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An appropriate Methodand Analysis forDetermination of Energy losses in Compressors

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Abstract: *This work investigates the energy losses in compressors used inUnited Nigerian Textile Limited, Kaduna. Simulation was carried out using Hysys simulation software in order to validate the experimental results and simulated results obtained. From the Experiment carried out, it was discovered that the power wasted was 2.563kW. The annual energy and cost savings were found to be11,700kWh/yr and ₦99450/yr respectively. The exit Temperature and Pressure were recorded as 172.8ºC and 412.0kPa.The results are very helpful for validating the performance of any system.*

Significances of the Paper: *Evaluation of energy losses is very vital in determining the economic analysis of energy systems.*

Keywords: *Energy savings, Simulation, Compressor, Isentropic work and Polytrophic compression.*

I. Introduction

1.1 Basic Calculations

For a compression process, the system pressure P is related to the volume V by:

$PV^n = Constant$ --- (1)

Where:
n = Exponent

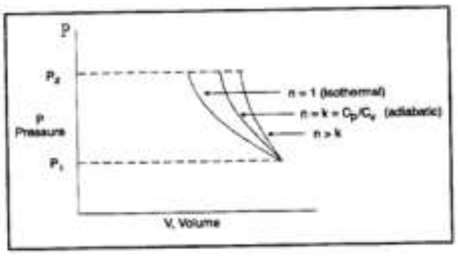

Fig.1 polytropic compression curve

The curve denoted by n=1 is an isothermal compression curve. For an ideal gas undergoing adiabatic compression, n is the ratio of specific heat at constant pressure to that at constant volume, i.e.,

$n = K = C_P / C_v$ --- (2.)

Where:
k = Ideal gas isentropic coefficient
c_p = Specific heat at constant pressure
c_v = Specific heat at constant volume
For a real gas, n > k.

The Mollier chart (Figure 2) plots the pressure versus the enthalpy, as a function of entropy and temperature. This chart is used to show the methods used to calculate the outlet conditions for the compressor as follows:

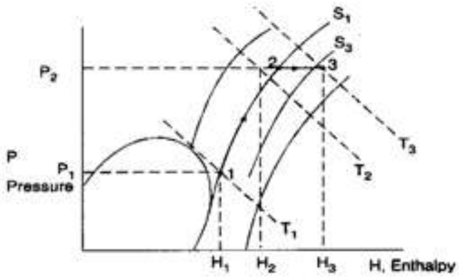

Fig.2:pressure versus the enthalpy

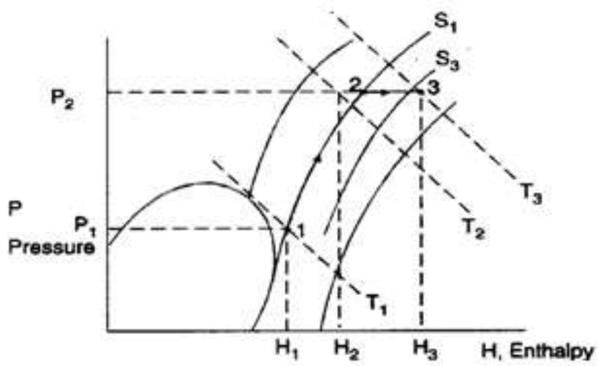

Figure 3 Typical Mollier chart for compression

A flash is performed on the inlet feed at pressure P_1, and temperature T_1, using a suitable K-value and enthalpy method. The entropy S_1, and enthalpy H_1 are obtained and the point (P_1, T_1, S_1, H_1) are obtained.
The constant entropy line through S_1 is followed until the user-specified outlet pressure is reached. This point represents the temperature (T_2) and enthalpy conditions (H_2) for an adiabatic efficiency of 100%. The adiabatic enthalpy change ΔH_{ad} is given by:

$$\Delta H_{ad} = H_2 - H_1 \text{-- (3)}$$

- If the adiabatic efficiency, γ_{ad}, is given as a value less than 100 %, the actual enthalpy change is calculated from:

$$\Delta H_{ac} = \Delta \frac{H_{ad}}{\gamma_{ad}} \text{--- (4)}$$

- The actual outlet stream enthalpy is then calculated using:

$$H_3 = H_1 + \Delta H_{ac}\ \mathrm{H_3}\text{-- (5)}$$

- Point 3 on the Mollier chart, representing the outlet conditions is then obtained. The phase split of the outlet stream is obtained by performing an equilibrium flash at the outlet conditions.
- The isentropic work (W_s) performed by the compressor is computed from:

$$W_s = (H_3 - H_1) \times J = \Delta H_{ac} \times J \text{ --- (6)}$$

Where:

J = mechanical equivalent of energy

The isentropic power required is:

$$GHP_{ad} = \Delta H_{ad} \times 778 \times \frac{F}{33000} \text{ --- (7)}$$

$$GHP_{ac} = \Delta H_{ac} \times 778 \times \frac{F}{33000} = \frac{GHP_{ad}}{\gamma_{ad}} \text{ --- (8)}$$

$$HEAD_{ad} = \Delta H \times 778 \text{ --- (9)}$$

Where:
GHP= work, hP
ΔH = enthalpy change,
F =mass flow rate,
$HEAD_{ad}$ =Adiabatic Head,
The factor 33000 is used to convert the units into horsepower.
The isentropic and polytropic coefficients, polytropic efficiency, and poly tropic work can be calculated using one of the two methods; the method from the *GPSA Engineering Data Book,* and the method from the *ASME Power Test Code 10.*

ASME Method

This method is more rigorous than the default GPSA method, and yields better results over a wider range of compression ratios and feed compositions.
For a real gas, as previously noted, the isentropic volume exponent (also known as the isentropic coefficient), n_s, is not the same as the compressibility ratio, k. The ASME method distinguishes between k and n_s for a real gas.

Adiabatic Efficiency Given
In this method, the isentropic coefficient $n_s$is calculated as:

$$n_s = \ln(P_2 / P_1) / \ln(V_1 / V_2) \text{ -- (10)}$$

Where:
V_1 = Volume at the inlet conditions
V_2 = Volume at the outlet pressure and inlet entropy conditions
P_1= Pressure at the inlet conditions
P_2= Pressure at the outlet conditions
The compressor work for a real gas is calculated from equation (2.8), and the factor from the following relationship:

$$W_{ac} = 144[n_s /(n_s - 1)] \times f \times P_1 \times V_1 \times [(P_2 / P_1)^{(n_s - 1)/n_s} - 1] \text{ ---------------------------- (11)}$$

The ASME factor f is usually close to 1. For a perfect gas, f is exactly equal to 1, and the isentropic coefficient $n_s$is equal to the compressibility factor k.
The polytropic coefficient, n, is defined by:

$$n = \ln(P_2 / P_1) / \ln(V_1 / V_2) \text{ -- (12)}$$

The polytropic work, i.e., the reversible work required to compress the gas in a polytropic compression process from die inlet conditions to the discharge conditions is computed using:

$$W_p = 144(\frac{n}{n-1)} \times f \times P_1 V_1 \{(P_2 - P_1) \times \frac{n-1}{n} - 1\} \text{ ------------------------------------- (13)}$$

Where:
W_p = polytropic work
For ideal or perfect gases, the factor f is equal to 1. The polytropic efficiency is then calculated by:

$$\gamma_p = \frac{W_s}{W_p} \gamma_p \text{ -- (14)}$$

Where W_s = isentropic work
This polytropic efficiency will not agree with the value calculated using the GPSA method which is computed using γ_p = {(n-l)/n) / {(k-l)/k}.

GPSA Method
This GPSA method is the default method, and is more commonly used in the chemical process industry [9].
In this method, the adiabatic head is calculated from equations above. Once this is calculated, the isentropic coefficient k is computed by trial and error using:

$$HEAD_{ac} = \{(Z_1 + Z_2) / RT_1 / \{(k-1)\} \{\frac{P_2}{P_1})^{9k-1)/k} - 1\} \text{[39]------------------------(15)}$$

Where:
Z_1, Z_2 = compressibility factors at the inlet and outlet conditions
R = gas constant
T_1 = temperature at inlet conditions.

