Proc. 11th New Zealand Geothermal Workshop 1989

FIELD TESTING A DOWNHOLE PNEUMATIC TURBINE-POWERED DRILLING MACHINE AT THE GEYSERS

Philip W. Johnson and William C. Lyons

New Mexico institute of Mining and Technology

ABSTRACT

A new air turbine-powered drilling machine has been successfully tested at the Geysers Steam Field. The machine consists of a turbine, a transmission, and a thrust bearing assembly contained inside a nine-inch housing. Special seals, bearings, lubricants, and alloys were used in construction so the tool could operate at up to 450° F. Before the Geysers test the hardware was bench tested and field performance was predicted with a numerical simulation. The field tests, run both at normal geothermal temperatures in Farmington, New Mexico and at high temperatures at the Geysers, verified the accuracy of the predictions and demonstrated the viability of the design. This machine provides the first technology for making course corrections while drilling with air.

INTRODUCTION

The Pneumatic Turbine Partnership of Santa Fe, New Mexico, under the direction of Dr. William Lyons, has developed a downhole pneumatic turbine-powered drilling machine.

In the first stage of development, the thermodynamic behavior of the machine was investigated, first by Than-Van-Nguyen, and later by P. W. Johnson. These rtudies indicated that the tool was unlikely to experience excessive cooling due to energy consumption by the turbine motor, and set the stage for further development.

A ready market for the tool was found at the Geysers Geothermal Field near Santa Rosa, California. Hells at the Geysers provide naturally occurring dry steam to power plants that generate about 1790 MW of California's electricty. New wells are continuously being drilled to expand and supplement the power production. The wells are drilled directionally to reduce the cost of surface piping, air is used as drilling fluid in the producing zones since the steam tend8 to dry muds into hard clay which plugs the steam-producing fissures, and the formations are hard, metamorphic rocks that do not usually collapre. In addition, large-diameter boreholes are required to conduct the tremendous volumes of steam, and this eases the size problem in designing a prototype drilling machine.

Armed with the results of theoretical studier and the economic analysis of the Geysers market, funding to develop and test the hardware was procured through the Hew Mexico Research and Development Institute (NMRDI), the Geothermal Drilling Organization (GDO), and the private investors who formed the Pneumatic Turbine Partnership (PTP). In the late winter of 1987 the first two prototypes were assembled and tested in Farmington, New Hexico.

Based on tho results of those initial tests, design modifications were incorporated into the prototype tools. The modified prototypes were again bench tested at the Clay-Groomer Machine Shop in Farmington and then field tested at the Geysers.

DRILLING MACHINE DESCRIPTION

The drilling machine consists of four components: a turbine motor to provide the power, a transmission to reduce the turbine rotary speed to an acceptable bit speed and increase the turbine torque to an acceptable bit torque, a thrust boaring assembly to carry the weight on bit and tho dynamic drilling loads, and a housing to contain and support the turbine and the transmission (ree Figure 1).

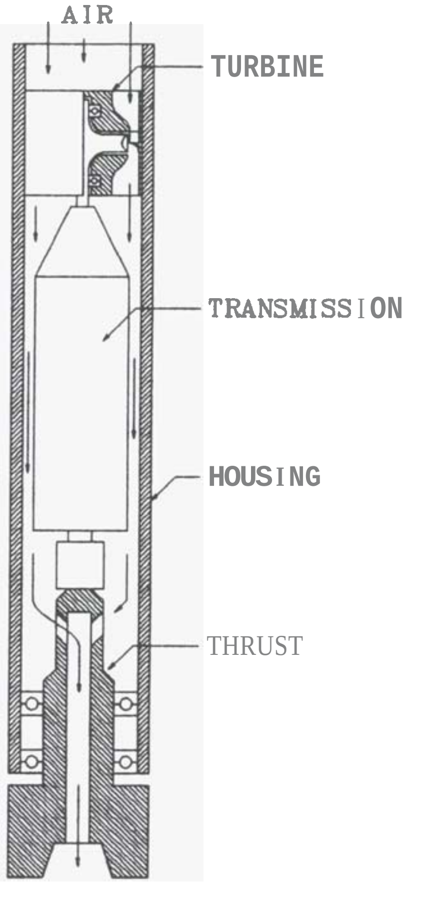

FIGURE 1. Schematic diagram of tho pneumatic turbine-powered drilling machine.

