DEVELOPMENT OF A MODEL TO CALCULATE MECHANICAL SPECIFIC ENERGY FOR AIR HAMMER DRILLING SYSTEMS

A Thesis

by

BOMA JEREMIAH OKUCHABA

Submitted to the Office of Graduate Studies of
Texas A&M University
in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

May 2008

Major Subject: Petroleum Engineering

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Approved by:

Chair of Committee,	Jerome J. Schubert
Committee Members,	Hans C. Juvkam-Wold
	Steve Suh
Head of Department,	Steve Holditch

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ABSTRACT

Development of a Model to Calculate Mechanical Specific Energy for Air Hammer Drilling Systems. (May 2008)
Boma Jeremiah Okuchaba, B.Tech, River State University of Science and Technology, Nigeria
Chair of Advisory Committee: Dr. Jerome J. Schubert

Drilling for hydrocarbons is an expensive operation; consequently operators try to save costs by reducing the number of days spent during this operation. Drilling efficiently with the highest attainable rate of penetration is one of the ways drilling time could be reduced. Real-time monitoring of Mechanical Specific Energy will enable drilling engineers to detect when the optimum drilling rate for a given set of drilling parameters is not being achieved.

Numerous works have been done on air hammers and rock Mechanical Specific Energy. Previous research has shown that Mechanical Specific Energy, which is a ratio that quantifies the input energy and Rate of Penetration (ROP) of a drilling system, is directly proportional to the rock compressive strength being drilled. The Mechanical Specific Energy model utilizes drilling parameters such as ROP, Weight on bit (WOB), RPM, torque, flow-rate, bottom-hole pressure, and bottom-hole temperature to show how effectively energy being put into the drill string is being converted to ROP at the bit.

This research effort proposes a new model to calculate the Mechanical Specific Energy for air hammer drilling systems. A thermodynamic model for the air hammer from which the piston impact velocity and kinetic energy is obtained is presented. To be able to estimate the effective energy delivered to the rock by the hammer, the stress wave propagation model is used and factored into the Mechanical Specific Energy model.

The Mechanical Specific Energy values obtained from the application of this model provide a qualitative indicator of formation pressure changes and a means for drilling engineers to detect when optimum drilling rate is not being achieved. It can be deduced from the model that the impact energy of the hammer is greatly affected by the pressure drop across the hammer and since the hammer accounts for about sixty percent of the energy required for destroying the rock, the ROP can be varied by varying the pressure drop across the hammer.

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TABLE OF CONTENTS

LIST OF FIGURES

CHAPTER I

INTRODUCTION: AIR ROTARY PERCUSSIVE DRILLING DEVELOPMENT

Increase in the world's energy demand, which is associated with a sharp increase in oil and natural gas prices, has outpaced the growth in oil and natural gas production. The oil industry response to this increase in energy demand is to increase oil and natural gas exploration and production. Oil companies are now forced by this high energy demand to drill in very difficult terrain.

Increase in drilling activity is seen across the oil industry, but limited rig supply calls for efficient and faster drilling so as to be able to drill more wells with the limited rig supply within a given period of time. The development and application of a more efficient and lower cost drilling technology will significantly reduce drilling time and cost, consequently, making drilling in deep, hard rock formations and older fields with inherent depleted reservoirs more economical. Under-balanced drilling and air rotary-percussive drilling are some of the technologies developed to overcome these challenges.

Air was substituted for liquid as the drilling fluid on an experimental basis on several wells in the early 1950's[1]. The resultant benefits of air drilling, a type of under-balanced drilling technology, over conventional mud drilling are, higher penetration rates, greater footage per bit, reduction in lost circulation, reduction in formation damage and reduced drilling costs. The need to further increase the rate of penetration during drilling led to the idea of rotary-percussive drilling.

Percussive drilling was first developed by the Chinese four thousand (4,000) years ago

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This thesis follows the style and format of *SPE Drilling and Completion Journal*.

and in its early stage entailed raising and dropping a heavy piercing tool to cut and loosen earth material. The first well drilled by Col. E.L Drake in 1859 was drilled using a percussion-type machine.

Over the years researchers have been able to develop upon the principles of the early percussive drilling machine to arrive at a powerful down-the-hole hammer. Experiments have shown that a combination of rotary air drilling and percussive drilling, referred to as air rotary-percussive drilling, exhibited a dramatic increase in rate of penetration over conventional rotary drilling.

Air rotary percussion drilling is used in many wells drilled in hard rock formations where formation fluid flow is negligible, especially in wells where formation damage, severe fluid loss, differential sticking and low penetration rates are concerns to the drilling engineer. However, the potential and theoretical improvements in drilling efficiency using combined percussion and air rotary drilling is sometimes difficult to achieve because of the vulnerability of the down-hole hammer to lose its ability to drill. The down-hole hammer accounts for 60% of the axial force used in rock destruction during drilling and a proper understanding of the factors affecting the down-hole hammer performance and development of a model that can give drillers an indication of down-hole hammer efficiency is an important step in achieving constant drilling efficiency while drilling with the air rotary-percussive drilling system.

CHAPTER II

LITERATURE REVIEW

E&P companies have always aimed at reducing drilling cost and presently, with the high energy demand, companies are faced with having to drill difficult wells; time is of essence as wells need to be completed quickly so as to drill more wells in order to meet the growing energy demand. One of the ways of reducing drilling time and cost among others is to drill with the highest rate of penetration attainable in a safe and efficient manner. To attain the highest rate of penetration (ROP), drilling engineers need to know the amount of energy required to destroy a unit volume of the various rock formation types encountered during the drilling process and a way of monitoring the energy being delivered to the rock by the drill bit.

Teale[2] proposed the concept of specific energy (SE) in rock drilling in 1965. He derived the specific energy equation by calculating the torsional and axial work performed by the bit and dividing this by the volume of rock drilled. Teale[1] then conducted lab tests that demonstrated the energy per volume of rock destroyed to be relatively constant, regardless of changes in ROP, weight on bit (WOB) or revolutions per minute (RPM). Teale noticed that laboratory drilling data showed the SE value to be numerically equal to rock compressive strength in pounds per square inch (psi), he however recognized that the SE cannot be represented by a single, accurate number due to the heterogeneity of the rock formations and the wide fluctuations of the drilling variables.

This specific energy concept is useful from an operations standpoint because it provides a reference point for efficiency. If the observed SE is close to the known confined rock strength, the bit is efficient. If not, energy is being lost. The value should change as the lithology changes.

Teale's specific energy concept which has evolved into 'Technical limit Specific Energy' and 'Mechanical Specific Energy' (MSE), has been used for determining the drilling efficiency for drill bit designs and in specialized field applications.

Pessier and Fear[3] gave a practical discussion of MSE and derived an equation for ROP based on the specific energy equation derived by R. Teale. They made modifications to Teale's specific energy model by substituting an equation they derived, that expresses torque as a function of WOB, bit diameter and a bit-specific coefficient of sliding friction. They further showed, by conducting tests, that under atmospheric drilling conditions the MSE is approximately equal to the unconfined compressive strength of the formation drilled and that when drilling under hydrostatic pressure the mechanical efficiency which is essentially the inverse of specific energy, dropped significantly. Their analysis of field data revealed a good correlation between their simulator model and field results. Bit selection exercise and the diagnosis of failures and drilling practices became more accurate and less ambiguous because of the use of mechanical efficiency, specific energy input, and a bit-specific coefficient of sliding friction as key indexes of drilling performance.

Waughman *et al.*[4], in their research, developed a concept that entailed real-time monitoring of specific energy data in combination with measurement while drilling (MWD) data and sonic data, that enhanced the decision process of when to pull the bit out of hole. They outlined a guide on applying the specific energy monitoring technique to the field. The initial stage in the application of the specific energy concept is to benchmark new bit specific energy in different formations then using these values to assess the bits dull state. The concept has been proven to work in synthetic based mud systems and water based mud treated with anti-balling chemicals.

Apart from the above two papers numerous publications exist that apply the specific energy concept as a basis for bit performance and selection; however Curry *et al.*[5] applies specific energy as an index to facilitate drilling performance evaluation.

Curry *et al.*[5] developed a method to represent the difficulty of drilling a particular formation in its down-hole pressure environment using the concept of Mechanical Specific Energy. An algorithm was developed to estimate the technical limit specific energy, from wire-line sonic, lithology and pressure data. They stated that, the technical limit specific energy represents the lowest specific energy that can be reasonably expected for a particular combination of rock properties and pressures. 'The average technical limit specific energy for a hole interval or well provides a rational basis for comparing drilling performance for wells drilled in different drilling environments.'[5]

Dupriest and Koederitz[6] adopted Teal's specific energy equation in present drilling units and arrived at a model for Mechanical Specific Energy that was used in a drilling information system for mud drilling and has been implemented successfully on different rigs. Dupriest and Koederitz showed the usefulness of MSE through practical field application. They also showed that bit hydraulics, though not incorporated in the MSE equation, had a noticeable effect on MSE and ROP.

