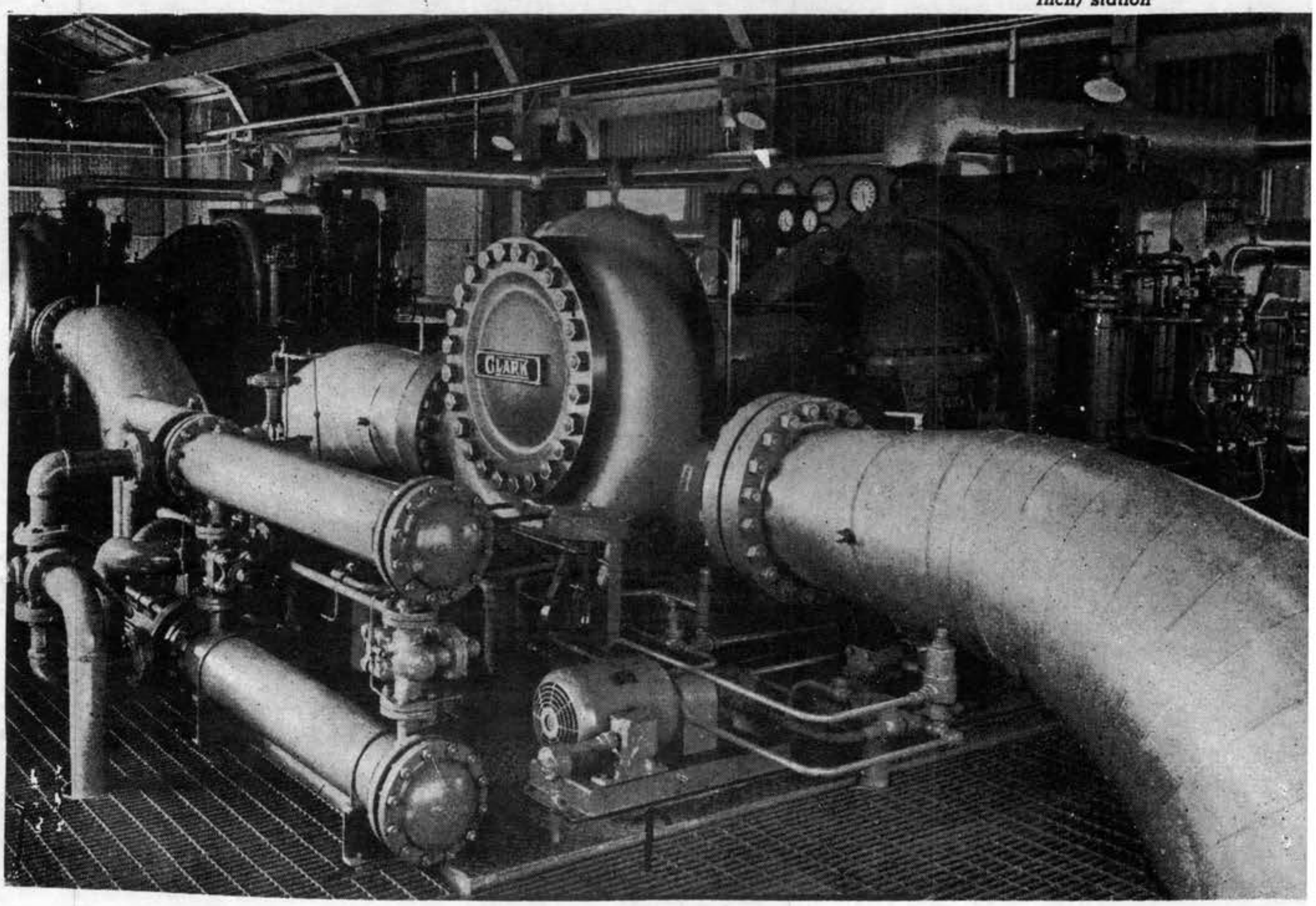


EXHIBIT "B"



Centrifugal compressors at Little Rock installed by Texas Eastern to replace centrifugal pumps of the same flange-to-flange dimensions. Equipment in front of each centrifugal compressor is to facilitate the handling of shaft seal oil, which is circulated by means of pumps, located in the motor room of each station which is separated from the compressor room by a fire wall. No changes have been made in valves and piping, within the stations, originally installed for wartime crude oil and products service. (Left) Twenty-inch line (Little Big Inch) station. (Below) Twenty-four-inch line (Big Inch) station



Combustion Gas Turbine—Important New Prime Mover for the Gas Pipe-Line Industry

Here are design and engineering evaluation data

by J. L. Oberseider*

FROM now on business decisions with respect to additions and replacements of prime movers for gas compressors should be based on economy studies that include the combustion gas turbine as an alternative. Units with rated capacities from 1,850 to 6,500 hp. are, or soon will be, available.