Tho air flow that ir exhausted from the turbine is deflected by an aerodynamic cone on tho top end cap of the tranrmirrion into an annular parrage between tho outride of the transmission and the inside of the housing. The air followr thir annular parrage part tho tranrmisrion, into a chamber between tho tranrmirrion and the thrust assembly and then through three ports into the rotating thrurt rhaft and out through the bit.

The single rtage, impulse turbine motor that powers the drilling machine is a compact, stainless steel unit about 6 inches long and about 7.4 inches in diameter. It mounts directly on the inside of tho housing and is supported by dowel pins and radial bolts. The turbine nozzles have a minimum width of about 1/8 inch, and are protected from plugging by debris by placement of a filter screen in the collarr upstream of the turbine.

Functionally the turbine conrirtr of a rtator to convert tho internal energy of the air to kinetic energy, a rotor to convert the kinetic energy of tho air into shaft rotational energy, a rotor suspension rystem consisting of enclored radial/thrurt bearingr at each end of the rotor rhaft, and a housing to rupport the other componentr. The bearings are lubricated with grease and are designed to carry both the radial loads impored by the rotation of the rotor and the axial loads produced by the pounding action of the bit. The grease is retained in the bearing cavities by lip seals, and for operation at the Geysers the grease and seals were both selected to resist temperaturer exceeding 460° F.

Under optimum condition8 the turbine can produce about 50 horrepower, but field tests demonstrated that less than 20 horsepower is usually required.

The output end of the rotor rhaft terminates in a male spline for connection to a transmission assembly that is about 6.3 incher in diameter and 4 feet long. Rectangular keys that protrude from the wall of tho houring mate with slots in the outride of the lower end of the tranrmirrion care to provide lateral and axial rupport. There keys carry both the static and dynamic weight of the transmission and the entire reactive torque of drilling. The assembly is centered within tho houring by there rue keys and by lugs on tho top end cap that contact tho inride wall of tho housing.

In the tranrmirrion, four retr of planetary gearr are used to reduce the turbine rpeed and increase the turbine torque by a factor of 168. As an example, consider the firrt rtage of the tranrmirrion. At tho center is a 32-tooth sun gear, surrounded by four 26-tooth planets. The planet shafts are mounted on bearings in a carrier cage, and they run within an 84-tooth ring gear that is machined into the tranrmirrion houring. The run gear is fixed on the lower end of the input rhaft, and the upper end of the input shaft terminater in a male rpline that ir connected to the turbine by a 7-inch shaft with female splines at each end. When the turbine rotater the input rhaft, the run gear rotater the planets. The planet gears rerpond by traveling around the ring gear, thur revolving around the run gear and rotating the carrier cage. The carrier cage is integrally connected to a shaft that terminater with the recond-rtage run gear. The second rtage has the same gearing as the first stage, a d the third and fourth rtager have 19 teeth on the run gear, 15 teeth on tho planet gears, and 49 teeth on the ring gear. At each successive rtage the carrier shaft is increared in diameter to accommodate the increased torque. The carrier shaft of the fourth rtage is also the output rhaft of the tranrmirrion, and it terminater in a male spline.

The third- and fourth-rtage gearr are lubricated by immersion in an oil bath. The bath also server as a sump from which a pup driven by the recond-stage carrier rhaft takes oil for rpray lubricating the gears and berringr in the firrt and recond stages.

Under low load conditionr, tho turbine rpeed can become so high that if the firrt rtage gearr were immersed in oil, the work of squeezing the oil from between the meshing teeth could cause failure by fatigue (thir phenomenon is called hydrodynamic wedging). The oil is retained in the tranrmirrion by double lip seals on tbe input rhaft and the fourth-rtage carrier rhaft. At the Geyserr, high-temperature seals are required aad a special gear alloy and lubricant (AeroShell 555, Shell Chemical Company) were required to prevent lubrication failure and resulting gear rcoring and tranrmisrion failure.

