Simulation of the Influence of Head on the Power Output of a Pelton Turbine

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***Abstract* - This work is set out to investigate the influence of head on the power output of a pelton turbine. Simulation was performed on the head operating conditions to determine the power output of a pelton turbine. High head and low flow with increased pressure operating conditions deliver more energy to the bucket splitter to generate a force on bucket surface which drives or retards the bucket motion compared to the energy delivered to the bucket splitter by low head and high flow with decreased pressure operating condition.**

Key Words*: *Head, Power, Flow rate, Pelton Turbine, Pressure

I. INTRODUCTION

The high demand for a clean source of energy continues to increase as indicated by the increase in distributed generation technologies and adoption of renewable energy resources. Climate change and global warming have made renewable energy the most appropriate and fitting means of answering all these changes in our environment. Micro-hydro power plant (MHPP) is considered as one of the most reliable renewable energy in the world**.** It is also one of the earliest small scales renewable energy and is still an important source of energy today. MHPP are appropriate in most causes for individual users or groups who are independent of the electricity supply grid. A MHPP is generally a hydroelectric power installation that can produce up to 100 kW of power. It does not encounter the problem of population displacement and is not expensive as solar or wind energy. To attain the megawatt power required in the developing countries as Nigeria requires optimizing our dams by introducing Pelton turbine for power generation scheme or upgrading the available sources of energy in the country **[1]**.

This paper therefore sets out to investigate through simulation the effect of head on the power output of a pelton turbine in order to enhance the power generation in Nigeria. In order to do this, development of mathematical equation for hydraulic power delivered by the jet of water to the wheel at varied head was done which influences the shaft output to the generator. Determination was also made with equation of the head delivered by the pump to the fluid, and consequently the power delivered to the turbine by the water.

II. DETERMINATION OF HEAD FOR MICRO HYDRO-POWER SITE

In a potential micro hydropower site, head is the vertical distance of waterfall. When evaluating a potential site, head is usually measured in feet, meters, or units of pressure. Head also is a function of the characteristics of the channel or pipe through which it flows. Most micro hydropower sites are categorized as low or high head. The higher the head the better because less water will be needed to produce a given amount of power and smaller, less expensive equipment can be used. Low head refers to a change in elevation of less than 10 feet (3 meters). A vertical drop of less than 2 feet (0.6 meters) will probably make a small-scale hydroelectric system unfeasible.

However, for extremely small power generation amounts, a flowing stream with as little as 13 inches of water can support a submersible turbine. This type of turbine was originally used to power scientific instruments towed behind oil exploration ships. When determining head, consideration is to be given to both gross head and net head. Gross head is the vertical distance between the top of the penstock that conveys the water under pressure and the point where the water discharges from the turbine. Net head equals gross head minus losses due to friction and turbulence in the piping.

A. Pressure Head for an Impulse Turbine System

Hydro power is obtained from the potential and kinetic energy of water flowing from a height. The energy contained in the water is converted into electricity by using a turbine coupled to a generator.

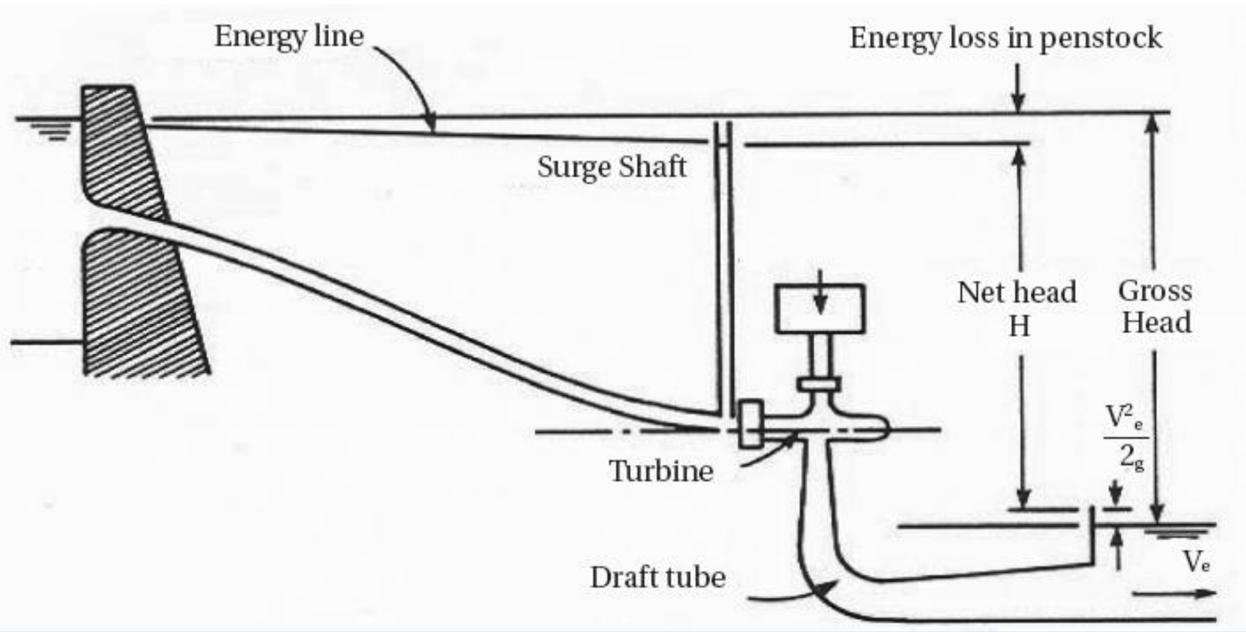

Fig.1. Components of MHPP unit.

A hydro power resource can be measured according to the amount of available power or energy per unit time. The power of a given situation is a function of head and rate of flow. (Figure1)

The energy in a MHPP starts out as potential energy by virtue of its height above the power house. Water under pressure in the penstock is able to do work when released so there is energy associated with the pressure as well. The transformation of energy is from potential to pressure to kinetic energy. The total energy is the sum of the potential, pressure and kinetic in a run – off – the river system.

Energy head per unit mass = $z + \frac{p}{\gamma} + \frac{v^2}{2g}$ [1]

Where z – elevation from a reference height (m)

p – Pressure (Nm^{-2})

γ – Specific weight (Nm^{-3})

v – Average velocity (ms^{-1})

g – Gravitational acceleration ($9.81ms^{-2}$)