This trial and error method of computing k produces inaccurate results when the compression ratio, becomes low. If the calculated compression ratio is less than a value set by the user, the value of k has to be calculated.If k does not satisfy $1.0 \leq k \leq 1.66667$, the isentropic coefficient, is calculated by trial and error based on the following:

$$T_2 = (Z_1 / Z_2) \times T_1 \times [(P_2 / P_1)^{(k-1)/k} \text{ --- (16)}$$

The polytrophic compressor equation is given by:

$$HEAD_p = [(Z_1 + Z_2)/2] \times RT_1 / \{(n-1)/n\}] \{(P_2 / P_1)^{(n-1)/n} - 1\}$$

Where $Head_p$ is polytrophic head

The adiabatic head is related to the polytropic head by:

$$\frac{HEAD_{ad}}{\gamma_{ad}} = \frac{HEAD_p}{\gamma_p} \text{ --- (17)}$$

The polytrophic efficiency n is calculated by:

$\gamma_p = [n/(n-1)]/[k/(k-1)]\gamma_p$--- (18)

The polytropic coefficient, n, the polytropic efficiency γ_p, and the polytropic head are determined by trial and error method. The polytropic gas horsepower is then given by:

$$GHP_p = HEAD_p \times \frac{F}{33000}]$$ ---(19)

II. Material And Methods

The entropy data needed for these calculations are obtained from a number of entropy calculation methods. These include die Soave-Redlich-Kwong cubic equation of state, and the Curl-Pitzer correlation method. Thermodynamic systems may be used to generate entropy data. User-added subroutines may also be used to generate entropy data.Once the entropy data are generated, the condition of the outlet conditions from the compressor and the compressor power requirements are computed, using either a user-input adiabatic or polytrophic efficiency.

III. Results And Discussion

Table 1: Air Facility Reading for 75KI01

Time	**Air Inlet Temp.(^{0}c)**	**Oil Cooler L/D Temp (^{0}c)**	**Oil Cooler L/D Outlet (^{0}c)**	**After Cool Air Outlet Temp. (^{0}c)**	**Turbine Inlet Steam Temp. (^{0}c)**	**Turbine Inlet Steam Pressure (N/M^2)**	**Turbine Exhaust Pressure (N/M^2)**
8:00 Hr	32	52	30	34	400	41	17.1
9:00 Hr	30	52	32	34	400	41	17.2
10:00 Hr	32	53	31	34	400	42	17.3
11:00 Hr	30	53	31	34	400	42	16.9
12:00 Hr	33	52	32	34	400	42	17.0
13:00 Hr	32	51	32	34	400	41	17.2
14:00 Hr	33	51	32	34	400	41	17.3
15:00 Hr	33	51	30	34	400	41	17.2
16:00 Hr	33	52	32	34	400	42	17.2
17:00 Hr	33	52	31	34	400	42	17.3
18:00 Hr	33	52	32	34	400	42	17.3
Average	**32.4**	**51.9**	**31.4**	**34**	**400**	**41.5**	**17.2**

Compressor Heat Loss

Assumption: air is an ideal gas. Steady operating condition exists. There is pressure losses:

$$W_{comp,\,in} = \frac{nRT_1}{\eta_{comp}(n-1)} \times \left(\left(\frac{P_2}{P_1} \right)^{\frac{(n-1)}{n}} - 1 \right)$$ ----------------------------- (20)

$$W_{comp,\,in} = \frac{1.4 \times (0.287) \times 293}{0.8(1.4-1)} \times \left(\left(\frac{801}{101} \right)^{\frac{(0.4)}{1.4}} - 1 \right) = 296.9KJ/Kg$$