Capital Investment

The fixed capital, that is, the installed cost of a gas-turbine-driven centrifugal compressor, plus the working capital, or cash which must be reserved for the payment of operating expenses, represents the capital investment required. Delivered and installed costs, exclusive of land, or gas-turbine-driven centrifugal compressor units are given in Figs. 1 and 2.

Land requirements are so small that the cost is within the limits of accuracy of preliminary cost estimates. A typical station, consisting of two 5,000-hp. gas-turbine-driven centrifugal compressors, occupies a building approximately 48 by 64 ft. and a total area of about 3,100 sq. ft. Working capital requirements should be about 1.75 per cent of the fixed capital investment.

Annual Costs

The annual costs of operating combustion gas-turbine-driven centrifugal compressors have been segregated into fixed costs and operating and maintenance expenses for the purpose of preparing estimates. These estimates are shown graphically in Figs. 3 and 4. They are based on the literature, private communications, and accounting practice.

A 97 per cent availability, that is 8,500 operating hours out of 8,760 per year, has been assumed on the basis of the two combustion gas-turbine installations in this country that have operated from 7 to 13 months.

Fixed costs.—This portion of the actual costs break down into:

1. Depreciation—straight-line method using zero salvage value and manu-

(Continued on page 104)

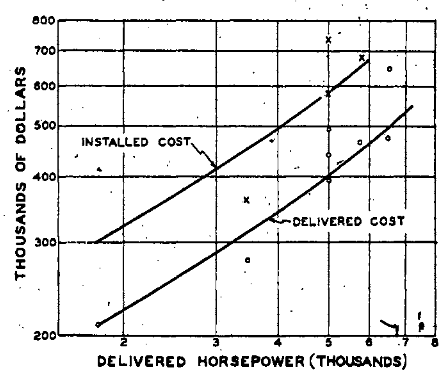


Fig. 1—The delivered and the installed cost of combustion-gas-turbine-driven centrifugal compressors without regenerator.

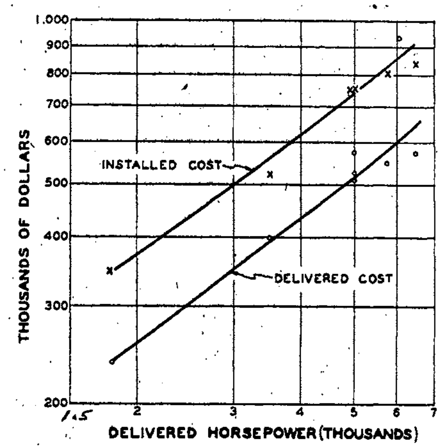


Fig. 2—The delivered and the installed cost of combustion-gas-turbine-driven centrifugal compressors with regenerator.

*Southern California Gas Co. Portion of paper presented at Pacific Coast Gas Association, Transmission Conference, Bakersfield, Calif.

TABLE 1—DESIGN SURVEY OF COMBUSTION GAS TURBINES

Manufacturer—	Allis-Chalmers Mfg. Co.	Brown Boveri Corp.		Clark Bros. Co., Inc.		General Electric		Westinghouse Electric Corp.		
General characteristics:	R	NR	R	NR	R	NR	R	NR	NR	NR
Rated capacity, hp.	3,500	5,100 kw.	4,700 kw.	5,500	5,500	5,000	5,000	1,850	5,000	6,500
Speed, r.p.m.	5,180	....	....	7,000	7,000	5,000	5,000	8,750	5,000	5,000
Altitude, ft.	....	....	....	14.2 psia.	14.2 psia.	1,000	1,000	....	1,000	1,000
Ambient temperature, °F.	....	68	68	60	60	80	80	80	80	80
Performance:										
Thermal efficiency, % based on L.H.V. of fuel	30	19	24	20	25	17	25	16	20	21
L.H.V. of fuel	....	....	....	...	1,029	1,020	1,020	1,000	....	....
Turbine:										
Inlet temperature, °F.	1,500	1,100	1,100	1,400	1,400	1,450	1,450	700 to 1,350	1,350	1,350
Stack temperature, °F.	....	...	....	....	...	...	566	....	....	....
Number of shafts	2	1	1	2	2	2	2	2	2	2
Shaft speeds, r.p.m.:										
Air compressor	5,180	....	...			6,700	6,700	8,750	5,000	5,000
Gas compressor	5,180	....	....	7,000	7,000	2,000-5,500	2,000-5,500	8,750	5,000	5,000
Number of turbine cylinders	2	1	1	2	2	2	2	2	2	2
Number of turbine cylinders in series	0	....	....	2	2	2	2	2	2	2
Combustor type	Double (2 in parallel)	Single	Single	Single reverse flow	Single reverse flow	Multiple (6 in parallel)	Multiple (6 in parallel)	Multiple (12 in parallel)		Multiple (6 in parallel)
Regenerator:										
Efficiency, %	60	75	75	....	75	....	....	....	....	....
Air discharge temperature, °F.	....	....	....	....	....	....	800	....	....	....
Air compressor:										
Type	axial	axial	axial	axial	axial	axial	axial	axial	axial	axial
Number of stages	20	....	....	10	10	14	14	....	16	16
Discharge pressure, psig.	44	....	....	....	....	70	70	5 to 60	75	75
Compression ratio	4	....	....	....	....	....	....	....	....	6
Compression efficiency, %	85+	....	...	...	....	....	....	....	....	....