III WORKING EQUATIONS

A. Determination of Nozzle Velocity V_{2jet}

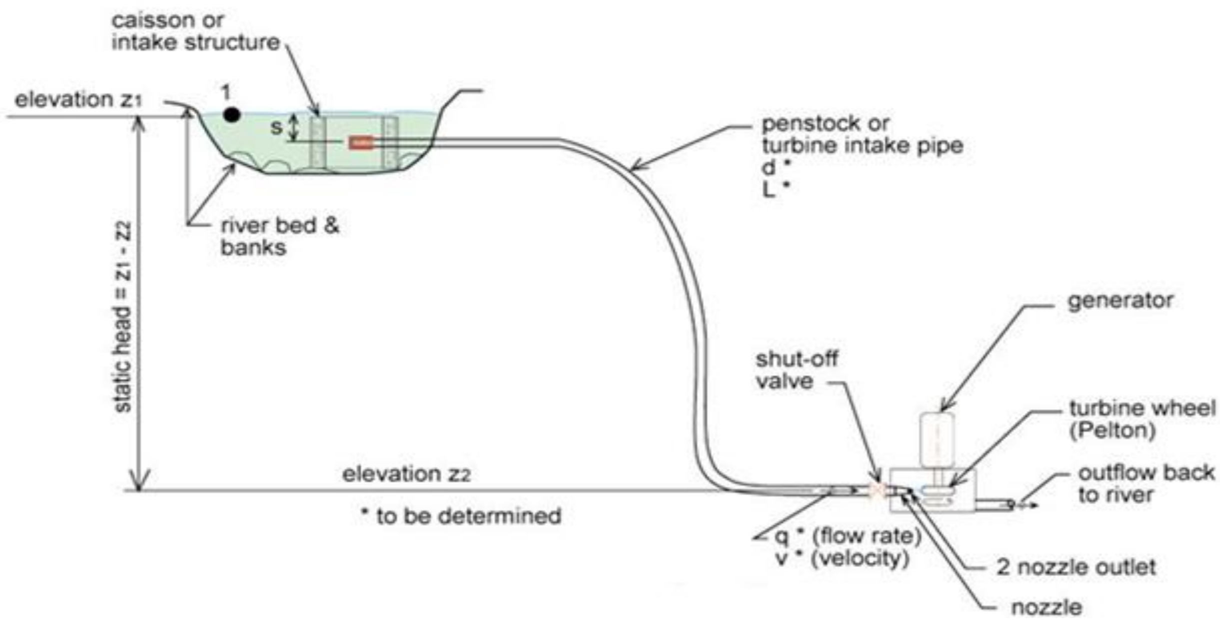

Fig.2 Flow from the reservoir to the nozzle outlet

Water to drive a pelton wheel is supplied through a pipe from a river bed or lake as shown above. The head loss due to friction in the pipeline will be considered while minor losses neglected.

Calculations for hydro turbine jet impact velocity are based on the same sort of calculations done for pump systems, except there is no pump. The energy is provided by the difference in elevation between the inlet and outlet of the system, as shown in figure2 .The inlet (point1) is defined as the surface elevation of the water source and the outlet is at the nozzle outlet (point2), the velocity v_1 is the velocity of fluid particle at the water source surface; the velocity v_2 is the velocity of the water jet at the nozzle. The pressure head at point 1 and 2 is equal to zero. Applying the Bernoulli's equation to link the flow from the reservoir to the nozzle outlet would yield:

$$Z_1 - Z_2 - H_f = \frac{V_2^2}{2}$$

$$V_{jet} \text{ (m/s)} = \sqrt{2 \, x \, g \, (Z_1(m) - Z_2(m) - H_f \, (m)} \quad \textbf{[2, 3]} \qquad [2]$$

B. Developed Power

Power delivered by flowing fluid to the pelton wheel is given as

P = ρgQH or

$$P = ½ \, \rho v^2 Q \qquad [3]$$

Where H is the available head and Q is the flow rate from the nozzle, ρ is the water density and v the water jet nozzle velocity. Either of these expressions gives theoretical power available from water jet. The pelton turbine is designed to produce maximum power when the peripheral speed is ½ of the water jet speed. The power transmitted to the turbine wheel is 0.5γHQ or 0.5 ρgQH. The 50 % is theoretical and is based on the fact that water jet is reversed due to the wheel cup design 180^o back towards its source the reversed water jet is no exactly in line so that the wheel itself has a real world efficiency of 90 % or better.

Power at the turbine wheel is

$$\text{Power} = 0.9 \text{ x } 0.5 \, \rho g Q H \qquad [4]$$

$$P = 0.9 \text{ x } 0.5 \text{ x } ½ \text{ x } \rho \text{ x } Q \text{ x } 1/g_c \text{ x } v_{jet}^{\;2} \qquad [5]$$

Force generated on the bucket surface due to splitter angle is given by:

$$F_{bucket} = \dot{m} \, (V_{jet} - U) \, (1 - \cos\beta_1) \qquad [6]$$

C. Shaft Power Output $\dot{W}_{shaft}$:

The shaft output is not just influenced by the force generated on the bucket as the jet impinges on the bucket splitter angle causing the rotary motion of wheel thereby affecting the shaft power output and the torque on shaft but the head which delivered energy through the nozzle to the bucket, therefore shaft power output is as a result of head on the system

The torque to the shaft is

$T = F_{bucket} \, r_m$

$$P_{shaft} = \rho \, Q \, U \, (U - V_{jet}) \, (1 - \cos\beta_2) = 2\rho Q \, (U^2 - UV_{jet}) \; \textbf{[4, 5]} \qquad [7]$$

Since $V_{jet} > U$, shaft work being done is negative since work is done on the system and torque is maximum when U = 0, also work is being done when the wheel is not turning, therefore shaft output is maximum when $d\dot{W}_{shaftmax} = 0$. So any value closer to zero from the negative axis has a greater output **[6]**

$$T_{shaft} = \frac{2\rho Q \, (U^2 - UV_{jet})}{\omega \, ({}^{rad}/_{s})} \qquad [8]$$

D. The Power Delivered to the Generator

The power delivered to the wheel is as a result of the energy the head delivers to the pelton wheel thereby influencing the shaft power from which the power required to convert the (mechanical energy) rotary motion of shaft to electricity (electrical energy) from the generator is achieved. This power is expressed as **[7]**

$$P = 2\pi N_S \frac{2\rho Q \, (U^2 - UV_{jet})}{\omega \, ({}^{rad}/_{s})} \, / \, 60_S \qquad [9]$$

IV. SIMULATION

Simulation Flow Chart

START

INPUT: l_p ṁ D_p D_n D_t
h_f β_1 β_2 γ ρ g H D_b

$$V_{jet} = \sqrt{2 \ x \ g \ (Z_1(m) - Z_2(m) - H_f(m)}$$

Power = 0.9 x 0.5 ρgQH

$$arc\ cot\gamma = \left(\frac{D_t - D_n}{\varphi_1{}^{1/2}\, k_{co} - \frac{D_b}{D_t} \sqrt{1 - \left(\frac{D_t - D_n}{D_b}\right)^2}} \right)$$

$$P_{wheel} = 2\rho g H \left(\frac{V_{jet}}{\sqrt{\varphi}} - \frac{1}{\sqrt{\varphi}}\right) \text{x} \ (1 + (1 - \Delta) \text{x} \ Cos\beta_2$$

$$F_{bucket} = \dot{m}\, r_m (V_{jet} - \text{U})\ (1 - \cos\beta)$$

$$\omega_{turbine} = 0.5 \ \text{x} \ 1.91 \ \text{x} \ 10^4 \ \text{x} \ V_{jet} \left(\frac{m}{s}\right)$$

$$N_s = \frac{N\sqrt{p}}{H^{\frac{5}{4}}}$$

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$$\eta_{max} = \frac{1 - cos\beta}{2}$$