The models in the above literature have applied the MSE concept to water or oil based muds, air as a drilling fluid has not been mentioned yet.

Air was substituted for liquid as a drilling fluid on an experimental basis on several wells in the early 1950's in order to further improve drilling rate. A further improvement in the drilling rate was achieved by the introduction of the air-operated, rotary percussion drilling tool.

Howard *et al.*[7] successfully developed an air-operated percussion tool for down-the-hole air drilling operations. In 1959, when their paper was first written, down-the-hole air-operated, rotary-percussion quarry drilling tools drilled with chisel bits, but they had little success when used with conventional roller cone bits because the percussive impact was beyond the strength of the bits. They gave a detailed description of the new hammer tool, which was basically a modification of the already existing air-operated quarry drilling tool. The modifications were in the tools valves spacing, hammer weight and stroke length. These were adjusted to decrease hammer impact but at a higher cyclic frequency which permitted the tool to be used with roller cone bits. Bit footages observed in laboratory shallow well tests showed an overall advantage of rotary percussion over air drilling of 1.3 times, whereas field tests showed an advantage of about 4 times.

Another pneumatic down-hole percussion tool that had just been developed then in 1965 was described by Bates[7]. Conditions affecting the tool's operation were also described. One obvious factor affecting the tools operation was the pressure across the tool as the percussive energy was directly proportional to the differential pressure across the tool. Bates also used a simple linear equation to calculate the kinetic energy of the percussive tool. $Kinetic\ Energy\ per\ blow,\ ft - lb = p \times A \times S$. Where p is the pressure drop across the hammer in pounds per square inch (*psi*), A is the piston area in square inches and S is the length of the piston stroke in feet (ft).

Whiteley and England[8] discussed the engineering design and operation of the flat-bottom percussion bit/hammer tool (FPB/HT) in air drilling operation. The hammer tool they discussed is valveless and the piston functions as a sliding valve to control the operating air cycle. They gave a detailed description of its operation and mentioned that the impact energy from the hammer tool is largely responsible for penetration, the weight on bit (WOB) applied during drilling with the FPB/HT is for optimum operation of the hammer tool. Excess WOB will prevent efficient operation of the hammer tool. Applications and limitations of the tool were stated as well as guidelines for optimization of the tool's performance.

As the years went by more air-operated percussive tools were developed and improved upon to give greater drilling rate and researchers constantly showed the advantages of the air hammer tool over conventional mud drilling.

Finger[9] did a study of mining and oilfield hammers and showed that industrial hammers could drill more than twice as fast as extremely high WOB rotary air and that the ROP with industrial hammers was three to six times as fast as with oilfield hammers. Finger identified sensitivity to WOB and gauge wear as two potential problems with the application of the industrial/mining hammer in the oilfield.

Pratt[10] elucidated on the above potential problem with ROP in his paper where he talked about the modifications made to equipment and computer hydraulics programs that enabled Shell Canada to drill more efficiently. Pratt indicated that one of the major factors affecting ROP was bit gauge wear. When a bit goes slightly under gauge, the driver sub causes noticeable torque and hangs up sufficiently to reduce ROP. The problem was however solved by redesigning the bit. Pratt compared ROP's in wells drilled by Shell in Canada with water/mud, air rotary and air hammer between 1979 and 1983 and showed that ROP increased with air over mud drilling with a further increase in ROP when the air hammer was used.

Pang *et al*[11] developed a complete model of a pneumatic jack hammer system. The application of their model required that two preliminary experiments be performed. The first experiment produced an empirical relationship between piston impact velocity and the pressures acting on the top and bottom surfaces of the piston. After obtaining the value for the impact velocity of the hammer, the impact energy would then be calculated. While from the second experiment they were able to determine the forceindentation behavior of the bit/target system. The overall model analysis led to the prediction of the jack hammer efficiency and target response, including target penetration and crack propagation.

Han *et al*[12], in an effort to improve understanding of drilling physics and the prediction of hammer performance, developed a 3D numerical simulator for air hammer drilling. One of the outputs of the simulator is a bit advancement plot (ROP plot). This helps in the estimation of ROP for different hammer energy and formation properties. Han *et al* had already, in 2005, developed an integrated simulation tool, which included a tool model, a rock mechanics model and a cuttings transport model. Han *et al*[12] in their present effort addressed the rock mechanics involved after compressive stresses pass from bit to rock. The 3D simulator requires rock property input and hammer tool specific inputs as stated in their work.

Various air hammers have been designed, developed and put into operation. Designers use different methods and equations to obtain their performance indicators such as impact energy, power, impact frequency and efficiency to mention a few. The industry requires a simple general equation to calculate performance indicators. Chiang and Stamm[13] proposed a design methodology for down-the-hole pneumatic hammers by developing a generic non-linear dynamic model used to compute hammer performance. The dynamic model consists of a set of differential equations and non-linear polynomial equations the solution of which would give hammer performance indicators such as, impact energy, power, efficiency and mass flow rate.

Chiang and Izquierdo[14] adopted the down-the-hole pneumatic hammer dynamic model developed by Chiang and Stamm[13] in their research effort that resulted in the development of a methodology to estimate the instantaneous specific rock energy using corrected down-the-hole (DTH) drill monitoring data. Consequently, they were able to generate a specific rock energy profile for every hole drilled and thus mapping an entire drilling site for this index. They stated the development procedure for a special data acquisition system used to measure and register operational variables that are inputs for two simulation models that estimate the power absorbed by the rock through impact and then the specific rock energy index. Correlations were found between the specific rock energy and impact frequency, as well as between the penetration rate and applied torque and between the penetration rate and impact frequency.

The majority of the literatures mentioned in this chapter have shown the application of the concept of Mechanical Specific Energy to bit performance and selection as well as to drilling performance evaluation in water and oil based mud systems, but little has been done in arriving at a direct model for MSE in air hammer drilling systems and this obviously requires more research work.

This research effort has been geared towards providing a model for calculating MSE for air hammer drilling systems, an important tool in evaluating drilling performance.

CHAPTER III

AIR HAMMERS

3.1 Description of Air Hammer Operation

The use of air as an energy carrier and drilling fluid in down-hole air hammer drilling has been known for many years. Also well known is the fact that down-hole air hammer drilling is by far the fastest method of penetration in hard rock material.

Generally, during drilling with down-hole air hammers, the tool is placed in front of the borehole right behind the bit, while energy is transferred through the drill string in the form of compressed air, mechanical torque and mechanical axial force. The main function of the air hammer is to convert energy from the compressed air into piston kinetic energy which, through the oscillating movement of the piston and eventual mechanical impact with the bit, is transferred to the bit and then to the rock. Rock fragmentation occurs at highly pressurized contact zones between the bit buttons and the rock. Bit rotation, which is required during down-hole air hammer drilling, will create new impact positions for the bit buttons and new rock will be fragmented and consequently, advancing the bit. Crushed rock is flushed to the outside of the drill string from under the bit through the borehole annulus to the surface by compressed air flowing through the bit nozzles. **Fig. 3.1** shows air hammers with flat bottom hammer bit.

3.2 Air Hammer Types and Description of Hammer Action

There are a variety of down-hole air hammer manufacturers with different proprietary air hammer designs, however there are two basic designs for the down-hole air hammers based on the flow path of the compressed air through the hammer. One design utilizes a flow path of the compressed air through a control rod (or feed tube) down the center of the hammer piston (or through passages in the piston) and then through the hammer bit, while the other design utilizes a flow path through a housing annulus passage (around the piston) and then through the hammer bit.

Fig. 3.1 Air Hammers with Flat Bottom Bit[20]

Fig. 3.2 shows a diagram of a typical control rod flow design down-hole air hammer. The hammer action of the piston on the top of the drill bit shank provides an impact force that is transmitted down the shank to the bit studs which in turn crush the rock at the rock face.

In shallow boreholes where there is little annulus back pressure, the piston impacts the top of the bit shank at a rate of from about 600 to 1,700 blows per minute (depending on volumetric flow rate of air). However, in deep boreholes where the annulus back pressure is usually high, impact rates can be as low as 100 to 300 blows per minute[15].