R—Regenerator. NR—No regenerator.

PART III

Description of Texas-Eastern Gas Transmission Corporation Station and Pipe Line facilities, electric power and fuel usage, and economics of gas transmission under actual operating conditions. (Big Inch System).

The expansion of natural gas pipe lines since the last war and the use of centrifugal compressors is the most significant fact in the gas industry today. One of the prime factors in this nationwide expansion of pipe lines was the conversion of the former Big Inch and Little Big Inch pipe lines to gas transportation by Texas Eastern Transmission Corporation.

In the wake of this one project came the headlong rush of construction of new lines and expansion of old ones which continues today. New areas have been opened up to natural gas service and the whole tempo of the industry has been stepped up to a point exceeding the most optimistic estimates of a decade ago.

The significance of the "Inch" lines in the post-war growth of the natural gas industry is easy to appraise. Here was a pipe line system "in-being" which at one stroke not only brought essential new supplies of natural gas to the largest traditional market, the Appalachian area, but at the same time pointed the way to new economic life for the manufactured gas utilities of the eastern seaboard.

The conversion of the "Inch" lines from oil to gas transmission was a unique project, so it is only natural that the operating history of Texas Eastern Transmission Corporation would be of interest to many engineers.

Operations actually began May 1, 1947, only two months after Texas

Eastern's successful bid for the "Inch" lines and more than six months before the $143,127,000 purchase was actually consummated.

This system is unique in many ways. Perhaps the most interesting difference between Texas Eastern's operations and thos of other pipe line companies is the use of electrically-driven centrifugal compressors in many of the pump stations. Although there is nothing new about the principle of centrifugal compression of gases, it was not until the end of the war that compressors were developed that would operate successfully under the high pressures of 750 psi or more that were required on long distance natural gas transmission lines.

The invention of these compressors was of particular benefit to Texas Eastern because on the "Inch" lines there were a series of idle centrifugal oil pumping stations which could be converted through the use of the new compressors. This conversion plan became the keynote of Texas Eastern's construction programs. Actually "conversion" is a misnomer. For, not only were the old oil pumps removed, stored, and replaced with the new compressors but the old buildings were torn down and replaced with new steel structures and masonry fire walls. The foundations themselves were extensively altered to accommodate the new shaft seal-oil systems for the compressors. Station piping was changed; multiple gas scrubber units installed; and new auxiliary equipment of all types, including explosion-proof phones and wiring was put in. New motor and cylinder-operated steel valves were substituted for cast-iron valves on headers, station suction and discharge lines, in compressor buildings and other locations. At each station these were co-ordinated into a single-control, gas-operated, emergency shutdown system.

Exhibit "D" is shown to illustrate the geographical location of the pipe line, along with station locations, the gas fields from which gas is supplied, and the location of the customers to whom gas is delivered.

It can be noted that the 20" OD - 10/32" wall thickness pipe line originated in the coastal area near Beaumont and Houston, whereas the 24" OD - 12/32" wall thickness pipe line originated near Longview and the great East Texas oilfield. The two pipe lines joined at Little Rock, Arkansas (Station 5); thence the two pipe lines were laid on the same right-of-way in passing to the East coast in New Jersey.

At the time these pipe lines were constructed (1942), there were 26 pump stations for the 24" line, each containing 3 - 1500 HP, 1800 RPM motors driving single stage pumps operating in series and capable of delivery of about 325,000 barrels per day with 720 psi pressure with the stations spaced about 50 miles apart. The 20" line (for products) had 28 pump stations, each containing 3 - 1250 HP - 3600 RPM motors, capable of pumping about 200,000 barrels per day with station spacing of approximately 60 miles until reaching Station #5.

The 1500 HP motors could not be used for gas compression with centrifugal compressors, because the speed was only 1800 RPM and speed increasing gears would have become necessary, since the centrifugal limitations of the impellor prevented the use of speeds in their design of not under or lower than 3600 RPM. Consequently the 1500 HP motors were re-wound for 3600 RPM and HP ratings of 1750 and 2000 HP output.

The capacity of the 20" line was 61.8% of the capacity of the 24" line. $(19.375^{2.65}/23.25^{2.65})$.

Thus under given conditions for 50 mile station spacing initial pressures of 750 psig, and compression ratios of about 1.5 to 1.6 average, the 24" line would handle 340,000,000 CFD and the 20" pipe line could handle 210,000,000 CFD, or a total of 550,000,000 CFD in the parallel pipe lines. (See Table 6 for actual delivery of gas.)