Tho thrurt assembly ir 9 incher in diameter and about 4 feet long, and it conrirtr of a rhaft that rotater on bearingr within a fixed external care. The care ir attached to the lower end of the houring with a large Stub Acme thread. The shaft is about 4 inches in diameter in the upper three feet and 9 incher in diameter in the lower one foot, At the upper end is a male spline that is attached to the output rhaft of the transmirrion by a connecting shaft that has female splines at each end. The expanded lower end of the thrurt rhaft contains a standard 6 1/2-inch API regular joint for connection to the rock bit, a rock bit rub, reamer or other component of the bottom hole assembly. There are three ports into the shaft about 6 incher below the rpline at the upper end and from there to the bottom end the rhaft is a 2-inch diameter internal air parrage.

The bearing package between the shaft and the care conrirtr of one thrurt boaring to carry the weight on bit, 5 radial bearingr to carry the rotary loads, and a radial/thrust bearing to carry any tensile loads that may be imposed while pulling out of the hole. The bearingr are lubricated with grease and the grease is kept in place by double lip seals just below the portr at the upper end of the shaft and just above the expanded section at the lower end of the shaft.

The houring ir errentially a pipe about 10 feet long and 9 incher in diameter. At the lower end is a male Stub Acme thread for connection to the thrurt assembly case. It the upper en& ir a female Stub Acme threrd for connection to the caring connection rub. the turbine is installed through the upper end, and the tranrmirrion is installed through the lower end, The housing server to rupport the tranrmirrion a d the turbine; to connect and align the turbine, transmission, and thrurt arrembly; to conduct the air into the turbine and from the turbine around the trmrmirrion and into the thrurt shaft; to transmit the back torque from the transmission to the caring; and to rupport the bending loads that may be impored by weight on bit or directional drilling.

THE BENCH TEST

In the air turbine motor the air flow, air tomporature, back prerrure (prerrure at the turbine outlet) and the torque load on the rotor rhaft determine the rotary speed. That is, for a given flow rate, temperature, and back prersure, the rhaft rotary rpeed ir a fixed function of the torque load on the rotor shaft, and this function it a characterirtic of the turbine design.

The entire drilling machine responds to variables analogour to those that control the turbine. Thus the bit speed is a function of the air flow rate, air temperature, bottom hole pressure, and weight on kit or shaft torque. Unlike the turbine motor performance, the drilling machine performance ir not predictable without recourse to mechanical testing. Two characteristics of the tool must be quantified before tho performance can be reliably predicted. First, the effect of the mechanical efficiencies of the turbine, transmission, and thrurt assemblies must be determined, and second, the presrure loss behavior of the tool murt be measured. Therefore, two bench tests, a pressure test, and a torque test were devised and performed in tho rue test facility.

A tubular steel containment vesrel supported upright within a rteel framework war used to simulate the bore bole (see Figure 2). For testing, the drilling machine is centered within this containment vesrel with the output shaft parsing through a seal at tho bottom and the inlet air pipe passing through a real at the top. The air passes through the tool by the normal pathway and is exhausted into the annulus between the outside of the tool houring and the inside of the containment vessel. A constant pressure valve placed in the containment vessel lid is set to simulate typical bottom hole pressures during drilling.

There are six ports for presrure taps in the tool housing wall, two each above the turbine, at the turbine outlet and below the transmission. Hoses from these pressure taps are taped to the tool housing and connected to fittings that penetrate through the containment vessel lid. Hoses from these six fittings and two more for monitoring the back prersure in the containment vessel run to gauger at the control station.