When the air hammer is suspended from a drill string lifted off the bottom (the shoulder of the bit is not in contact with the shoulder of the driver sub) as shown in Fig. 3.2 and in the first schematic in **Fig. 3.3**, the compressed air flows through the pin connection at the top of the hammer to the bit without actuating the piston action. When the hammer is placed on the bottom of the borehole and weight placed on the hammer, the bit shank will be pushed up inside the hammer housing until the bit shoulder is in contact with the shoulder of the driver sub. This action aligns one of the piston ports with one of the control rod windows. This allows the compressed air to flow to the space below the piston which in turn forces the piston upward in the hammer housing. During this upward stroke of the piston, no air passes through the bit shank to the rock face. In essence, rock cuttings transport is momentarily suspended during this upward stroke of the piston. For example, at a piston impact rate of 600 blows per minute the air through the bit is suspended for about 0.05 seconds per cycle. This is so short a time that the air flow rate through the bit into the annulus can be assumed as a continuous flow[15].

When the piston reaches the top of its stroke, another one of the piston ports aligns with one of the control rod windows and supplies compressed air to the open space above the top of the piston. This air flow forces the piston downward until it impacts the top of the bit shank. At the same time the air flows to the space above the piston, the foot valve at the bottom of the control rod opens and air inside the drill string is exhausted through the control rod, bit shank and the bit nozzles to the rock face. This compressed air exhaust carries the rock cuttings created by the drill bit for transport up the annulus to the surface. The impact force on the bit allows the rotary action of the drill bit to be very effective in destroying rock at the rock face and this in turn allows the air hammer to drill with low WOB.

Down-hole air hammers are lubricated by occasionally injecting oil type lubricants into the compressed air and down the drill string during drilling operations. The lubricant serves to lubricate the piston surfaces as it moves up and down in the hammer housing.

Down-hole air hammers are used exclusively for vertical drilling operations as the short, rapid blows of the piston minimizes the effect of dipping and formation damage. There is however, on going research efforts to develop down-hole hammers for directional drilling.

Down-hole air hammers are available in various sizes with associated suitability for drilling different borehole sizes. The available down-hole air hammer sizes are 3 inches to 16 inches. The 3 inch diameter hammer can drill a borehole as small as 3 5/8 inches and the 16 inch diameter hammer can drill boreholes from 17 ½ inches to 33 inches.

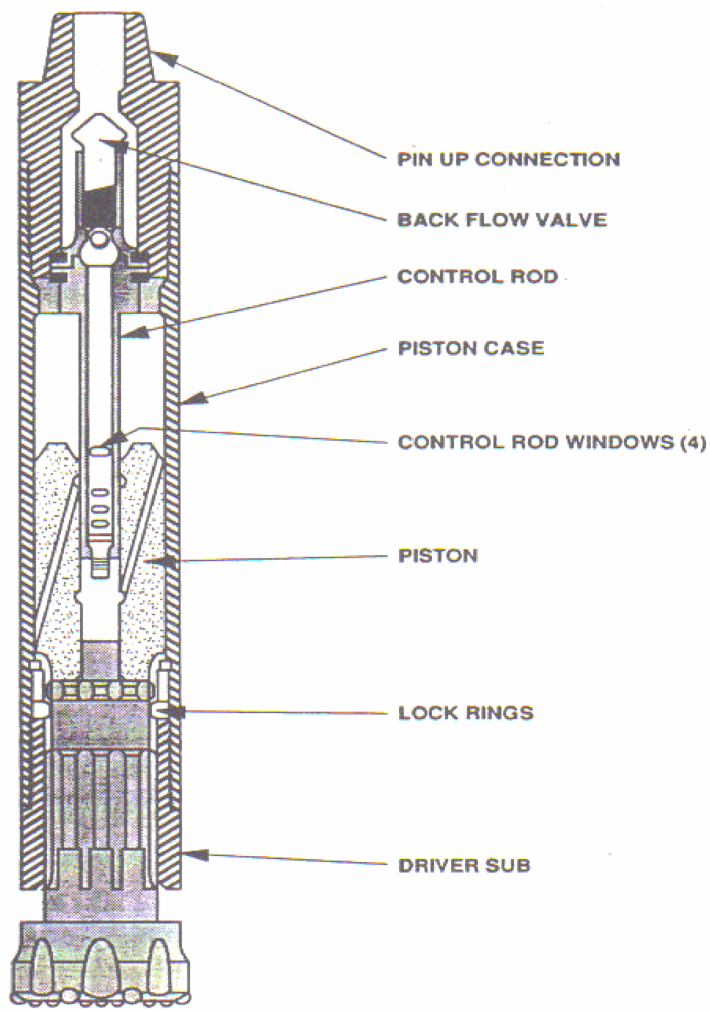

Fig. 3.2 Schematic Cutaway of a Typical Air Hammer[15]

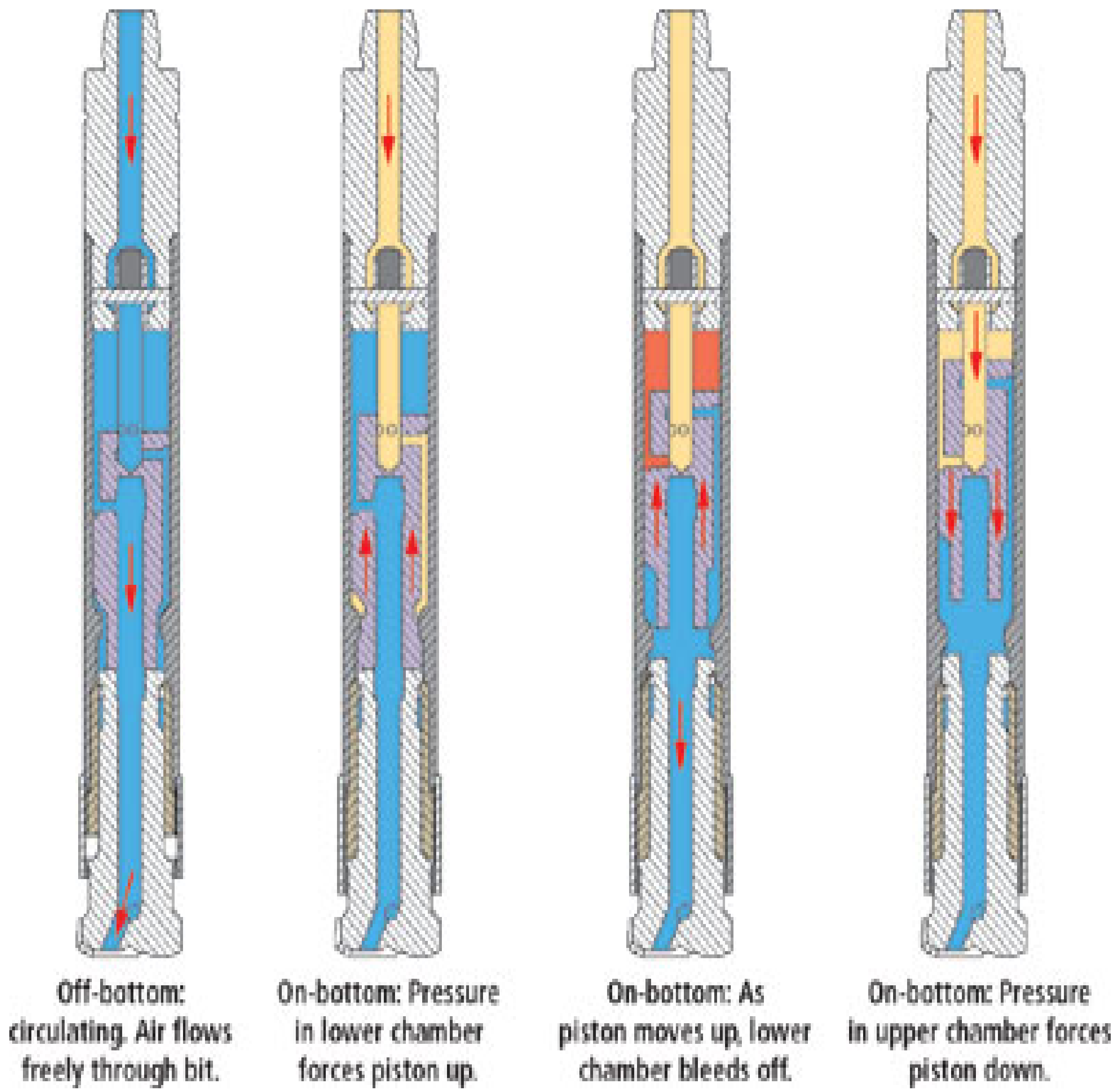

Fig. 3.3 Air Hammer Sequence Schematic[19]

CHAPTER IV

MECHANICAL SPECIFIC ENERGY MODEL DEVELOPMENT

4.1 Derivation of the Specific Energy Model

'Specific Energy' is the work done per unit volume drilled. Mechanical Specific Energy (MSE) is a ratio that quantifies the relationship between input energy and ROP. This ratio should be constant for a given rock, which is to say that a given volume of rock requires a given amount of energy to destroy the rock. R. Teale[2], in his paper "The Concept of Specific Energy in Rock Drilling" published in 1965 in the International Journal of Rock Mechanics and Mining Science, derived the Specific energy equation by calculating the torsional and axial work performed by the bit and dividing this by the volume of rock drilled. Teale then conducted laboratory tests that demonstrated the energy per volume of rock destroyed to be relatively constant, regardless of changes in ROP, WOB or RPM.