$$\dot{W}_{\dot{W}shaft} = 2\rho Q\ (U^2 - UV_{jet})$$

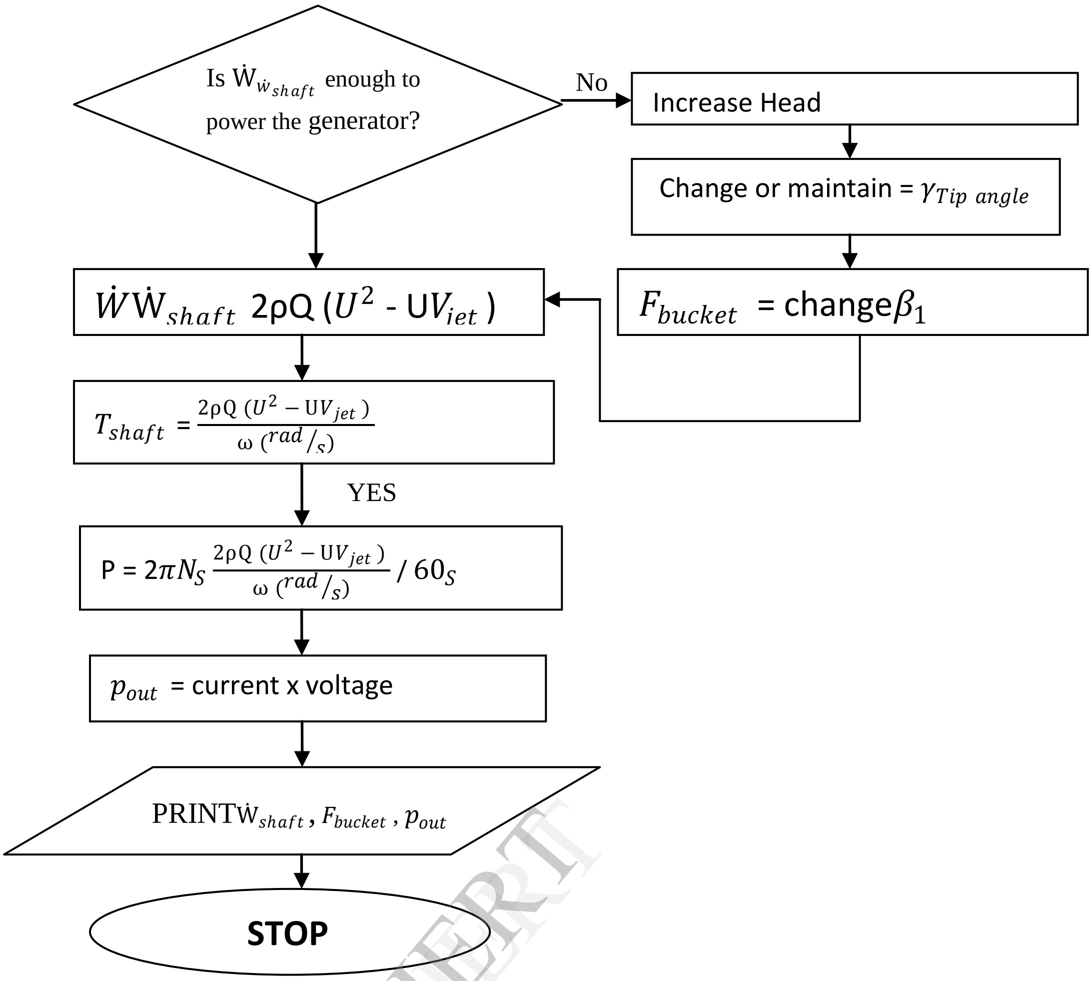

Fig 3: Simulation program flow chart

A. Simulation Dialog Box:

Showing the input values to be simulated during our test process, power output which determines the effectiveness of the turbine will be compared to the result from the pelton turbine in the lab to predict the turbine head best for power output which the MHPP will use to attain the required megawatt power the country needs.

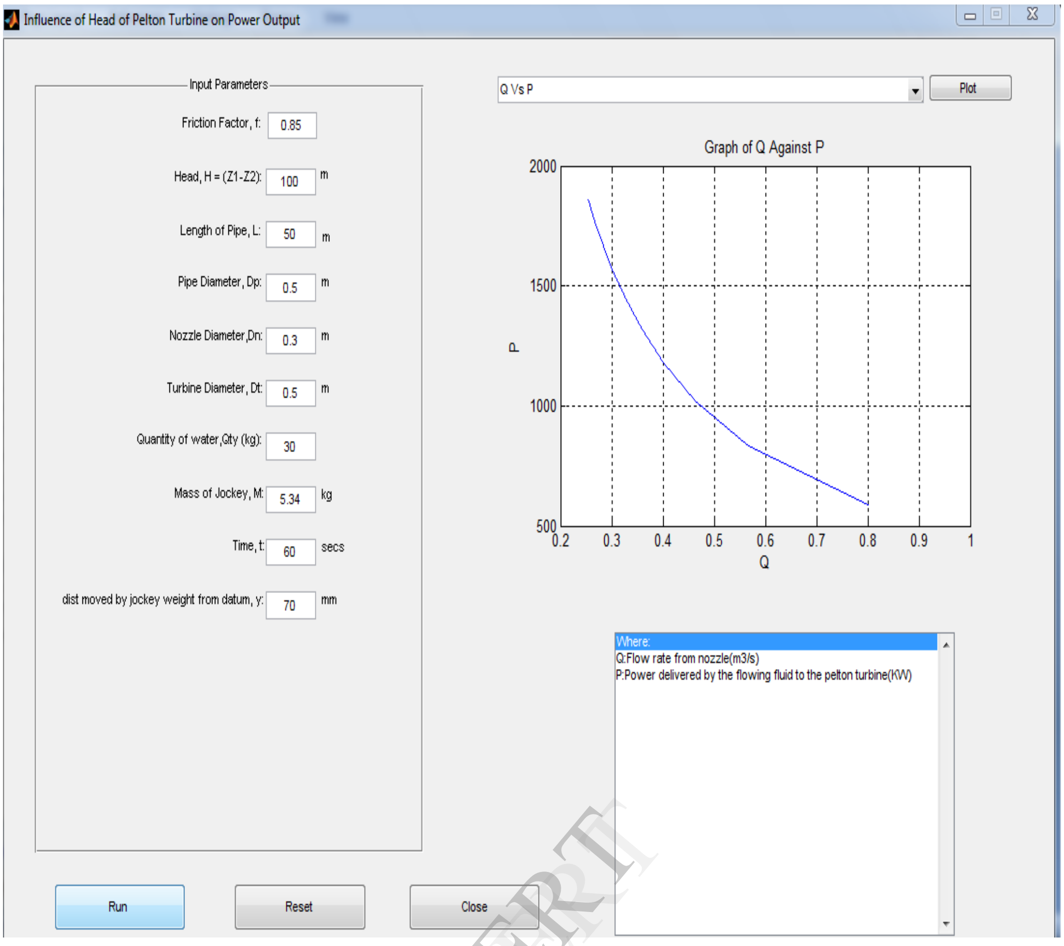

Fig.4.The simulation dialog box with the input values

B. Results and Discussion

The following graphs were obtained:

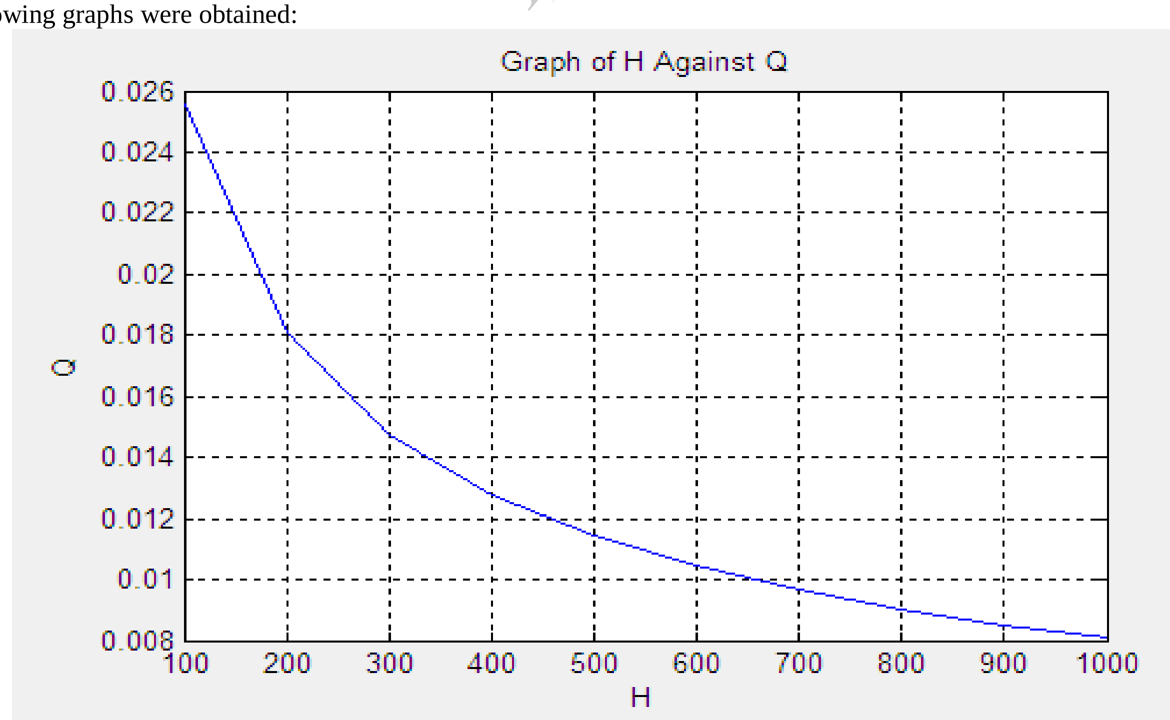

Fig.5. Graph of flow rate from nozzle against head

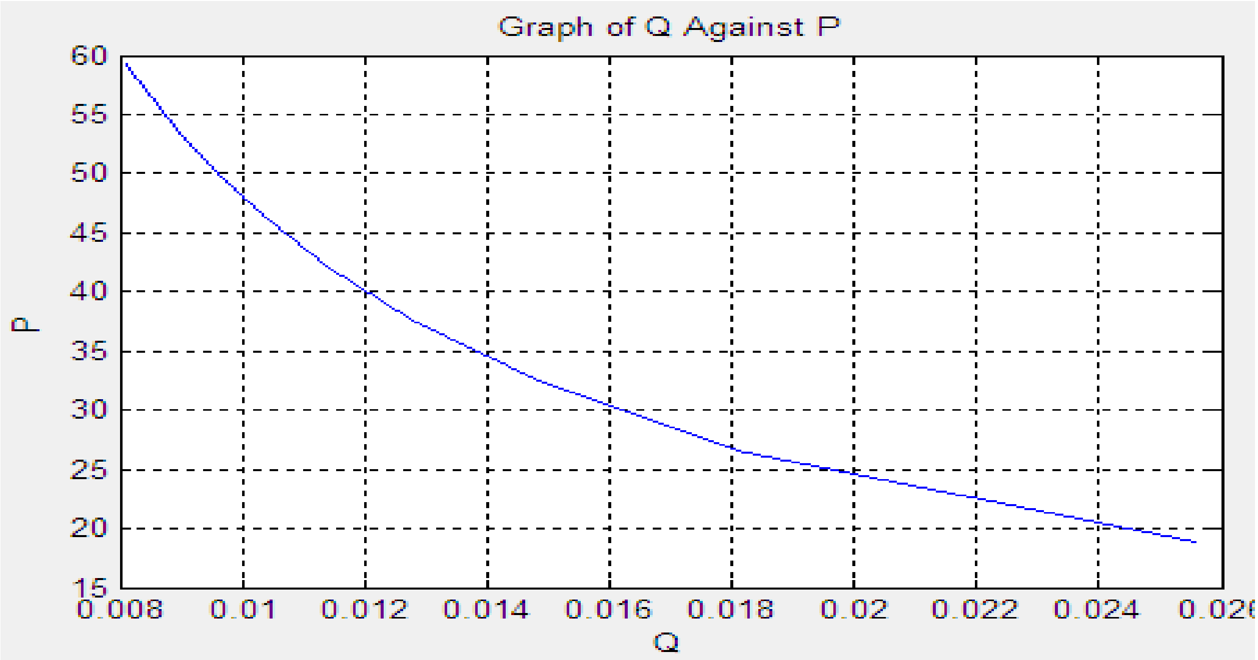

Fig.6. Graph of power on the wheel against flow from nozzle

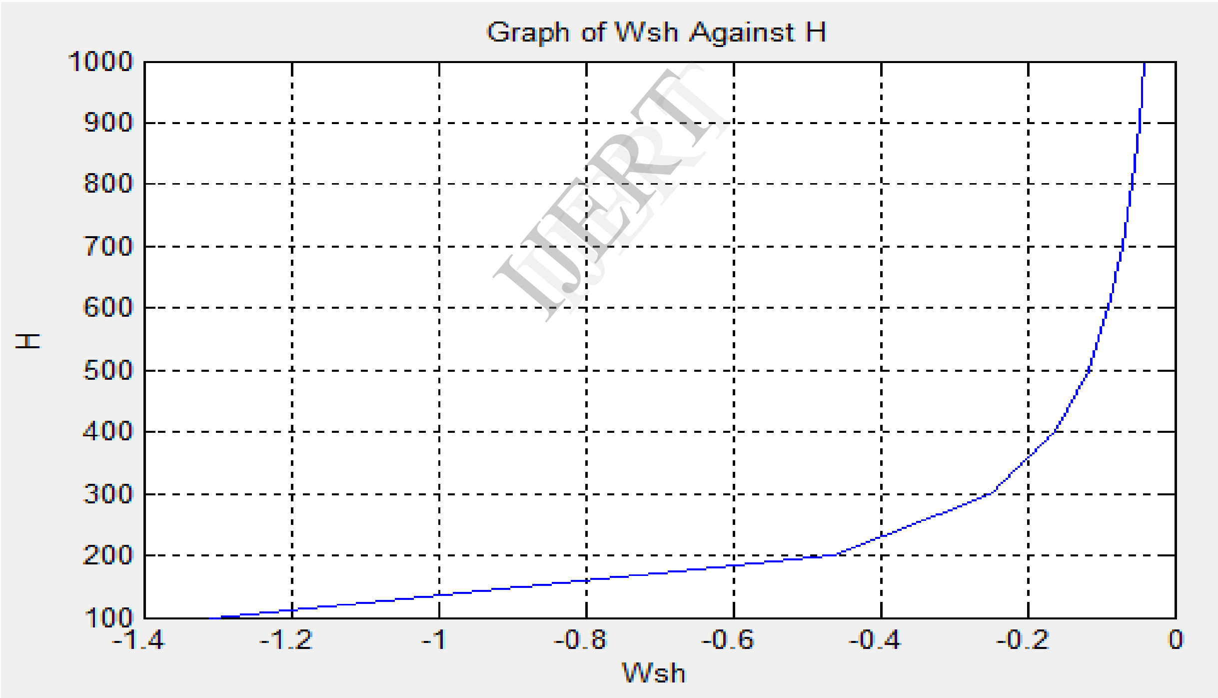

Fig.7. Graph of shaft power output against head.