In the follwing Table 5, the horsepower at each station is shown, along with the type of equipment installed, and the distance in miles between the booster stations from Texas to Station #21, where the original Big Inch system (20" and 24") joins with a new 30" line, a portion of which is shown in Exhibit "E". (Oil & Gas Journal, Sept. 27, 1951.) Electric power average costs are also tabulated, as prevailing in early 1950.

Data have been obtained from Texas Eastern Gas Transmission Company covering their operations for the month of October, 1951, showing the volume of gas delivered at various stations in Louisiana and Texas, the volume of gas and its constituency passing each station along with average gauge suction and discharge pressures, and other pertinent information, so that a fairly accurate determination of actual pumping costs can be made. Table 6 is shown containing as much data as necessary for such determination. The data is only tabulated for the operation from Station 5 to Station 9, a distance of 215.64 miles (Table 5) or average distance between stations of 53.80 miles. This section of the pipe line is a twin line consisting of a 23.25" ID line parallel with a 19.375" ID line. Three of the pump stations are electric motor driven centrifugal compressors, (5, 7, and 8) whereas Station 6 is a gas engine driven reciprocating compressor station. The two pipe lines are manifolded together with equal pressures on each line. The results of this operation will show the energy and fuel requirements of a typical gas transmission line as compared to theoretical costs obtained from the design data included in Part II.

It can be noted from Exhibit "D" that the 24" line from the East Texas gas fields joins up with the 20" line from the Gulf Coast gas fields at Station 5, thence both lines pass along on same right-of-way to Eastern

TABLE 5 (1950)

Equipment Installed on Texas Eastern Gas Pipe Line

Station	Miles to next Station	Number of Units	HP Reciprocating	HP Centrifugal	Av. Power Cost ¢/KWH basis Jan-Jul 1950
E (20")	63.021	5	5,500		
F	56.474	4	4,400		
G	60,334	6		7,500	.58
2 (24")	52.428	4		8,000	.705
3	53,393	8	8,300		
4	54.532	4		8,000	.538
5(20"-24")	54.904	10		15,250	.536
6	54.494	14	15,400		
7	52.58	10		19,000	.611
8	53.67	10		13,500	.615
9	44.856	14	14,000		
10	53.616	8		13,000	.988
11	56.694	13	13,000		
12	42.50	10		16,500	.785
13	58.872	10	10,000		
14	41.225	10		19,000	.79
15	54.664	6		9,250	.775
16-B			5,500		
16	57,683	10		14,000	.931
17	48.057	8		16,000	.945
18	52.084	8	8,800		
19	52.428	6		12,000	
20		5		10,000	
Totals		113 Elec 85 Recip	88,900	188,750	x .732 x average

TABLE 6

Actual Data From Texas Eastern Gas Transmission Company

October 1951: Average gas condition: Sp. Gr. .609, Temp. 80° F., Super-compressibility .108, Base 15.02 psia, Pipe Line Eff. .91, K 1.3.

Station	Miles to next Station	Energy KWH	Cents/ KWH	MCF Flow Past Station
5	54.90	9,228,000	.585	16,639,485
6	54.49			16,543,717
7	52.58	8,027,302	.745	16,543,417
8	53.67	7,683,030	.762	16,521,912
Total	215.64	24,938,332	.695	
Average	53.8	1.505/MCF		16,568,300

Table 6 continued:

Station	MCF Daily	Average KW	Engine BHP	Engine Fuel/MCF
5	537,000	12,400		
6	533,500		15,400	95,597
7	533,500	10,780		
8	533,000	10,300		
Total	535,000	33,480	15,400	
Average	535,000			per BHPHR 8.33 CF

KWH/1000CF/100 miles .923

Seaboard.

From above Table 6 it can be shown that the engine fuel economy is (11,450,000 hp hrs.) 8.33 CF per hp-hour average in Station #6. Also it is noted that it requires .692 hp hrs to pump 1000 CF of gas through 52.58 miles of line, or 1.32 hp hrs per 100 miles with gas engines and reciprocating compressors.

The electric stations, 5, 7 and 8, require 33.480 KWH per Hr. electric input while pumping through 163.06 miles of line or 1.505 KW-hr and .920 KW-hr per 1000 of per 100 miles.

Another matter disclosed from the operating data which should receive attention, is the great amount of lubricating oil required by the engine reciprocating compressor station.

The operating data reveals that (May 1950) 85,000 hp of engine stations operating 92% of time, and 85% of load, required 21,391 gallons or .45 gals per 1000 hp-hrs, whereas, in the case of the electric motor centrifugal stations 181,000 hp operating 82% of time and 94% loaded, the lubricating oil required amounted to 996 gallons or .0000087 gals per 1000 hp hrs. With this type of lubricating oil worth 50¢ per gallon, the ratio becomes 22.5¢ to .00043¢ per 1000 hp hrs.