The equation is derived below:

The work done in one minute by the bit, during rock destruction and exaction, is the sum of the axial work done by the WOB and the torsional work done due to rotation of the bit. If ROP is in inch per minute and torque in inch pounds (in-lbs), then the axial work done in one minute to excavate one inch of rock is $WOB \times ROP$ and the torsional work done is $2\pi NT$, where N is the bit rotation in rev/min and T is torque. Therefore the total work done $= WOB \times ROP + 2\pi NT$ …………………………………………….. (4.1)

Now the Specific Energy is work done per unit volume. Let the circular area of the bit involved in the drilling be "A", then the volume of rock excavated in one minute is $A \times ROP\left(in^3 / \min\right)$

Specific Energy $= \dfrac{WOB \times ROP}{A \times ROP} + \dfrac{2\pi NT}{A \times ROP} = \dfrac{WOB}{A} + \dfrac{2\pi NT}{A \times ROP}$ …………………... (4.2)

Teale noticed that laboratory drilling data showed the MSE value to be numerically equal to rock compressive strength in pound per square inch (psi). This is useful from an operations standpoint because it provides a reference point for efficiency. If the observed MSE is close to the known confined rock strength, the bit is efficient. If not, energy is being lost. The value should change as the lithology changes. However, field experience has shown that the energy losses that occur when the bit founders (The point at which the ROP stops responding linearly with increasing WOB is referred to as the "flounder" or "founder" point) are usually so large that they cannot be confused with the small changes that occur with rock compressive strength.

In the drilling environment ROP is expressed in ft/hr and torque in ft-lb. F. E Dupriest and W. L Koederitz in their paper SPE/IADC 92194 adopted Teal's equation in present drilling units and arrived at

$$MSE = \frac{WOB}{A} + \frac{120\pi NT}{A \times ROP} = \frac{4 \times WOB}{Dia^2 \times \pi} + \frac{480 \times T \times N}{Dia^2 \times ROP} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(4.3)}$$

Where, $A = \frac{\pi Dia^2}{4}$, ROP in ft/hr and Torque in ft-lb.

Dupriest and Koederitz[6] showed the usefulness of MSE through practical field application, they also showed that Bit Hydraulics, though not incorporated in the MSE equation, had a noticeable effect on MSE and ROP.

4.2 Development of Mechanical Specific Energy Model

The pneumatic-hammer utilizes an internal piston (or hammer) that is actuated by the compressed air (or other gas) flow inside the drill string. The internal piston moves up and down in a chamber under the action of air pressure applied either below or above the piston through points in the inside of the air hammer. In the downward stroke, the hammer strikes the bottom of the upper end of the drill bit shaft (via a coupling shaft) and imparts an impact load to the drill bit. The drill bit in turn transfers this impact load to the rock face of the bit. This impact load creates a crushing action on the rock face.

The rock destruction and excavation process is basically the same for air drilling as it is for mud drilling. However in the case of down-the-hole air hammer drilling the energy for rock destruction is supplied by three sources: impact (from air hammer), axial downward drill string force (WOB) and rotation (torque). Hydraulics though important has been shown (below) to have little contribution to the energy required for rock destruction. However, hydraulic energy is considered for efficient hole cleaning and because of its effect on MSE and ROP.

Using the earlier procedure for the derivation of the equation for Specific Energy which is the work done per unit volume, or which can also be defined as the energy required to destroy and excavate a unit volume of rock, we can express MSE as

$$MSE = \frac{E_{hammer}}{V_{unit-vol}} + \frac{E_{axial-thrust}}{V_{unit-Vol}} + \frac{E_{Torque}}{V_{unit-Vol}} \qquad (4.4)$$

Where

E_{hammer} = Hammer impact Energy

$E_{axial-thrust}$ = Axial Energy by virtue of applying WOB

E_{Torque} = Rotational Energy

Let the circular area of the bit involved in the drilling be "A" in inches and ROP in inch per minute, torque in inch-lb and hammer impact energy in inch-lb, then the volume of rock excavated in one minute is $A \times ROP\left(in^3 / \min\right)$

$$MSE = \frac{E_{hammer}}{A \times ROP} + \frac{WOB \times ROP}{A \times ROP} + \frac{2\pi NT}{A \times ROP}$$

$$= \frac{E_{hammer}}{A \times ROP} + \frac{WOB}{A} + \frac{2\pi NT}{A \times ROP} \quad \text{.. (4.5)}$$

Eq. 4.5 can be expressed in oilfield measurement units

$$MSE = \frac{60 \cdot 12 \cdot E_{hammer}}{12 \times A \times ROP} + \frac{WOB}{A} + \frac{120\pi NT}{A \times ROP} \quad \text{.. (4.6)}$$

$$MSE = \frac{240 \cdot E_{hammer}}{Dia^2 \times \pi \times ROP} + \frac{4 \times WOB}{Dia^2 \times \pi} + \frac{480 \times T \times N}{Dia^2 \times ROP} \quad \text{.................................. (4.7)}$$

Where $A = \frac{\pi Dia^2}{4}$, ROP in ft/hr, torque in ft-lb and E_{hammer} in ft-lb. This is consistent with oilfield measurement units.

It has been experimentally shown by Chiang and Izquierdo[14] that about 60% of the energy required for rock destruction in Down-The-Hole (DTH) percussive drilling comes from the pneumatic hammer. Therefore adequate knowledge of the DTH pneumatic hammer is required. The hammer piston delivers certain impact energy every time it strikes the bit shank and looking at the hammer as a pneumatic engine, then the total impact energy delivered in one minute can be approximated as the power delivered by this engine in one minute.

$$E_{hammer} \cong Pow_{hammer} = Pow_{raw} \cdot \eta_{transmission} \quad \text{.. (4.8)}$$

From the above the Pow_{raw} is the raw power delivered by the pneumatic hammer engine in one minute which is the piston impact energy times the impact frequency. The magnitude of Pow_{raw} depends on the thermodynamic behavior of the hammer, and that is how efficiently the high pressure air energy is converted into kinetic energy of the piston at impact. Kinetic energy from the piston is transferred upon impact on the bit shank by way of stress wave that propagates toward the bit and then to the rock. Depending on the geometry of the hammer components and the mechanical properties of the rock, a percentage of the stress wave energy is absorbed by the rock, causing its failure, and the remaining is reflected back and dissipated by the hammer and supporting structures. The magnitude of the energy transmission efficiency, $\eta_{transmission}$, takes the inefficiency in the transfer of the piston kinetic energy into account.

The contribution of the impact power of the hammer on the rock destruction can be modeled by a thermodynamic model. The impact power developed by a DTH hammer originates from air supplied at high pressure. The balance of pressures in the front and rear chambers in a DTH hammer causes alternate up and down motion of the piston. At the end of the forward stroke, the moving piston impacts a drill bit, initiating a stress wave that travels towards the rock.

To simulate the thermodynamic operation of the DTH hammer, a model developed by Chiang and Stamm[13] in their paper titled "*Design Optimization of Valveless DTH Pneumatic Hammers by a Weighted Pseudo-Gradient Search Method"* (1998) Journal of Mechanical Design, will be used. The model relates air pressure, piston velocity and impact frequency for a given DTH hammer geometry. The impact restitution coefficient *e* between the piston and drill bit takes rock behavior into account.
Chiang *et al.* defined the coefficient of restitution *e* as

$$e = \frac{piston\,'rebound'\,velocity}{piston\,impact\ factor} = \frac{v_{pB}}{v_{pA}} \quad \text{.....} \quad (4.9)$$

Where v_{pB} is the piston reflected velocity and v_{pA} is the piston impact velocity. Chiang *et al*'s thermodynamic model has been experimentally validated for a number of DTH hammers in an experimental bench test. More so, laboratory and field measurements reveal that the predominant variable in the operation of any DTH hammer is the input air pressure.