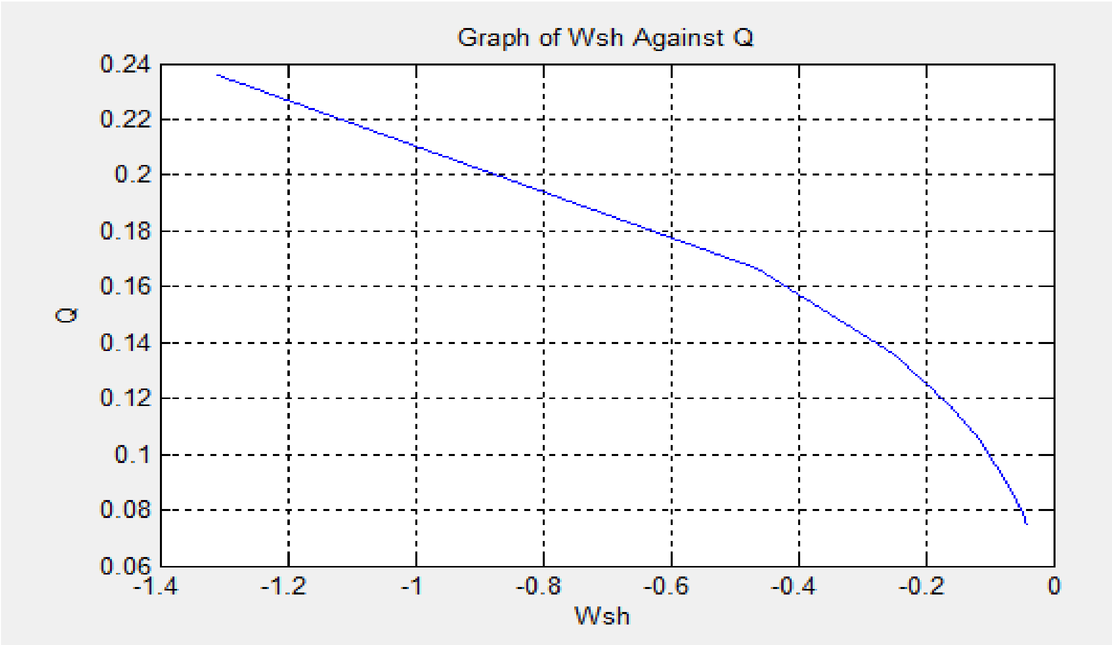

Fig.8. Graph of volumetric flow against shaft power output

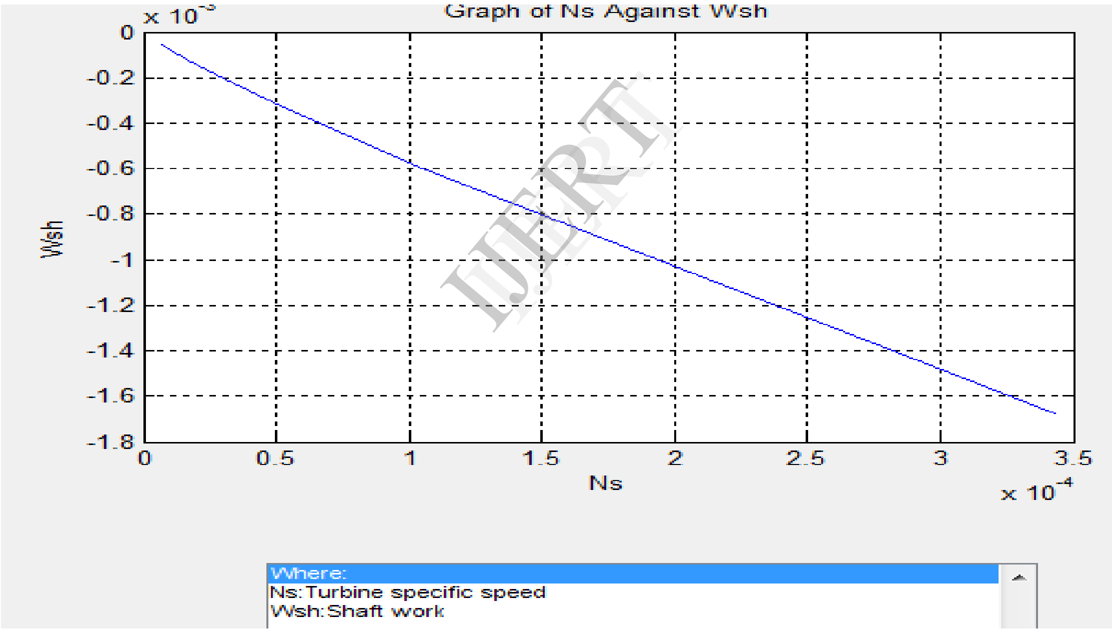

Fig.9. Graph of turbine specific speed against shaft work

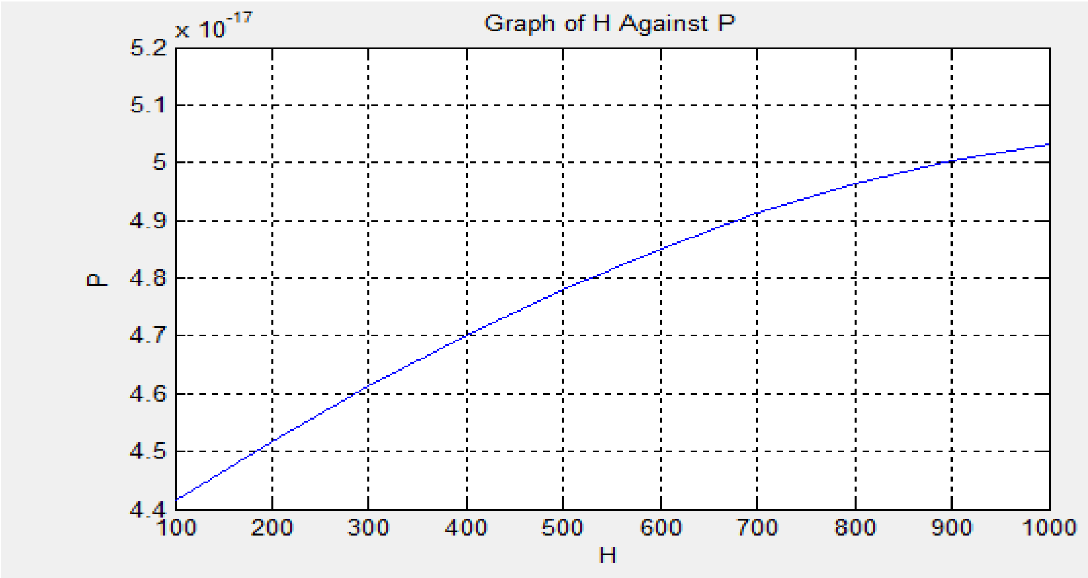

Fig.10. Graph Head against power to generator.