Mounted on the output rhaft outside the containment vesrel is the disk of a large disk brake assembly. The brake padr are on a cage that is centered on the output shaft with bearingr so that the cage can rotate independently of the shaft. A cable runs from the periphery of the cage to a load cell. Thus, friction between the pads and the rotating disk produces tension in the cable and a force on the load cell that can be interpreted in terms of output shaft torque. During terting the entire power output of the tool is converted to heat at the brake; therefore, a rpray of water is used for cooling. An idler wheel running against the outside diameter of the brake disk drives a tachometer to measure the output shaft rotary rpeed.

Standard oil field compressor units are connected to the tool to supply the air flow necesrary for operation. The flow rate war measured by a turbine meter that war loaned to PTP by Haliburton for the duration of the teat. The air temperature was measured at the inlet line with a single standard thermometer, taped to the outride diameter of the line and bound with inrulation.

Testing was performed from the control station. The controls consirted of two ball valves to open or close tho pipeline between the comprerrorr and the tool and a hydraulic pump to control the prerrure between tho brake pads and the dirk. The parameters of the teat were measured with a digital readout of actual air flow rate (later converted to scfm) and gauge mearurements of air temperature, hydraulic pressure in the brake cylinders, disk rotary speed, hydraulic pressure in the load cell (directly interpretable as torque), as well as two measurements of the air pressure at each of four points within the system and one mearurement of the air pressure at the flow meter.

Three people were required to perform the testing in addition to the personnel running the compressors. A mechanic was present to set the back pressure valve, adjurt the water flow to the brake, and monitor the integrity of the mechanical system, an engineer ran the machine by controlling the ball valves and the brake presrure: and a recond engineer recorded data and requested each new stage of the teat.

A typical test was performed as follows. The compressors were started and warmed up with the main ball valves between the compressors and the tool remaining clored. Then the brake pad pressure was pumped to a high level so that no rhaft rotation could occur, and the ball valves were opened. Next the back presrure was adjusted to the desired level and the flow rate, temperature, system prerrurer and torque were recorded. There pressure, flow rate, and temperature data constitute one pressure test, and the torque datum constitutes the firrt point of a torque test. Finally the brake pad pressure was docreased until rotation commenced and stabilized and the torque and rotary speed were recorded. Then the pressure was again reduced and a new torque and rotary rpeed were

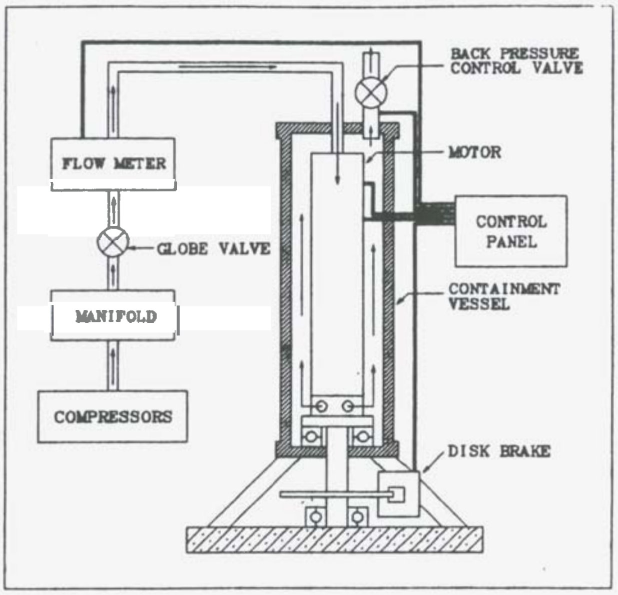

FIGURE 2. Schematic diagram of the bench-terting facility.

recorded, etc. Typically five to ten torque and RPM measurements were recorded at each flow rate and back pressure. Two tools were tested, each at nine different flow rates and back pressures, in preparation for the Geysers field teat.

PERFORMANCE PREDICTION

In the prototype drilling machines, the bit rpeed is controlled by the air flow rate, the rize of the bit jets, the bottom hole temperature, and the weight on bit. For a drilling run tho first three variables are errentially fixed, and the fourth, the weight on bit, is used to control the tool. Thus the driller ir driving the tool by controlling the weight on bit. Under some conditions (open bit jets, for instance) the tool speed can become so high as to cause transmission damage if the weight is removed from the bit. Thus it was necessary to accurately predict field performance before initiating the field teats.