From the above explanations, considering the fact that the raw power (maximum available power) of a DTH hammer is obtained as the product between the piston kinetic energy difference (Impact and the rebound, represented by e^2) and the percussion frequency, it is possible to compute the raw power as:

$$Pow_{raw} = \frac{1}{2} m_{piston} V_{\mathrm{Im}pact}^2 \left(1 - e^2\right) F \quad \text{……………………………………………………} \quad (4.10)$$

Where m_{piston} is the mass of the piston, V_{impact} is the impact velocity of the piston and F is the impact frequency (blows per minute). Substituting Eq. 4.10 into Eq. 4.8

$$E_{hammer} \cong Pow_{hammer} = Pow_{raw} \cdot \eta_{transmission} = \frac{1}{2} m_{piston} \cdot V_{\mathrm{Im}pact}^2 \left(1 - e^2\right) \cdot F \cdot \eta_{transmission} \quad \ldots \quad (4.11)$$

Substituting Eq. 4.11 into Eq. 4.7, the MSE model becomes:

$$MSE = \frac{120 \cdot m_{piston} \cdot V_{\mathrm{Im}pact}^2 \left(1 - e^2\right) \cdot F \cdot \eta_{transmission}}{Dia^2 \times \pi \times ROP} + \frac{4 \times WOB}{Dia^2 \times \pi} + \frac{480 \times T \times N}{Dia^2 \times ROP} \quad \text{………} \quad (4.12)$$

For the calculation of the impact velocity, we will be considering the downward movement of the piston and the velocity just before impact should be the maximum velocity of the piston. At this point the position of the piston will be at the maximum downward stroke of the piston which means the rear chamber behind this piston would be occupied by high pressure air and the air in the front chamber would have been exhausted. During actual operation of the DTH pneumatic hammer, at the same instant the air flows to the space behind the piston (Rear Chamber), the foot valve at bottom opens and air is exhausted from the front chamber.

Fig. 3.2 in chapter III shows the air hammer suspended from a drill string lifted off-bottom. In this position, compressed air flows through the pin connection at the top of the hammer to the bit without the piston action. When the hammer is placed on the bottom of the hole, and weight is placed on the hammer, the bit shank will be pushed up inside the hammer housing until the bit shoulder is in contact with shoulder of the driver sub. This action aligns one of the piston ports (of one of the flow passages through the piston) with one of the control rod windows. This allows the compressed air to flow to the space below the piston which in turn forces the piston upward in the hammer housing. During this upward stroke of the piston, no air passes through the bit shank to the rock face. In essence, rock cuttings transport is suspended during this upward stroke of the piston[17].

When the piston reaches the top of its stroke, another one of the piston ports aligns with one of the control rod windows and supplies compressed air to the open space above the piston. This air flow forces the piston downward until it impacts the top of the bit shank. This impact force on the bit allows the rotary action of the drill bit to be very effective in destroying rock at the rock face. This in turn allows the air hammer to drill with low WOB.

4.3 Cylinder Model

The hammer housing where the conversion of air flow and pressure into the kinetic energy of the piston can be considered as a cylinder as shown in **Fig. 4.1.** Fig. 4.1 shows a simplified diagram of the downward stroke of the piston of the pneumatic Hammer used to derive the thermodynamic model for the Hammer. **Fig. 4.2** is another simplified diagram of the air hammer highlighting the piston, bit and formation, with the piston moving downward to impact the bit.

The flow of air through the pneumatic cylinder is considered to be isothermal and adiabatic, while the flow through the cylinder's inlet and outlet valves is considered isentropic in this research, therefore a definition of these flow processes is worth mentioning at this point.

An isothermal process is one in which there is no change in temperature, while an adiabatic process is one in which no heat is added to or taken away from the flow system. An isentropic process is a frictionless adiabatic process.

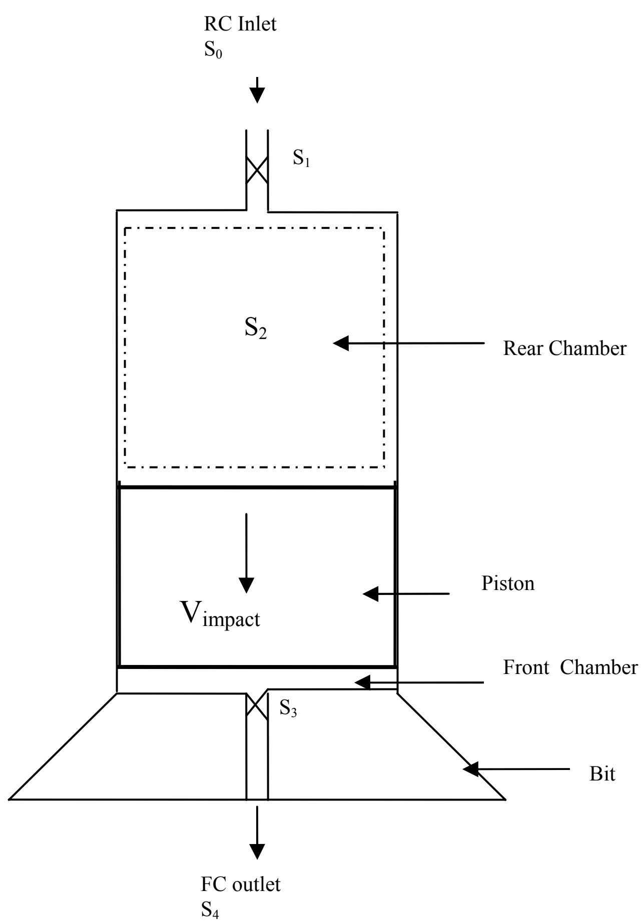

Fig. 4.1 Simplified Cylinder Diagram

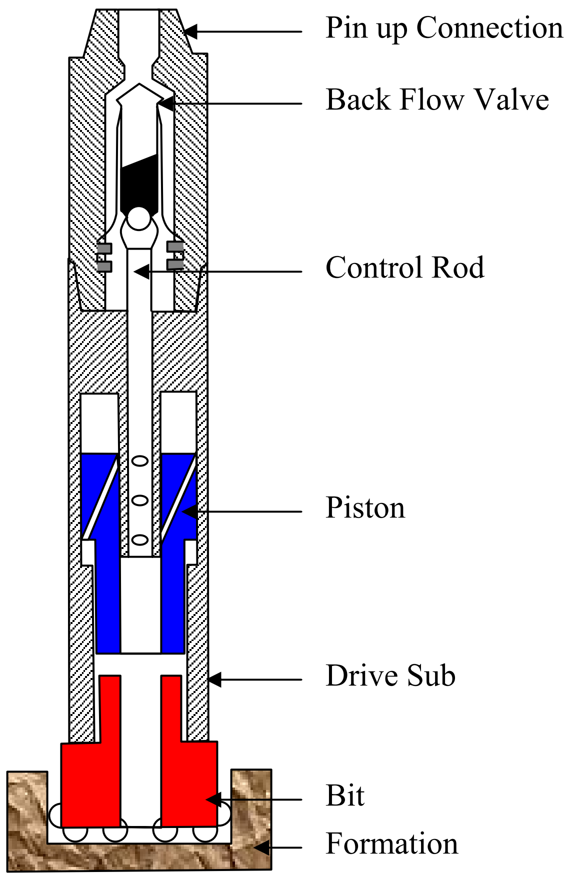

Fig. 4.2 Air Hammer Cross Section Highlighting the Piston, Bit and Formation

Below are conditions associated with the above flow processes.

For an isothermal flow – pv = constant

For adiabatic flow - pv^k = constant, k is defined in the equation below

$$k = {c_p}/{c_v} \quad \ldots\ldots (4.13)$$

Where, k is the ratio of the specific heats. The terms c_p and c_v are the specific heat at constant pressure and constant volume, respectively.

There are four main variables to calculate in the simulation of pneumatic cylinders: air mass W_g, piston velocity v_{pA}, air temperature (which will be bottom hole temperature), and pressure P at the cylinder inlet and outlet, which is effectively the pressure drop across the cylinder (or hammer in this case). The four equations necessary to know these variables are respectively: the equation of continuity, the equation of motion, the energy equation and the state equation of ideal gases. It is assumed that: (i) the gas (which is air in this case) is perfect, (ii) the pressures and temperature within each cylinder chamber is homogeneous and, (iii) kinetic and potential energy terms are negligible.

The Ideal gas equation of state is:

$$\frac{pv}{T} = mR \quad \text{(4.14)}$$

Where p is the absolute pressure of the gas, v is the volume of the gas, T is the absolute temperature of the gas, and m is mass of the gas and R is the ideal gas constant.