Another interesting matter disclosed by the operating data, is that approximately 1% of all gas received is lost or unaccounted for. No doubt this loss can be attributed to the accuracy of meters, and changes in temperature and volume calculations as the gas is metered many times along the pipe line, at each station, and as it is put into the line by the supplier, and then when sold to the customer. However, this percentage of unaccounted for gas seems to prevail throughout the industry.

SUMMARY

Since the installed cost of the "Big Inch" system was about $147,000,000, and later purchased by Texas Eastern Gas Transmission Company for $143,000,000, it was thought that the cost of pumping today would be of interest, to see how it compares, in a broad sense, with the theoretical costs determined in Part II wherein the most economic pipe diameters and types of motive power were selected.

It is estimated that the portion of the present system from Station 5 to Station 9, involving 215.64 miles of 24" and 20" line cost as determined in 1943 about $75,000 per mile, or a total of $16,150,000, to which has to be added the 15,400 HP gas engine entirely new station at #6 at $200 per hp or $3,080,000 and rehabilitating the three electric stations of 47,750 hp at $75 per hp or $3,570,000 at #5, 7 and 8, making a total investment in that portion of the whole pipe line $22,800,000. The pipe line handles about 535,000,000 CF daily or 195,000,000,000 CF per year.

Table 7 is shown to tabulate all the pumping costs, along with fixed charges on the pipes and stations based on actual operating results from Table 6, all expressed in cents per 1000 CF per 100 miles.

Note from Table 6 that the average daily gas flow through the 24" and 20" lines is 535,000 MCF, and the average distance for the electric stations is 53.8 miles.

Since the average electric load per station is 11,160 KW and the assumed motor efficiency is 94%, the shaft HP becomes 14,066 representing 26.3 HP per million CF per day.

By referring to Figure 2, it is noted that the average compression ratio becomes 1.62. The initial discharge pressures are limited to 765 psia and the corresponding value of "F" for use in Figure 3, becomes 6,750, which means that the value of "Q" becomes 410 M^2 CFD for the 23.5" pipe line.

TABLE 7

(20" and 24") (Delivery 195 M^3CF)

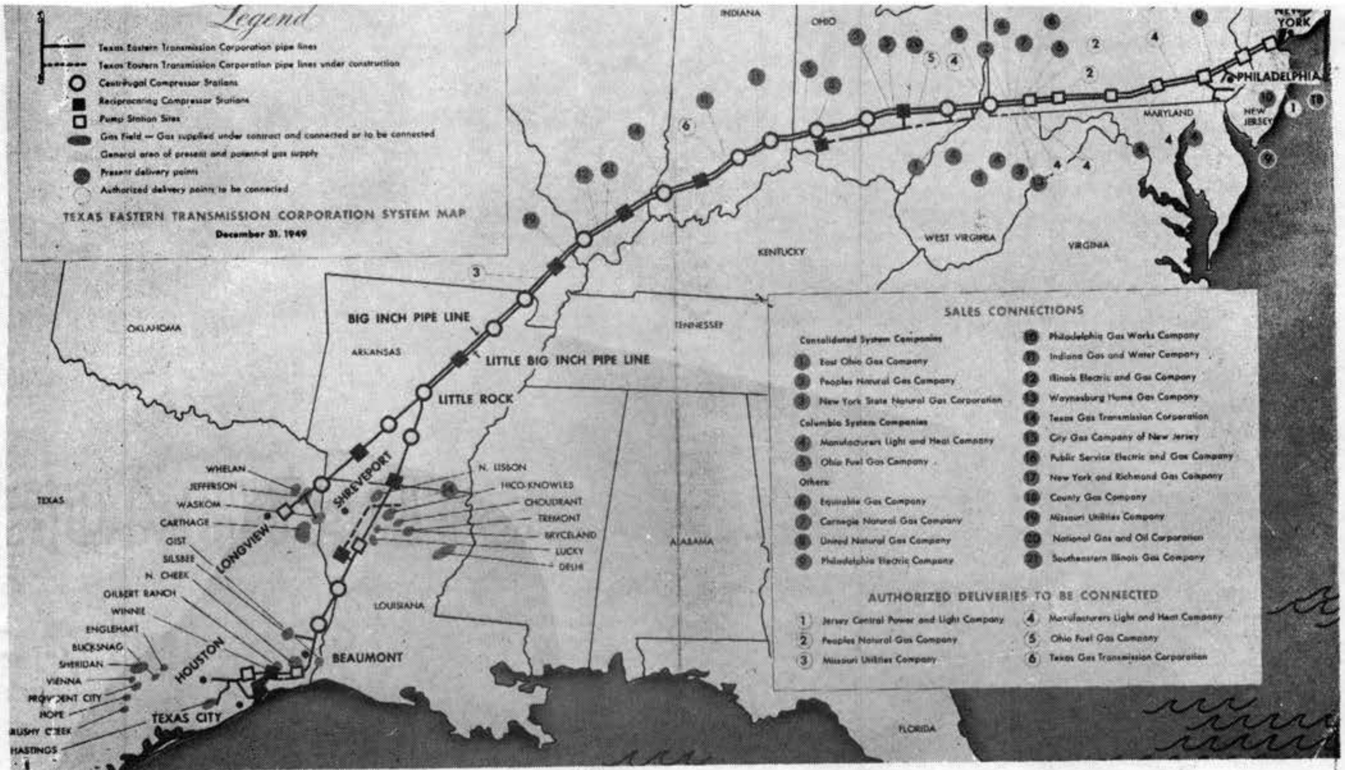
Fixed Charges	Annual Costs $
215.64 miles of 24" and 20" @ 22% on $16,150,000	$3,560,000
Station #6 Gas Engine @ 23% on $3,080,000	710,000
Stations #5, 7 and 8 Electric @ 23% on $3,570,000	824,000
Total Fixed Charges	$5,114,000