To predict the field performance, three steps of analysis were required. First, the presrure, temperature, and flow rate data from the bench teat were analyzed so that air velocities in the turbine could be predicted for any combination of prersure above the bit, temperature, and air flow rate. **Second,** the torque data were normalized based on ideal turbine performance for the measured air flows, and the normalized data for all the teats were combined to produce a simple linear torque model. And third, the torque and prersure losr models were incorporated into a borehole prersure loss simulation in order to predict field performance.

A least squarer analysis of the bench test prersure data was used to determine the effective L/D ratios of the flow paths through thrust and around tho transmission and the effective area of the ports entering the thrust shaft. Analysis of the prersure data also indicated that the presrure drop across the turbine was tho same as that predicted for an ideal choke with an area the same as that of the turbine stator. The effective values were ured to calculate the presrure drop across the turbine and the air velocity within the turbine for both the bench and field testing environments.

The air velocity within the turbine (as the air exits the stator) and the air flow were used to predict the maximum output torque and maximum speed of an ideal turbine. In an ideal turbine, the maximum torque would occur when the rotor is stationary and the air

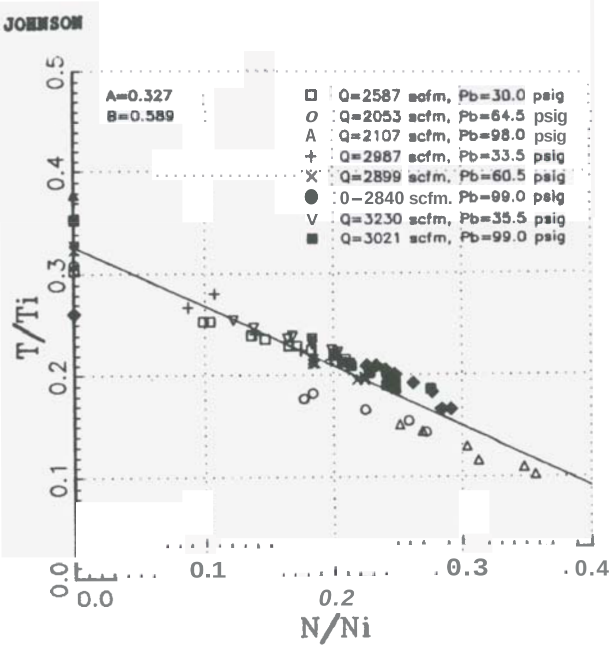

FIGURE 3. The normalized torque versus normalized RPM data from the bench teat of tool number one.

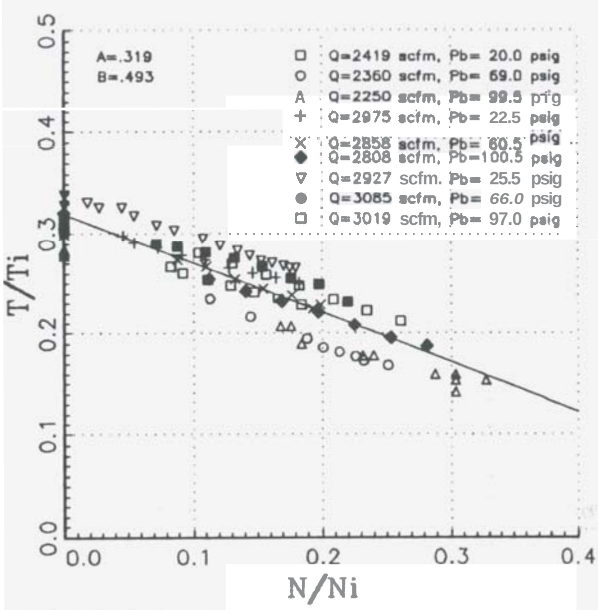

FIGURE 4. The normalized torque versus normalized RPH data from the bench test of tool number two.