From the equation of continuity, the air mass changes at the rear chamber and front chamber are, respectively,

$$\frac{dW_{iRC}}{dt} = \dot{w}_{iRC} \quad \text{(4.15)}$$

$$\frac{dW_{iFC}}{dt} = -\dot{w}_{iFC} \quad \text{(4.16)}$$

Using Newton's second law of motion, the equation of motion for the piston, assuming a negligible effect of friction, is

$$Force, F_p = m_p \frac{dv_p}{dt} = A_{RP}P_{RC} - A_{FP}P_{FC} + m_p g \quad \text{(4.17)}$$

If the piston has a uniform rear and front chamber area, then we have

$$Force, F_p = m_p \frac{dv_p}{dt} = A_p (P_{RC} - P_{FC}) + m_p g = \Delta P A_p + m_p g \quad \text{(4.18)}$$

This is the resultant force on the piston. This force upon traveling the length of piston stroke, *S,* will have an approximate energy equivalent to the product of the force times the piston stroke. However, a more accurate model for the energy of the piston will be to use the energy equation from the first law of thermodynamics. The first law of thermodynamics is most easily stated for a material volume as: *the rate of change of stored energy equals the sum of rate of work done and rate of heat addition to a material volume.*

For the pneumatic cylinder in Fig. 4.1 and from the first law of thermodynamics and assuming no heat loss occurs in the cylinder, since we have considered the process to be adiabatic (i.e., the heat loss is small relative to the work and enthalpy terms), the energy equation in it basic form can be written as:

$$\dot{U} = \dot{H} - \dot{W} \quad \text{(4.19)}$$

Where U is the internal of the air in the cylinder chambers, H is the enthalpy added to the cylinder chambers and W is the work done by the cylinder chambers.

Using the simplified cylinder diagram in Fig. 4.1 and the considering the cylinder piston to be in the downward motion right before impact with the bit, the various terms in Eq. 4.19 can be expressed as:

$$\dot{U} = \frac{dU}{dt} = c_p \dot{w}_{2RC} T_{2RC} + c_p w_{2RC} \dot{T}_{2RC} \quad \text{(4.20)}$$

$$\dot{H} = \frac{dH}{dt} = c_p \dot{w}_{1RC} T_{1RC} - c_p \dot{w}_{3FC} T_{3FC} \quad \text{(4.21)}$$

$$\dot{W} = \frac{dW}{dt} = P_{2RC}\dot{V}_{2RC} = P_{2RC}A_{RP}v_p \quad \text{(4.22)}$$

It is assumed in the derivation of Eq. 4.21 that the enthalpy term associated with the time rate of change of temperature is small relative to that associated with the mass flow rate. Substituting Eqs. 4.20, 4.21 and 4.22 into Eq. 4.19 and simplifying, yields,

$$\frac{dT_{2RC}}{dt} = \dot{T}_{2RC} = \frac{1}{w_{2RC}}(-\dot{w}_{2RC}T_{2RC} + \dot{w}_{1RC}T_{1RC} - \dot{w}_{3FC}T_{3FC}) - \frac{P_{2RC}A_{RP}v_p}{w_{2RC}c_p} \quad \text{(4.23)}$$

Also from the pneumatic cylinder design, when air flows into the rear chamber of the cylinder, air also exits the front chamber of the cylinder as the piston moves,

$$\dot{w}_{2RC} = \dot{w}_{1RC} - \dot{w}_{3FC} \quad \text{(4.24)}$$

4.3.1 Rear Chamber Equations

$$V_{2RC} = V_{mrear} + A_{RP}x_p \quad \text{(4.25)}$$

$$v_{2RC} = \frac{V_{2RC}}{w_{2RC}} \quad \text{(4.26)}$$

$$P_{2RC} = \frac{RT_{2RC}}{v_{2RC}} \quad \text{(4.27)}$$

$$C_{RC} = P_{2RC}v_{2RC}^n \quad \text{(4.28)}$$

The critical pressure at the inlet section is given by

$$P_{1RC}^* = P_{0RC}\left(\frac{2}{n+1}\right)^{n/(n-1)} \quad \text{(4.29)}$$

The actual pressure at the inlet throat P_{d1} is now computed. If, $P_{2RC} < P^*_{1RC}$ then, $P_{d1} = P^*_{1RC}$ otherwise $P_{d1} = P_{2RC}$. Thus the mass flow rate,

$$\dot{w}_{1RC} = C_{m1RC} A_{d1RC} \times \left[\frac{2c_p P_{0RC}}{R v_{0RC}} \left(\left(\frac{P_{d1}}{P_{0RC}} \right)^{2/n} - \left(\frac{P_{d1}}{P_{0RC}} \right)^{(1-n)/n} \right) \right]^{1/2} \quad \text{……………………} (4.30)$$

The critical pressure at the outlet section

$$P^*_{3FC} = P_{2FC} \left(\frac{2}{n+1} \right)^{n/(n-1)} \quad \text{…………………………………………………….…} (4.31)$$

The actual pressure at the outlet throat P_{d3} is now calculated. If, $P_{4FC} < P^*_{3FC}$ then $P_{d3} = P^*_{3FC}$, otherwise $P_{d3} = P_{4FC}$. Thus the mass flow rate,

$$\dot{w}_{3FC} = C_{m3RC} A_{d3FC} \times \left[\frac{2c_p P_{2FC}}{R v_{2FC}} \left(\left(\frac{P_{d3}}{P_{2FC}} \right)^{2/n} - \left(\frac{P_{d3}}{P_{2FC}} \right)^{(1-n)/n} \right) \right]^{1/2} \quad \text{…………………...…} (4.32)$$

4.3.2 Front Chamber Equations

$$V_{2FC} = V_{mfront} + A_{FP} x_p \quad \text{……………………………………………………….………} (4.33)$$

$$v_{2FC} = \frac{V_{2FC}}{w_{2FC}} \quad \text{………………………….………………………………...…………} (4.34)$$

$$P_{2FC} = \frac{RT_{2FC}}{v_{2FC}} \quad \text{………………………………………………………..………} (4.35)$$

$$C_{FC} = P_{2FC} v^n_{2FC} \quad \text{…………………………………………………..…...…} (4.36)$$

4.3.3 Energy Transmission Efficiency

A DTH pneumatic hammer converts pneumatic energy into piston kinetic energy. This kinetic energy in then transferred to the drill bit through an impact at every blow cycle. A stress wave is thus generated at the impact section that propagates toward the drill bit and piston end. The portion of the stress wave that travels toward the bit-end eventually reaches into the rock. Most of the incident stress wave gets absorbed by the rock resulting in its breakage, while some of it gets reflected back to the drill bit, piston and hammer cylinder. The piston receives the resulting reflected wave and acquires a return velocity. Simulations carried out by Chiang and Stamm[13], show that the piston and the bit can stay in contact and separate alternatively several times before interaction is completely over. On the other hand the drill string receives a portion of the reflected waves by virtue of the hammer cylinder which is holding the bit in contact with the bottom of the hole while exerting a thrust force over the drill bit.

Energy transmission efficiency is a very important issue in pneumatic hammer design. From a performance stand point, it affects the rate of penetration no matter how good the conversion of pressure energy to piston kinetic energy is, inside the hammer. The hammer cycle is also affected because the return velocity is determined by the amount of reflected energy after impact, which in turn has an effect over both the piston stroke length and cycle duration. Estimation of the energy transferred to the bit and then the rock during DTH percussive drilling is a very complex process. However, Chiang and Stamm[13] established a simplification based on earlier works on the subject by Lundberg (1982). Hence a detailed analysis of the stress wave propagation is carried out for the piston-bit interaction and an empirical factor is used for the actual energy absorbed by the rock. This empirical factor depends on the rock tenacity and drill bit shape, and must be obtained by field tests.

From the above explanation, one can analytically deduce relations, if A and B are used to denote immediately before and after impact.

From linear conservation of momentum

$$m_p v_{pA} = m_p v_{pB} + m_b v_{bB} \quad \text{……………………………..………………} \quad (4.37)$$

From the energy conservation

$$\frac{1}{2} m_p v_{pA}^2 = \frac{1}{2} m_p v_{pB}^2 + \frac{1}{2} m_b v_{bB}^2 \quad \text{……………………………..……………} \quad (4.38)$$

Thus

$$v_{pB} = -\frac{(m_b - m_p)}{(m_b + m_p)} v_{pA} \quad \text{…………………………………………..……} \quad (4.39)$$

And

$$v_{bB} = \frac{(2m_p)}{(m_b + m_p)} v_{pA} \quad \text{……………………………………………………..} \quad (4.40)$$

The energy absorbed by the rock is the kinetic energy transmitted to the bit upon impact by the piston

$$E_R = \frac{1}{2} m_b v_{bB}^2 \quad \text{…………………………………………...…..…..} \quad (4.41)$$

The energy reflected through the piston is

$$E_P = \frac{1}{2} m_b v_{pB}^2 \quad \text{…………………………………………...……} \quad (4.42)$$

Note that the coefficient of restitution e is defined as $e = v_{pB} / v_{pA}$. The energy transmission efficiency was derived by Li *et al*[16] and will be briefly described here,

Where $\alpha = 2(m_b / m_p)\beta$ and $\beta = K\tau / (\rho c A_p)$

When α > 1, $\beta \geq 0.5\pi\alpha / (\alpha - 1)^{0.5}$

$$\eta = 2 \cdot \frac{\left(1 + e^{-\pi / \sqrt{\alpha - 1}}\right)^2}{\beta} \quad \text{……………………………………..……….. (4.43)}$$

When α > 1, $\beta < 0.5\pi\alpha / (\alpha - 1)^{0.5}$

$$\eta = 2 \cdot \frac{1 + e^{-4(\beta/\alpha)} + \frac{2}{(\alpha-1)} \cdot \left(\sin\left(2 \cdot \sqrt{\alpha-1} \cdot \frac{\beta}{\alpha}\right) + \sqrt{\alpha-1} \cdot \cos\left(2 \cdot \sqrt{\alpha-1} \cdot \frac{\beta}{\alpha}\right)\right) \cdot \left(e^{-4(\beta/\alpha)} \cdot \sin\left(2 \cdot \sqrt{\alpha-1} \cdot \frac{\beta}{\alpha}\right) - \sqrt{\alpha-1} \cdot e^{-2 \cdot (\beta/\alpha)}\right)}{\beta}$$

……………………………………………………………………….……………………... (4.44)

Since $\alpha = 2(m_b / m_p)\beta$, from the equation $\beta = 0.5\pi\alpha / (\alpha - 1)^{0.5}$.