Operations (Pumping 215.64 Miles)

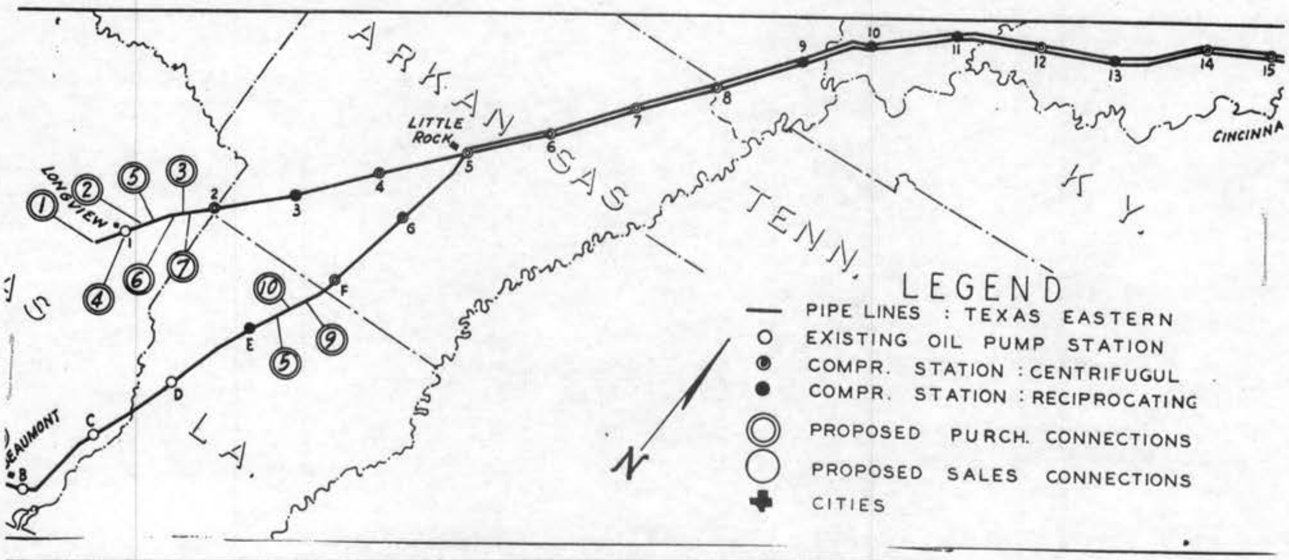
Station #6 52.58 miles	Labor:	31 men ? (N. A.)[x]	$ 124,000
	Fuel:	.692 hp hrs, 8.33 cf/hp hr 195,000,000 MCF @ 26¢ MCF	292,000
Stations #5, 7 and 8 163.06 miles	Labor:	60 men ? (N. A.)[x]	240,000
	Electric Power:	1.505 KWH x 195,000,000 MCF or 294,000 MKWH @ Average Cost (Table 6) @ .695¢/KWH	2,050,000
Total Operating Expenses			$2,706,000
Total Fixed and Operating Cost, Pumping 215.64 Mi.			$7,820,000
Cost per 1000 CF per 100 miles (including allowable return)			1.68¢