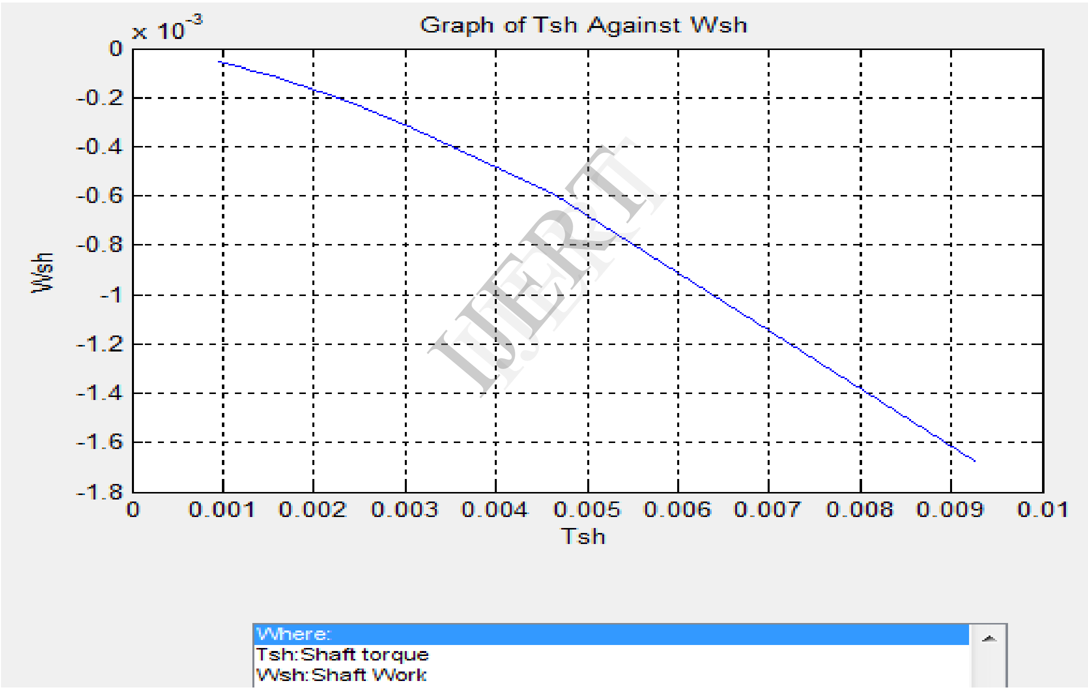

Fig.11. Graph of torque against shaft output

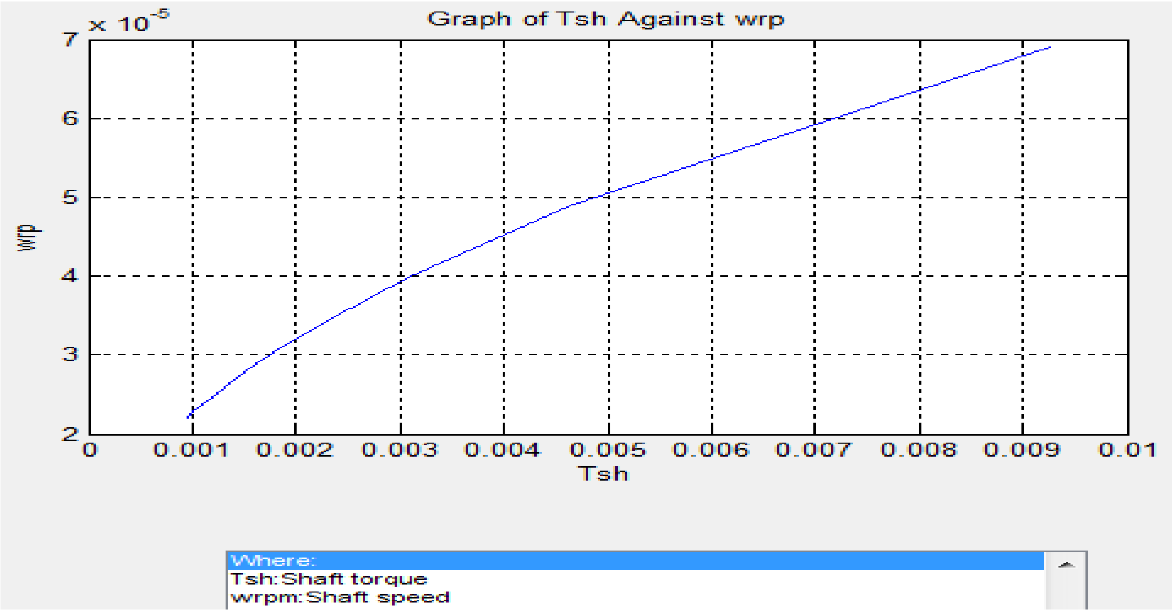

Fig.12. Graph of torque against shaft speed

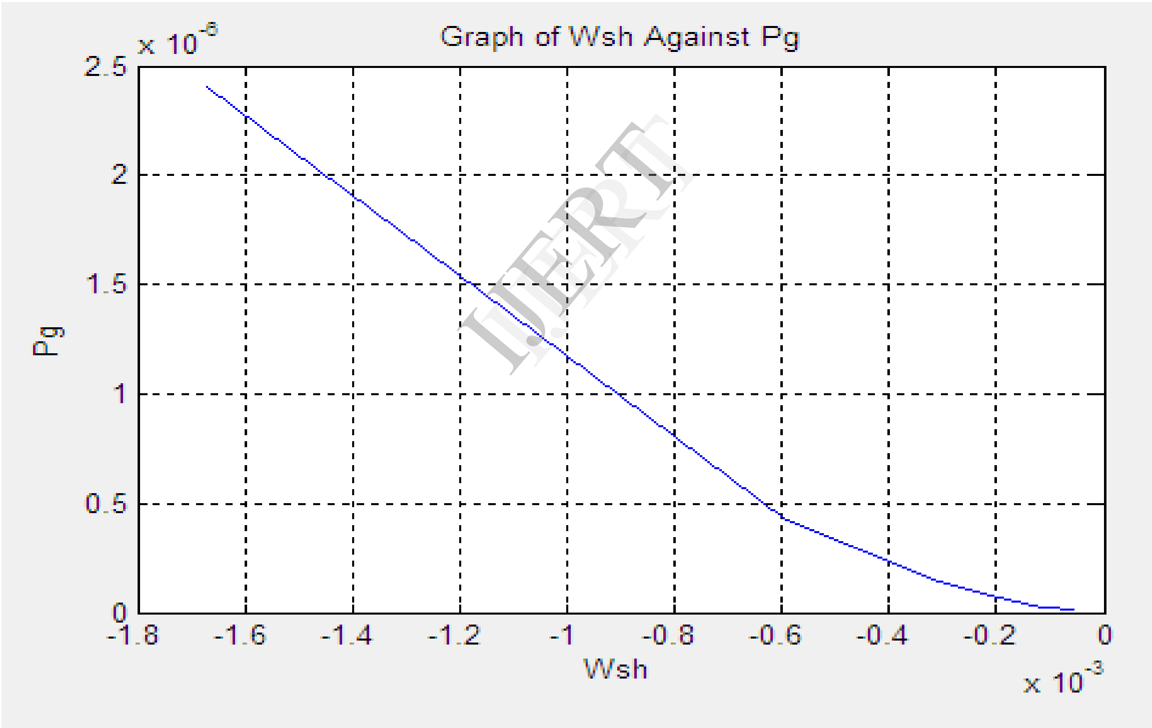

Fig.13. Graph of power delivered to the generator against shaft output

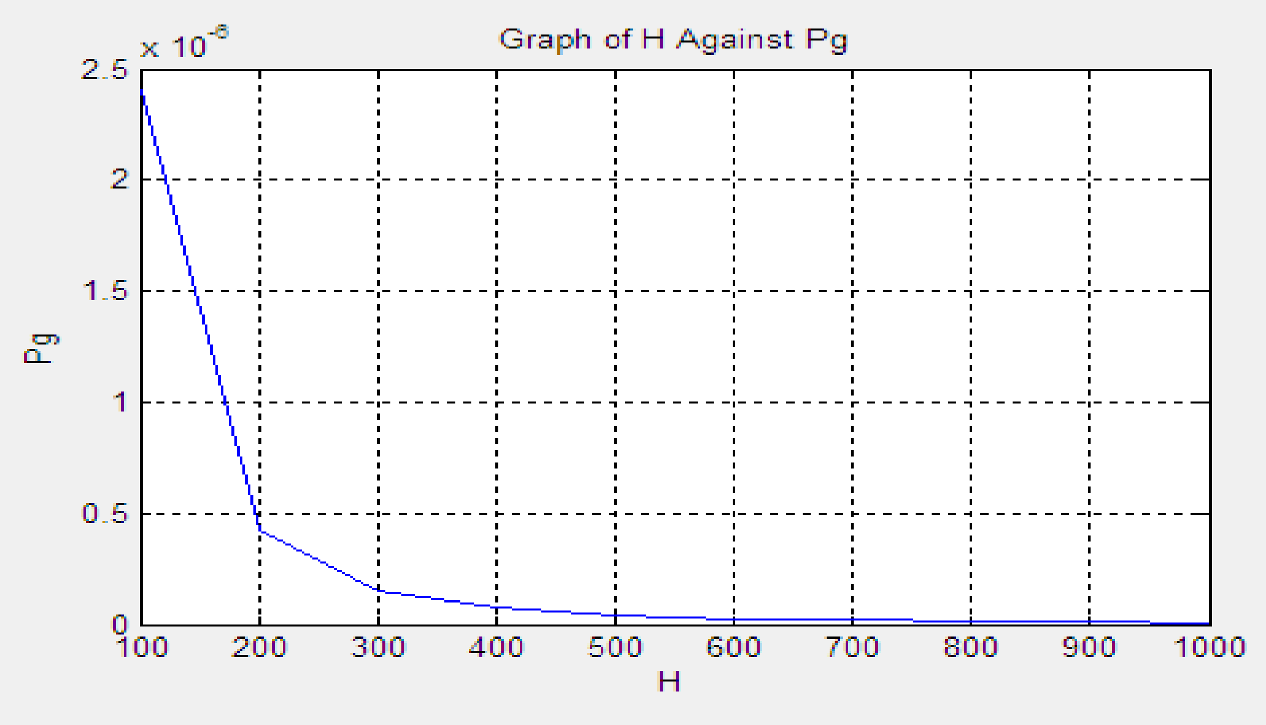

Fig.14. Graph of power delivered to the generator against Head

From Figure 5, the graph of head against flow rate through the nozzle, as the head increases from 100m to 1000m, the jet of water from the nozzle impinging on the bucket was decreasing ($0.24m^3$/ kg to $0.06m^3$/ kg). With increased head, therefore, the force it will generate on the bucket surface will also increase. As the specific volume of water decreases ($0.24m^3$/ kg to $0.06m^3$/ kg) the pressure is increased, which definitely influences the power delivered to the wheel by the jet of water.

From figures 6 and 7, as the jet of water through the nozzle impinges on the bucket, it causes the rotary motion of the wheel. Power is being delivered to the wheel by the flow rate through the nozzle. As the flow decreases ($0.24m^3$/ kg to $0.06m^3$/ kg) with increased elevation or head (100m to 1000m), power delivered to the wheel is increased, while the shaft power output increases with negative value as the head or elevation increases from 100m to 1000m. When the absolute velocity of the fluid exiting the turbine is zero the shaft power become maximum, and the velocity of the jet is greater