The estimated *K* values of very hard rocks (σ_c = 200 MPa). Medium hard rocks (σ_c = 80 MPa), and soft rocks (σ_c = 40 MPa) are 250, 90 and 50 MN/m respectively. Other models, one of which is that by Lundberg and Okrouhlik[17], exist that could be used to compute the energy transmission efficiency.

4.4 Impact Energy of Air Flowing through the Bit

On the first pass, the jet impact force was modeled as follows: To model the Hydraulic impact force at the bit, it was assumed that all the fluid momentum is transferred to the hole bottom. Since the fluid is traveling at a vertical velocity v_n before striking the hole bottom and is traveling at a zero vertical velocity after striking the hole bottom, the time rate of change of momentum (in field units) is given by[18]:

$$F_j = \frac{\Delta(m\bar{v})}{\Delta t} \cong \left(\frac{m}{\Delta t}\right)\Delta\bar{v} = \frac{(\rho q)\overline{v_n}}{32.17(60)} \quad \text{…………………………………………...……… (4.45)}$$

Where (ρq) is the mass rate of the fluid through the bit, and ρ is in lb/ft^3, q is in ft^3/min, v_n is in ft/sec and g = 32.17 ft/sec^2.

From the book, *Gas Volume Requirements for Underbalanced Drilling* by Boyun Guo and Ali Ghalambor, the flow of air through the bit, q_b is given by

$$q_b = 6.02CA_tP_{ai}\sqrt{\frac{k}{(1-k)S_gT_{bh}}\left[\left(\frac{P_{bh}}{P_{ai}}\right)^{\frac{2}{k}} - \left(\frac{P_{bh}}{P_{ai}}\right)^{\frac{k+1}{k}}\right]} \quad \text{.....................} (4.46)$$

Where C is the flow coefficient, approximately 1.0 for drill bit nozzles and 0.6 for bit orifices. A_t is in ft^2.

From the above equation, the velocity of the flow through the bit, v_n will be

$$v_n = \frac{q_b}{A_t(60)} = \frac{6.02}{60}CP_{ai}\sqrt{\frac{k}{(1-k)S_gT_{bh}}\left[\left(\frac{P_{bh}}{P_{ai}}\right)^{\frac{2}{k}} - \left(\frac{P_{bh}}{P_{ai}}\right)^{\frac{k+1}{k}}\right]} \quad \text{.........................} (4.47)$$

In Eq. 4.45, the air density at the bit can be expressed at

$$\rho_{air} = \frac{P_{ai}S_g}{R_aT_{bh}} \quad \text{...} (4.48)$$

Where P_{ai} is the Pressure just above the bit in the drill string and the temperature just above the bit is assumed to be the same as the bottom hole temperature. R_a = 53.36 ft-lb/lb-°R and S_g is the specific gravity of air. Substituting Eqs. 4.46, 4.47 and 4.48 into Eq. 4.45

$$F_j = 0.000518\frac{(6.02)^2}{60}\left(\frac{P_{ai}S_g}{R_aT_{bh}}\right)C^2A_tP_{ai}^2\left(\frac{k}{(1-k)S_gT_{bh}}\right)\left[\left(\frac{P_{bh}}{P_{ai}}\right)^{\frac{2}{k}} - \left(\frac{P_{bh}}{P_{ai}}\right)^{\frac{k+1}{k}}\right] \quad \text{.........} (4.49)$$

From Lyon's *Air and Gas Drilling Manual*[17], assuming a sub-sonic flow through the bit.

$$P_{ai} = P_{bh}\left[\frac{\left(\frac{w_g}{A_t}\right)^2}{2g\left(\frac{k}{k-1}\right)P_{bh}\gamma_{bh}} + 1\right]^{\frac{k}{k-1}} \quad \text{...} (4.50)$$

Substituting Eq. 4.50 into Eq. 4.49 and simplifying, yields

$$F_j = 3.1288\times10^{-4}\,\frac{S_g C^2 A_t}{R_a T_{bh}}\left(\frac{k}{(1-k)S_g T_{bh}}\right)P_{bh}^3\left[\frac{\left(\frac{w_g}{A_t}\right)^2}{2g\left(\frac{k}{k-1}\right)P_{bh}\gamma_{bh}} + 1\right]^{\frac{3k}{k-1}}\left[\left(\frac{\left(\frac{w_g}{A_t}\right)^2}{2g\left(\frac{k}{k-1}\right)P_{bh}\gamma_{bh}} + 1\right)^{\frac{-2}{k-1}} - \left(\frac{\left(\frac{w_g}{A_t}\right)^2}{2g\left(\frac{k}{k-1}\right)P_{bh}\gamma_{bh}} + 1\right)^{\frac{-(k+1)}{k-1}}\right]$$

$$F_j = 3.1288\times10^{-4}\,\frac{S_g C^2 A_t}{R_a T_{bh}}\left(\frac{k}{(1-k)S_g T_{bh}}\right)P_{bh}^3\left[\left(\frac{\left(\frac{w_g}{A_t}\right)^2}{2g\left(\frac{k}{k-1}\right)P_{bh}\gamma_{bh}} + 1\right)^{\frac{3k-2}{k-1}} - \left(\frac{\left(\frac{w_g}{A_t}\right)^2}{2g\left(\frac{k}{k-1}\right)P_{bh}\gamma_{bh}} + 1\right)^{\frac{2k-1}{k-1}}\right]$$

.... (4.51)

Where the weight rate of flow of air, $w_g = \gamma_g Q_g$ and $Q_g = \frac{q_g(scf/\min)}{60}(ft^3/\sec)$

and $\gamma_{bh} = \frac{P_{bh}S_g}{R_a T_{bh}}$.

The bottom hole pressure can be calculated using the expression below

$$P_{bh} = \left[\left(P_{at}^2 + b_a T_{av}^2\right) e^{\frac{2a_a H}{T_{av}}} - b_a T_{av}^2 \right]^{0.5} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(4.52)}$$

Where $b_a = \dfrac{f}{2g\left(D_h - D_p\right)}\left(\dfrac{R_a}{S}\right)^2 \dfrac{w_g^2}{\left(\dfrac{\pi}{4}\right)^2 \left(D_h^2 - D_p^2\right)^2}$ and $a_a = \left(\dfrac{S_g}{R_a}\right)\left[1 + \left(\dfrac{w_s}{w_g}\right)\right]$, where the weight rate of flow of solid rock cutting (lb/sec), $w_s = \left(\dfrac{\pi}{4}\right) D_h^2 (62.4) S_s \left[\dfrac{ROP}{3600}\right]$, where ROP is in ft/hr and S_s is the specific weight of the cuttings. The empirical von karman relationship for determining the fanning friction factor for the annulus is:

$$f = \left[\frac{1}{2\log\left(\dfrac{D_h - D_p}{e}\right) + 1.14} \right]^2$$

For follow-on calculations for flow in the annulus the absolute roughness for commercial pipe, e_p= 0.00015 ft will be used for the outside surfaces of the drill pipe and drill collars, and inside surface of the casing. The open hole surfaces of boreholes can be approximated with an absolute roughness, e_{oh} = 0.01 ft. The average absolute roughness of the annulus is approximated by using a surface area weight average relationship between the open surface area and its roughness and the outside surface of the drill string and its roughness. Thus the value for e_{avo} is

$e_{avo} = \dfrac{e_{oh}\left(\dfrac{\pi}{4}\right) D_h^2 H_o + e_p \left(\dfrac{\pi}{4}\right) D_p^2 H_o}{\left(\dfrac{\pi}{4}\right) D_h^2 H_o + \left(\dfrac{\pi}{4}\right) D_p^2 H_o}$, the term H_o cancels and the above reduces to

$$e_{avo} = \frac{e_{oh}\left(\frac{\pi}{4}\right)D_h^2 + e_p\left(\frac{\pi}{4}\right)D_p^2}{\left(\frac{\pi}{4}\right)D_h^2 + \left(\frac{\pi}{4}\right)D_p^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(4.53)}$$

$T_{bh} = T_r + GH$, where G is the temperature gradient and is usually 0.01 ˚F/ft and

$T_r = (t_r + 459.67)°R$, where t_r is the approximate average temperature at the surface location in ˚F.