flow is exactly reversed by the rotor. At the bit, thir ideal torque is

$$T_i = 2Gr\dot{m}v \quad , \qquad [1]$$

where $\dot{m}$ is the mass flow rate of air, G is the gear ratio, r is the radiur of the turbine rotor, and v ir the air velocity. Likewise the maximum speed of an ideal turbine occurs when the tip of the rotor moves at the air velocity, and at the bit thir ideal rotary rpeed, in rpm, is

$$N_i = 60v/(2\pi Gr) \quad . \qquad [2]$$

The bench teat torque data were normalized by dividing the measured torquer by the appropriate ideal torque and dividing the mearured bit rpeedr by the ideal bit rpeed. Typical normalized torque versus normalized rpm data are plotted in Figurer 3 and 4. Least squares analysis was again ured to fit the normalized torque and rpm data to a simple linear torque model

$$T/T_i = A - B\,N/N_i \quad . \qquad [3]$$

Thur for any field conditions where the air flow rate is known and the velocity of the air in the turbine has been calculated, the maximum bit torque occurs at the rtall condition and can be calculated as

$$T_m = A\,T_i \quad . \qquad [4]$$

and the maximum bit rpeed occurs at the no load condition and can be calculated as

$$N_m = AN_i/B \quad . \qquad [5]$$

During operation the load torque is controlled by the weight on bit, W, the bit diameter, D, and a formation factor, K,

$$T = K\,W^{1.5}\,D^{2.5} \quad . \qquad [6]$$

Thur Equations [3] and [6] can be rolved together to give the bit rpeed as a function of the weight on bit,

$$N = N_i\,[A - K\,W^{1.5}\,D^{2.5}/T_i]/B \quad , \qquad [7]$$

and then [3] alone can be solved to give the bit torque. The rtall weight on bit occurs at N=0, or

$$W = [\,AT_i/(KD^{2.5})]^{2/3} \quad . \qquad [8]$$

Once the torque and rotary speed are known, then the bit power is the product of torque times speed.

Since Equations [1] through [5] and Equations [7] and [8] depend upon the air velocity in the turbine rtator, v, then calculation of thir parameter ir critical to predicting field performance. thir velocity depends on the bottom hole prerrure, the rize of the bit jetr, the L/D ratio of the flow path in the tool below the turbine, the area of the nozzles in the turbine rtator, the air temperature at the turbine, and the mars flow rate of air.

The L/D ratio was determined in the bench teats, and the area of the turbine rtator nozzles is known. The bottom hole temperature and the air temperature are generally equal, and the mars flow rate of air can be measured at the comprersors. The bottom hole prerrure was calculated by Ikoku's method, modified to include the effects of rteam entries at various depthr and variable borehole diameter. All the forgoing parameters were incorporated into a numerical rimulation that was ured to generate practical charta for selecting bit jets and predicting tool output in field situations.

Examples of the rerulting charta of rtall WOB, peak horsepower, and no load bit rpeed versus bit Jet rize are prerented in Figures 5, 6, and 7.

FIELD TESTS

When the GDO agreed to fund part of the development costs of the pneumatic turbine drilling machine, a requirement of the agreement was four hours of field drilling at the Geyserr. This was eventually accomplirhed in two drilling runs, the firrt of which lasted 45 minuter, and the recond of which lasted 3 hourr and 15 minuter. The details and rerultr of the two tests are prerented in Figures 8 and 9.