than that of the wheel so the value of the shaft power become negative as the head increases, so we have negative shaft work (at low speed) which increases the high speed shaft with the help of a gear to transmit the power, so negative shaft gives a better output.

From figure 8, as the volumetric flow rate decreased from $0.24m^3$/ kg to $0.06m^3$/ kg with increased head and pressure the shaft output was increasing.

From figures 9 and 10, the jet of water which impinges on the bucket splitter increases the turbine speed. As the shaft speed increases the turbine speed also increases toward the negative graph thereby increasing the power output from the shaft. Shaft work is maximum when we have a negative work and the absolute velocity exiting the turbine is zero.

As the head increases in elevation from 100m to 1000m, the hydraulic power in water fall which is given a direction in a pipe line increases the power delivered to the wheel from 4.4 to 5.05 $x10^{-17}$.

From figures 11 and 12, as the speed of wheel was constantly increased, the shaft output was increased which affect the torque on shaft thereby increasing the torque on the shaft and as the speed of shaft was increased from $2x10^{-5}$rpm to $7 \times 10^{-5} rpm$ with increased head (100 – 1000m) the torque on shaft was affected positively, increasing from 0.001 to 0.01 showing a better output.

From the results obtained from the graph we can deduce that the turbine is operating on high head and low flow conditions.