The flowing assumptions were made while arriving at the model for flow through the bit nozzle:

1. That the flow is isentropic.
2. Elevation changes are ignored
3. The bottom hole temperature is the same as that just above the bit.

The equation for the jet impact force was tried out with example 8.3b, in Lyon's book (*Air and Gas Drilling Manual*). In the example P_{bh} = 29,454 lb/ft^2, T_{bh} = 604.41˚R, atmospheric pressure, p_{at} = 12.685 psia, P_{at} = p_{at} x 144 = 1,827 lb/ft^2,

$$\gamma_g = \frac{P_{at}S}{R_aT} = \frac{1827 \times 1}{53.36 \times 516.67} = 0.0659 lb/ft^3$$

$$\gamma_{bh} = \frac{P_{bh}S}{R_aT_{bh}} = \frac{29454 \times 1}{53.36 \times 604.41} = 0.913 lb/ft^3,\ Q_g = \frac{q_g(scf/\min)}{60} = \frac{2400}{60} = 40(ft^3/\text{sec})$$

$w_g = \gamma_g Q_g = 0.0659 \times 40 = 2.632 lb/\text{sec}$, P_{ai} = 31,330 lb/ft^2, k = 1.4 for air,

A_t = 0.00802ft^2

Using Eq. 4.49 above for simplicity

$$F_j = 0.000518\frac{(6.02)^2}{60}\left(\frac{P_{ai}S_g}{R_aT_{bh}}\right)C^2A_tP_{ai}^2\left(\frac{k}{(1-k)S_gT_{bh}}\right)\left[\left(\frac{P_{bh}}{P_{ai}}\right)^{\frac{2}{k}} - \left(\frac{P_{bh}}{P_{ai}}\right)^{\frac{k+1}{k}}\right]$$

$$F_j = 3.1288 \times 10^{-4} \left(\frac{31330 \times 1}{53.36 \times 604.41} \right) 0.00802 (31330)^2 \left(\frac{1.4}{(1.1.4)1 \times 604.41} \right) \left[\left(\frac{29454}{31330} \right)^{\frac{2}{1.4}} - \left(\frac{29454}{31330} \right)^{\frac{1.4+1}{1.4}} \right]$$

$$F_j = 0.02 lb$$

The force is almost negligible and this is not surprising because of the polytropic expansion of the air as it exits the bit nozzle.

In designing the air volume requirement for Air drilling, we use the minimum kinetic energy (3.0 ft-lb/ft^3) required at the bit to lift the cuttings, with this value we do not expect a lot of impact force. However, as argued earlier, we have to take the kinetic energy at the bit into account as it affects the cutting removal from the bit which in turn affects the ROP and MSE.

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

From the theoretical study of the air hammer and the derivation of the Mechanical Specific Energy model, the following conclusions were made:

1. A model for calculating Mechanical Specific Energy for air hammer drilling system has been developed as derived in Eq. 4.12.
2. The pneumatic impact force, as shown in Chapter IV, of air exiting the bit does not contribute to the total rock destruction impact force from the bit. Air however, is very important for operating the hammer and for the removal of cuttings from under the bit. Inadequate cuttings removal will, however, affect rate of penetration and Mechanical Specific Energy values.
3. The differential pressure across the hammer is an important input for calculating hammer impact energy and can be used to control the hammer impact energy and consequently the rate of penetration during drilling.
4. Mechanical Specific Energy values, when properly applied can be utilized as a qualitative indicator of formation pressure changes during drilling. For as formation pore pressure changes, Mechanical Specific Energy values will change as well.

5.2 Recommendations

1. Field Data is required for further validation of the Mechanical Specific Energy model for air hammer drilling systems.
2. In spite of the results from this research, we should consider the fact that experience in the use of Mechanical Specific Energy in air hammer drilling is very limited. Therefore, a proper field experimentation, application and possible modification (if necessary) of the model developed in this research is required.

3. A possible further research area will be to investigate the effects of pneumatics, cuttings build-up and friction on the Mechanical Specific Energy values during air hammer drilling.

NOMENCLATURE

WOB	Weight on Bit, lb
ROP	Rate of penetration, ft/hr
N	Bit revolutions per minute, rpm
T	Torque, ft-lb
Dia	Bit diameter, in
A	Circular area of the bit, in^2
v_n	Fluid vertical velocity, ft/sec
m	Fluid mass, lb
ρ	Density of fluid, lb-sec^2/ft^4
q	Flow rate of fluid, ft^3/sec
c_d	Nozzle discharge coefficient (0.95 is recommended value)
P_b	Bottom-hole pressure, lb/ft^2
F_j	Jet impact force, lb
A_t	Total nozzle area $\left(\frac{\pi}{4(32)^2}\sum D_n^2\right)$ $where \sum D_n^2 = D_1^2 + D_2^2 + ...D_n^2$
e	Coefficient of restitution, $e = \frac{v_{pB}}{v_{pA}}$
v_p, v_{pA}, V_{impact}	Piston impact velocity, ft/sec
v_{pB}	Piston reflected velocity, ft/sec
Pow_{raw}	Hammer raw power, ft-lb/min
m_{piston}, m_p	Piston mass, lb
m_b	Drill bit mass, lb
F	Impact frequency, blows/min
$\eta_{transmission}$	Energy transmission efficiency
w_{iFC}, w_{iRC}	Mass of air at point *i* of front chamber or rear chamber, lb
$\dot{w}_{iFc}$, $\dot{w}_{iRC}$	Air mass flow at point *i* of front chamber or rear chamber, lb/sec

F_p	Resultant force of piston, lb
A_p	Area of piston, ft^2
T_{iFc}, T_{iRC}	Air temperature of point *i* of front chamber (FC) or rear chamber (RC), º R
P_{iFC}, P_{iRC}	Air Pressure of point *i* of front chamber or rear chamber, lb/ft^2
P^*_{iFC}, P^*_{iRC}	Critical air Pressure of point *i* of front chamber (FC) or rear chamber (RC), lb/ft^2
c_p	Specific heat at constant pressure
k	Adiabatic expansion exponent
n	Polytropic expansion exponent
P_{d1}, P_{d3}	Discharge pressure at points 1&3, lb/ft^2
A_{RP}, A_{FP}	Piston rear area (RP) and front area (FP), ft^2
x_p	Piston position, ft
C_{m1FC}, C_{m3FC}	Front chamber input discharge loss coefficient (m1FC), output Discharge loss coefficient (m3FC)
C_{m1RC}, C_{m3RC}	Rear chamber input discharge loss coefficient (m1RC), output Discharge loss coefficient (m3RC)
V_{iFC}, V_{iRC}	Air specific volume at point *i* of front chamber (FC) or rear chamber (RC), ft^3
V_{mfront}	Passive volume front chamber, ft^3
V_{mrear}	Passive volume rear chamber, ft^3
v_{iFC}, v_{iRC}	Air specific volume at point *i* of the front chamber (FC) and rear chamber (RC), ft^3
R	Air constant for Ideal gas, ft-lb/lb-º R
C_{RC}, C_{FC}	Polytropic expansion constant
A_{d1RC}	Rear chamber input discharge area (d1RC), ft^2
A_{d3FC}	Front chamber output discharge area (d3FC), ft^2

For energy transmission efficiency model (units as in reference #16)

ρ	Density of steel
c	One dimensional wave velocity
A	Cross-sectional area of the anvil, and/or piston
K	Rock impact resistance index
τ	Duration of impact

For the impact energy of the air flowing through the bit

q_b	Air flow through the bit, ft^3/sec
P_{ai}	Pressure just above the bit in the drill string, lb/ft^2
P_{bh}	Bottom-hole pressure, lb/ft^2
T_{bh}	Bottom-hole temperature, º R
k	1.4 for air
S_g	Specific gravity of air
S_s	Specific weight of cuttings
A_t	Total nozzle area, ft
T_{av}	Average temperature of the hole. º R
G	Temperature gradient, º F/ft
D_h	Hole diameter, ft
D_p	Drill pipe outer diameter, ft
w_g	Weight rate of flow of air, lb/sec
e_p	Absolute roughness of commercial pipe, ft
f	Fanning friction factor
H	Hole depth, ft
R_a	Gas constant, ft-lb/lb-º R
C	Nozzle discharge coefficient

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