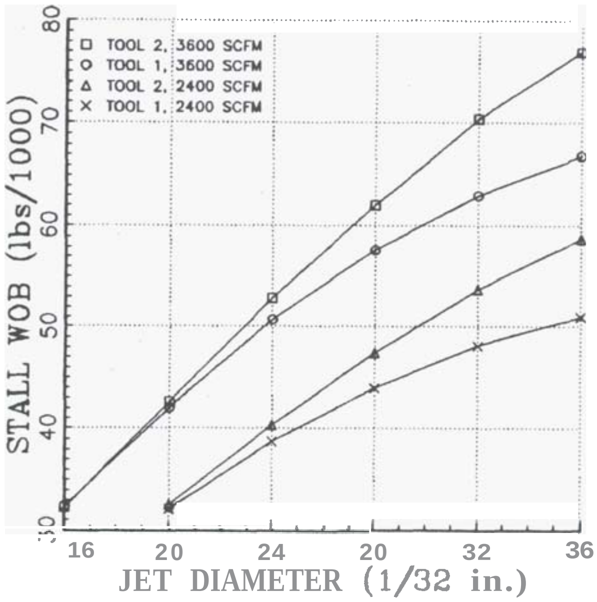

FIGURE 5. The predicted effect of the diameter of the bit jets on the required weight on bit to stall the drilling machine, the Geysers, 2/88.

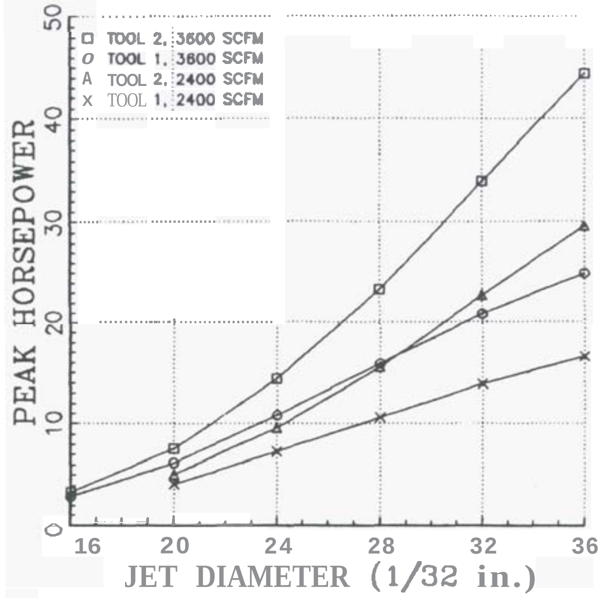

FIGURE 6. The predicted effect of the diameter of the bit jets on the maximum horsepower of the drilling machine, Geysers, 2/88.

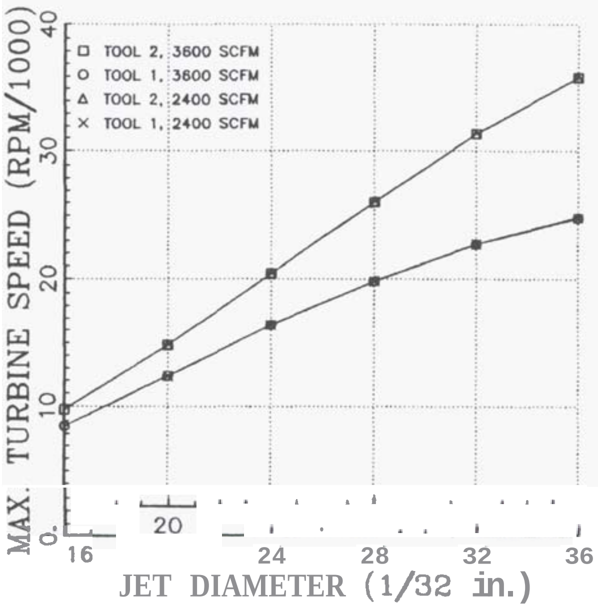

FIGURE 7. The predicted effect of the diameter of the bit jets on the maximum turbine rotor RPM, the Geysers, 2/88.

DISCUSSION

During field testing the only available direct indications about the output of the motor are the stall weight on bit and the surface air prersure. The rate of penetration indicates ruccessful drilling but does not indicate bit rpm or turbine power. Thus, correlation of rtall weight to predicted rtall weight is an important indicator of performance. In the firrt test the predicted rtall weight war 48,000 poundr, and the mearured stall weight was 65,000 pounds. Bowever, the driller mearured about 20,000 poundr of drag on lowering and on picking up weight, so it was believed that the predicted rtall weight was reasonably accurate. In the second tert the predicted rtrll weight war 40,000 pounds, and the mearured stall was 38,000 pounds. It is believed that the predictions of tool performance were suprisingly accurate.