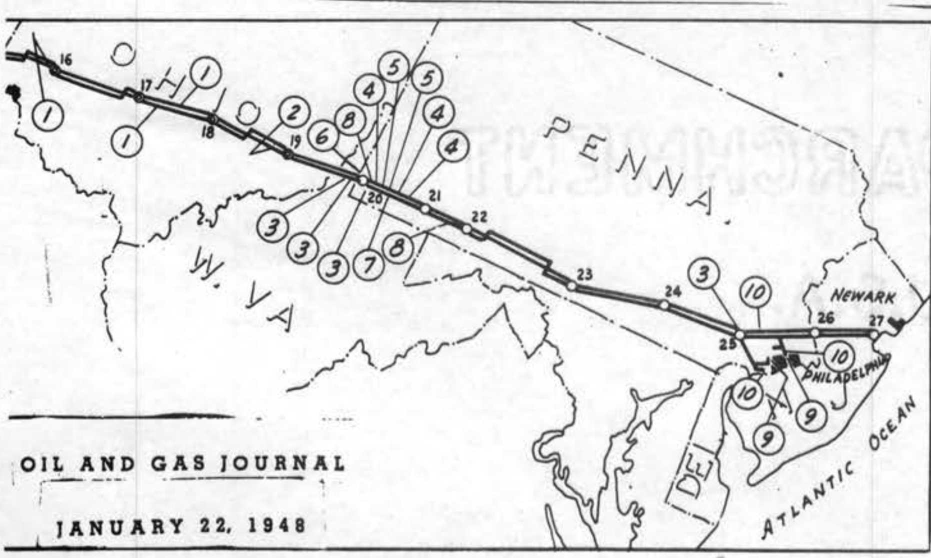
[x] N. A. (Not Available)

After correcting this value of "Q" for 80°F. temperature, specific gravity .61, pipe line efficiency .91, base 15.02 psia to 14.4 psia and reduction in diameter from 23.5" to 23.25", the multiplier becomes .812 resulting in a theortical flow of 333 M^2 CFD. Since the 20" line, 19.375" inside diameter will handle .618% of the flow in the 24" OD line, the total flow through both pipe lines becomes 535 M^2 CFD. This calculation made from actual capacities and known conditions verifies the working theoretical values as depicted in Figure 3.