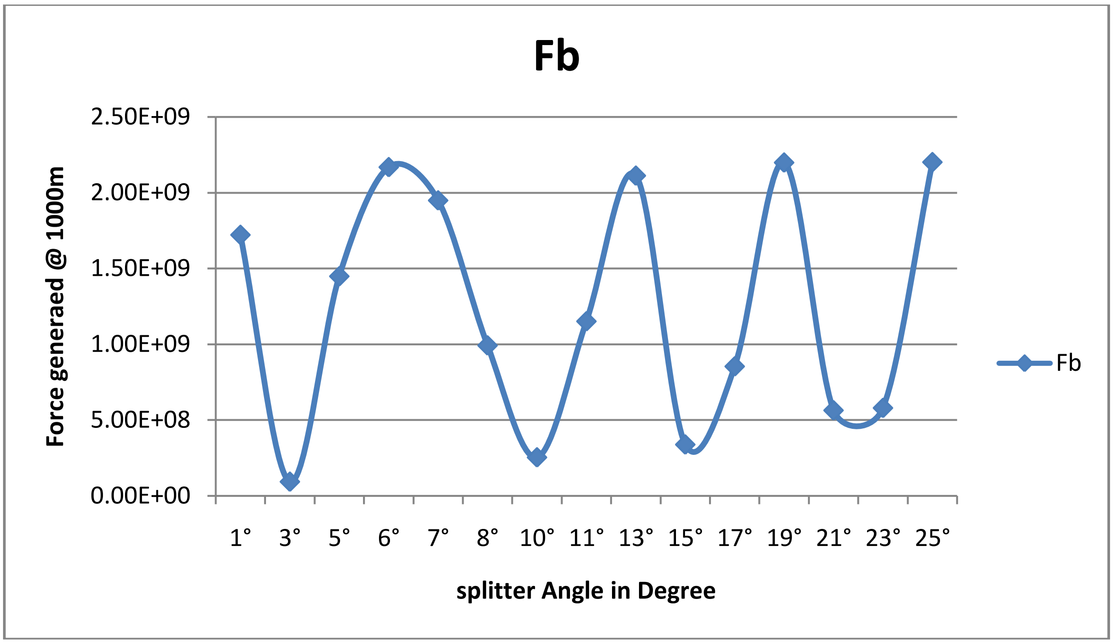

Fig.15: Graph of force generated at 1000m Vs splitter angle

The graph above shows the force generated at 1000m head for both the predicted and already existing splitter angles for pelton bucket and this was calculated using $\frac{dF}{d\theta} = 0$.

From the reading we can say that the bucket generated greater force at 25° followed by 6° 19° and 13°, showing that the predicted angle could give a better result compared to that of the existing angles (10° – 15°).

V. CONCLUSION

The modeling equation for the simulation gave a high head and low flow operation. As the head was increasing in elevation from 100m – 1000m, the flow through the nozzle was reducing with increased pressure. Therefore the jet through the nozzle impinging on the bucket directed more force on it which caused the bucket to deflect faster thereby rotating the wheel and shaft. Also as the head was increasing the power delivered to the generator was also increasing and hence the output from the generator was affected positively. From the result obtained through the simulation model and other researches made on pelton turbine we can conclude that high head and low flow with increased pressure is suitable for commercial use or for large MHPP over 1000megawatt output which when applied to the Nigerian power situation will definitely improve the power generation, since we have been blessed with a number of dams in the country.

Nomenclature

A_1= Area at which water enters the pipe from the river bed to turbine.
A_2 = Area of the nozzle (m)
C_v= coefficient of velocity at nozzle

D_n, D_{jet} ,= diameter of nozzle (m)
D_t = Turbine diameter (m)
dL = angular momentum
dt = time in seconds
E = specific energy
E_{max} = maximum energy developed or head
E_1 = Energy possessed by the fluid at section 1
E_2 = = Energy possessed by the fluid at section 2
g = gravity (m/sec2)
F = force generated on the vane
F_j = force of the jet in the x- direction
H = total head developed or the elevation (m)
h_L = Energy losses from the system due to friction in the pipe or minor losses.
h_R = Energy removed from the fluid by the turbine motor

h_A = Energy added to the fluid by the pump
J = is the rate of momentum flow in jet at impact on plate
K_{uwheel} = frictional coefficient of wheel
L = Distance from center of vane to the pivot
L = Length of pipe
MHPP = micro hydro power plant
N_s= specific speed of the wheel
N_{wheel} = speed of the wheel
PCD = pitch center diameter
p_n = pressure at nozzle exit
p_{imp} = pressure at impart with vane
P, P_{max} = power developed to the wheel by jet of water (kW)
Q = volumetric flow rate from the nozzle (kg/m^3)
r = Radius of the runner at striking fluid (m)
S = Height of the vane above the nozzle (m)
T = torque (N)
U = Absolute velocity of the bucket (m/s)
u_1, v_{jet} = absolute velocity at nozzle exit (m/s)
u_{imp} = velocity at impact with the vane (m/s)
u_{wheel} = The wheel speed (m/s)
V_1 = Absolute velocity of the jet just before striking the bucket (m/s)
V_2 = Absolute velocity of the fluid leaving the bucket (m/s)
V_{w1} = Whirl velocity at inlet (m/s)
V_{w2} = Whirl velocity at exit (m/s)
V_{r1}, V_{r2} relative velocity of the bucket (m/s)
W = Jockey weight (kg)
Y = Jockey displacement (m)
ZΔ = Elevation from the river bed to wheel (m)
Z_{imp} = the elevation of impact with the hemispherical cup (m)
Z_n = the elevation of the nozzle (m)
ω = Angular velocity of the wheel (rad/sec)
$\dot{m}$ = Mass flow rate (kg/sec)
$\dot{m}_w$= Mass flow rate (kg/sec)
ρ = density of fluid (kg/m^3)
β_1 = bucket inlet angle
β_2 = angle through which fluid is deflected by bucket.
η_{wheel} = efficiency of the wheel
$\eta_{turbine}$ = efficiency of the turbine
$\dot{w}$ $\dot{w}_{shear}$ = work on shear
$\dot{w}$ $\dot{w}_{other}$ = work on the surrounding
$\dot{w}$ $\dot{w}_s$ = work on the system

REFERENCES

1. Nwocha E. N (2013) "Effect of head and Bucket Splitter Angle on the Power Output of a Pelton Turbine". MENG Thesis, Department of Mechanical Engineering, Nnamdi Azikiwe University, Awka, Nigeria.
2. Ainley, D.G and Matheson (1957)."A Method of performance Estimation G. C. R for Axial flow turbines, Aero". Res. Council and m. 2974.
3. Rajput, R.K (1998). "Fluid mechanics and hydraulic Machines". New Delhi S.Chend and co. Ltd.
4. Brackbill, J. U., Kothe, D. B., and Zemach, C. (1992), "A continuum method for modeling surface tension". Journal of Computational Physics 100 335–354
5. Brekke, H B. Velensek and M. Bajd, B. Velensek and M. Bajd,(1984). "A general study of the design of vertical pelton turbines". Eds., vol. 1, pp.383–397. Proceedings of the conference on hydraulic machinery and flow measurements.
6. Zhang, Zh. Muggli, F., Parkinson, E., and Scha¨rer, Ch.(2000), " Experimental investigation of a low head jet flow at a model nozzle of a Pelton turbine". Proceedings of the 11th International Seminar on Hydropower Plants, Vienna, Austria, pp. 181–188.
7. I. U. Attaneyake (2000). "Analytical study on flow through pelton turbine using boundary layer theory". International Journal of Engineering & technology by IJET VOL: 9 NO: 19. Mechanical engineering dept, Open University of Sri Lanka Nawala.