During the firrt test the rate of penetration was 1.5 to 2 tires the normal rate for rotary air drilling in the area. However, 65,000 poundr of weight on bit was required to rtall the tool for connections. This was almost two and one half timer the amount needed for normal drilling and war more weight than the operator wished to carry in the hole. So, for the recond test, 3/4-inch jets were ured in the bit instead of 1-inch jetr. This had the effect of reducing the peak horrepower from 18 to 8 and reducing the stall weight to 38000 poundr. the rate of penetration war of course lowered, but thir war not considered a problem. The main use for the tool in the Geyrers Field is for course correction., and a high ROP is not required.

the oil leak that occurred during the first tert war caured by a design flaw in the transmission meals. After the seals were redesigned, no further leaks occurred. In fact, all the tert evidence indicates that the tool can be run reliably in the geothermal environment.

In the prototype tools, it war necessary to maintain weight on bit as long as air war being delivered to the tool. This was an operational handicap, becaure it required that the tool be stalled by high weight. in order to blow the hole clean prior to connections. The production models that are now in the derign rtage will include a governor system that will allow for operation without weight on bit. The governor will also permit operation without jetting the bit, rerulting in higher powers and higher rates of penetration.

ACKNOWLEDGEMENTS

The authors would like to thank the New Mexico Energy Rerearch Institute, the Geothermal Drilling Organization, the New Mexico Institute of Mining and technology, and the Pneumatic Turbine Partnerrhip for supporting thir work.

FLOW	
	10 degrees inclination
BOTTOM HOLE ASSEMBLY	: bit, Hughes 10 5/8", nozzles jetted to 1" 3 point reamer, Grant pneumatic turbine motor doublo pin XO rub 3 point reamer, Grant monel collar 3 point reamer, Grant nine 8" drill collars jars three 8" drill collars XO sub

DISCUSSION : Before entering the hole, the PTP tool joints were torqued with the tongs to about 6000 ft-lbs, and locked with insert keys. Next, a single 1200 acfm compressor was connected and a rotation tort conducted with part of the flow diverted ta the bypass line and part diverted to the tool. By 2 p.m. the trip into the hole was complete and tha compressors were started with 55,000 lbr WOB. The tool drilled-off weight down to 35,000 lbr, and tha UOB was reret to 60,000 lbr. The tool continued ta drill-off at this weight. Rotation of the drill string was then initiated, and the WOB was ret and maintained in the range of 25,000-30,000 lbr for 22 minutes, during which time the ROP was mearured by timing 1-foot increments of movement at the kelley. Since it was now almost time to make a connection, drill string rotation war ceased and a stall tert conducted. 65,000 lbr WOB were required for rtall, 20,000 lbr of which were believed to be carried by the borehole, and 45,000 lbr by the tool. Efforts to restart the tool were unsuccessful, and later disassembly and inspection revealed an oil leak in in the transmirrion that had led to siezing.

FIGURE 9.
Summary of Field Tort 2
Geysers Steam Field, California, 8/6/88

ROP	: 15 to 27 ft/hr
TOTAL ROTATING TIM	: 3 hours 14 min
AVERAGE WOB	: 25,000 lbs
STALL WOB	: 38,000 lbr
ESTIMATED BIT SPEED	: 43 rpm, all from motor
AIR FLOW	: 2400 scfm
MISTING	: 2 gpm water
STANDPIPE PRESSURE	: 281 psia
BOTTOM HOLE TEMPERATURE	: 250 degrees I
BOREHOLE	: 16" casing, surface-900' 11 3/4" liner, 900-5130' 10 518" open, 5130-5423' 11 degrees inclination
BOTTOM HOLE ASSEMBLY	: bit, Hughes 10 5/8", nozzles jetted to 3/4" 3 point reamer, Grant pneumatic turbine motor double pin XO sub 3 point reamer, Grant monel collar 3 point reamer, Grant nine 8" drill collars jars three 8" drill collars XO sub

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