March, 1951 EXHIBIT "D" PIPE LINE NEWS 25



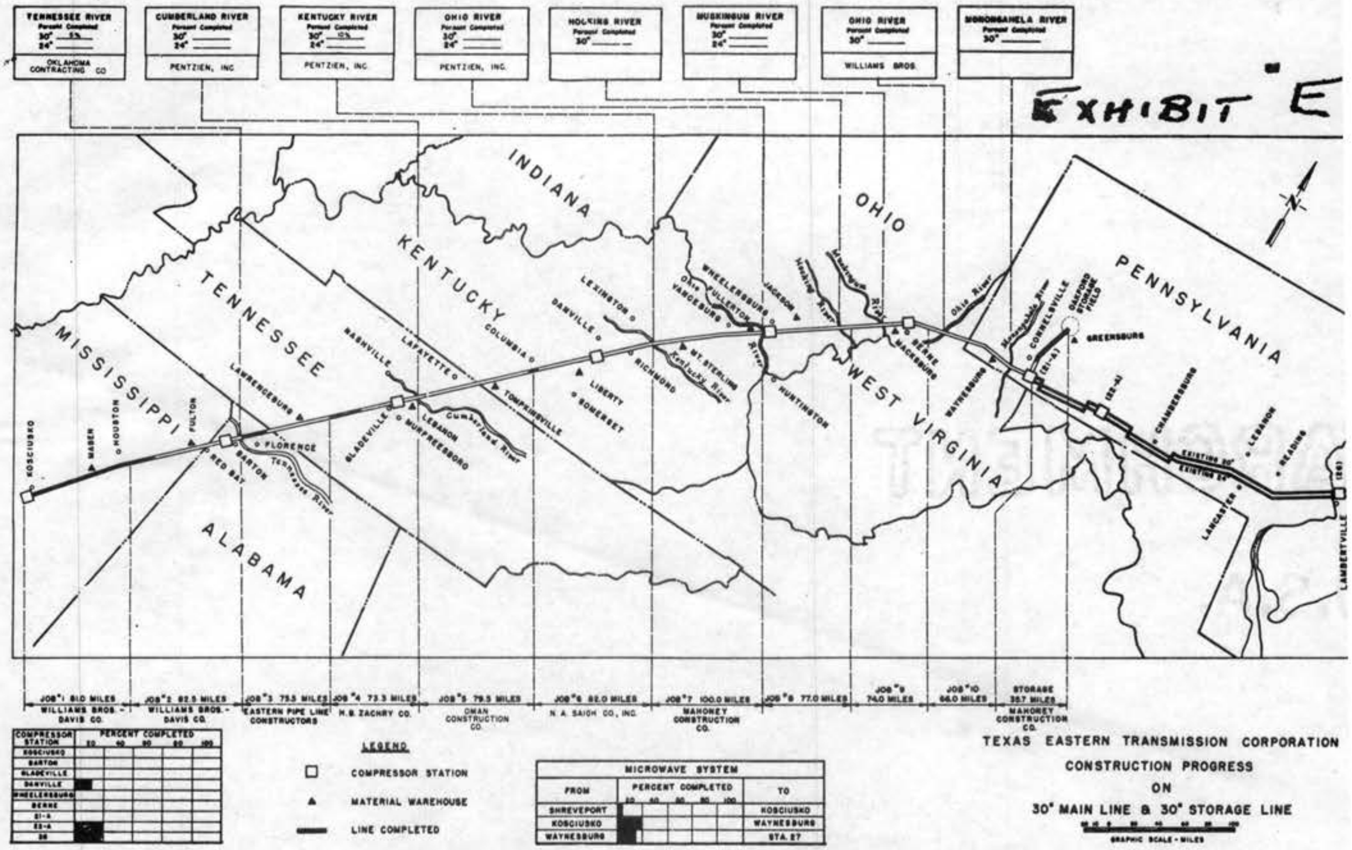


MAP OF TERRITORY SERVED BY TEXAS EASTERN TRANSMISSION CORP.

Purchase connections: (1) Lone Star Gas Co.; (2) Whelan Bros.; (3) Ark. La. Gas Co.; (4) United Gas Pipeline Co.; (5) H. L. Hunt; (6) P. H. Pewitt; (7) Stewart Petroleum Co. and Atlantic Refining Co.; (8) McCarthy Oil & Gas Co.; (9) Hassie Hunt Trust; (10) Southwestern Production Co. and The California Co.

Sales connections: (1) Ohio Fuel Gas Co.; (2) East Ohio Gas Co.; (3) Manufacturers Light & Heat Co.; (4) Equitable Gas Co.; (5) Carnegie Natural Gas Co.; (6) United Natural Gas Co.; (7) New York State Natural Gas Corp.; (8) Peoples Natural Gas Co.; (9) Philadelphia Gas Works Co.; (10) Philadelphia Electric Co.

EXHIBIT E



TEXAS EASTERN PROJECT.—This map shows various spreads contracted by Texas Eastern Gas Transmission Co. on its Mississippi-to Pennsylvania natural-gas line.

Texas Eastern Project

Big expansion program including 791-mile gas line from Mississippi to Pennsylvania moving ahead on schedule

F. Lawrence Resen

SHREVEPORT. — Construction is moving ahead on Texas Eastern Gas Transmission Corp.'s 791-mile, 30-in. natural-gas line from Kosciusko, Miss., to a tie-in point with its existing system near Connellsville, Pa.

As of the first of the month, 150.1 miles of line had been laid, 325 miles of pipe shipped from the factory, construction started on a 30-in. crossing of the Tennessee River, and 740 miles of right-of-way purchased.

Present schedules call for completion of the line as far north as the first crossing of the Ohio River by January 1952.

The over-all construction program, which will result in a total system of over 4,200 miles and call for sales of gas at the rate of 1.2 billion cubic feet daily, also includes development of a 19,000-acre natural-gas storage field in western Pennsylvania, 12 new compressor stations, and a 35-mile, 30-in. connecting line from Connellsville to the storage field. Brown & Root, Inc., Houston, is general contractor on the entire expansion project.

Capacity.—The new line will have a capacity of 400,000,000 cu. ft. of gas daily, but by use of the storage field and facilities east of Connellsville, increased sales capacity over the system's present capacity will be 465,000,000 cu. ft. per day. Cost of the line, plus new compressor stations, will be approximately $99,200,000. Coupled with development of the storage field, cost will total around $114,300,000.

Compressor stations.—Compressor stations along the line, with their capacities, will be: Kosciusko, 12,500 hp.; Mattes, Pa., 6,600 hp., eventually to be increased to 11,880 hp.; Danville, Ky., 8,800 hp.; Connellsville, Pa., 4,400-hp.; Lambertville, N. J., 4,400 hp.; Chambersburg, Marietta, and Phoenixville, Pa., aggregating 30,000 hp.; Barton, Ala., Gladeville, Tenn., and Wheelersburg, Ohio, 7,500 hp. each; and Berne, Ohio, 10,000 hp.

New compressor stations will provide 70,080 additional horsepower fo the Texas Eastern system, and the re activation of four pumping station will provide 34,400 hp., giving th total system an over-all compresso horsepower of 381,880.

River crossings.—Oklahoma Contracting Co. has contract for the Tennessee River crossing. Other river-crossing contracts were let to Pentzien, Inc. for the Kentucky-Ohio job and Williams Bros. for the Ohio-West Virginia job. Crossings not yet contracted include the Cumberland, Kentucky Hocking, Muskingum, and Monongahela rivers.

Storage project.—Texas Eastern has joined with New York State Natural Gas Corp. in developing the Oakford underground storage project 35 miles north of Connellsville. The reservoir is a substantially depleted gas field with an original capacity of 500 billion cubic feet. In addition to the wells already drilled, several new ones will be drilled to increase deliverability of gas both to and from the field.

A compressor and injection station costing $7,500,000, will be erected at the field with a total horsepower of 30,000. In the initial stages, 45 billion cubic feet of gas will be injected as a base, and 60 billion cubic feet deposited against future withdrawals.

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