A = πD^2/4 =π (3 x 10^{-3}m^2)/4 = 7.069 x 10^{-6}m^2

Air leaking through the hole is determined to be

$$m_{air} = C_{discharge} \left(\frac{2}{K+1} \right)^{\frac{1}{(k-1)}} \times \left(\frac{P_{line}}{RT_{line}} \right) A \times \sqrt{KR\left(\frac{2}{K+1} \right) T_{line}}$$

$$m_{air} = 0.65 \left(\frac{2}{1.4+1} \right)^{\frac{1}{(1.4-1)}} \times \left(\frac{801}{0.287 \times 297} \right) 7.069 \times 10^{-6} \times \sqrt{1.4 \times 0.287 \left(\frac{2}{1.4+1} \right) 297}$$

= 0.008632kg/s

Then the power wasted by the leaking compressed air becomes

$$Power\ wasted = m_{air} \times W_{comp,in} \text{ -------------------------------------- } (21)$$

= 0.008632 x 296.9

= 2.563kW

The compressor operates for 4200 hours a year, and the motor efficiency is 0.92.
Then the annual energy and cost savings are:

Energy savings = $\dfrac{\text{power saved x operating hours}}{\eta_{motor}}$

$= \dfrac{2.563\text{kW x } 4200}{0.92}$

= 11,700kWh/yr

Cost savings = (energy savings) x (unit cost of energy)

= 11700 x ₦8.50k/wh

= ₦99450/yr

Compressor Simulation Results:

Table 2: compressor simulation

S/No	Simulated Parameters	Simulated Values
1.	Duty (kW)	5.0684e+02
2.	Polytropic Exponential	1.626
3.	Adiabatic Efficiency	71.72
4.	Adiabatic Head (m)	7586
5.	Isentropic Exponential	1.402
6.	Polytropic Efficiency	74.26
7.	Polytropic Head (m)	7852
8.	Polytropic Head Factor	1.000

Table 3: Compressor Rating Curves

S/NO.	Flow (ACT m^3/h)	Head (m)	Efficiency (%)
1.	7812	7680	69.20
2.	8388	7575	72.00
3.	8964	7841	72.48
4.	9504	7347	72.58
5.	1.008e+004	7153	73.08
6.	1.062e+004	6717	72.46
7.	1.120e+004	5858	69.39
8.	1.148e+004	4957	62.91

Table 4: Compressor Flow Limits

S/NO.	Flow Limits	
1.	Surge Curve:	Inactive
2.	Speed	Flow
3.	Stone Wall Curve:	Inactive
4.	Field Flow Rate (ACT_m^3/h)	8330
5.	Stone Wall Flow ---	
6.	Compressor Volume (m^3)	10.00
7.	Rotational inertia (kg/m^2)	6.000
8.	Radius of gyration (m)	0.2000
9.	Mass (kg)	150.0
10.	Friction loss factor (rad/min)	3.000

Table 5: Compressor inlet Conditions

	Overall
Vapour/Phase Fraction	1.0000
Temperature: (^{0}C)	70.00
Pressure: (kPa)	208.0
Molar Flow (kgmol/h)	607.7
Mass Flow (kg/h)	1.759e+004
Std Ideal LiqVol Flow (m^3/h)	20.00
Molar Enthalpy (kJ/kgmol)	1283
Mass Enthalpy (kJ/kg)	44.32
Molar Entropy (kJ/kgmol-^{0}C)	116.2
Mass Entropy (kJ/kg-^{0}C)	4.015

Heat Flow (kJ/h)	7.797e+005
Molar Density (kgmol/m^3)	7.295e-002
Mass Density (kg/m^3)	2.112
Std Ideal Liq Mass Density (kg/m^3)	879.6
Molar Heat Capacity (kJ/kgmol-^{0}C)	29.00
Mass Heat Capacity (kJ/kg-^{0}C)	1.002
Thermal Conductivity (W/m-K)	2.775e-002
Viscosity (cP)	2.093e-002
Molecular Weight	28.95
Z Factor	0.9994
Cp/Cv	1.406
Act. Vol. Flow (m3/h)	8330

Table 6: Compressor Outlet Conditions

	Overall
Vapour/Phase Fraction	1.0000
Temperature: (^{0}C)	172.8
Pressure: (kPa)	412.0
Molar Flow (kgmol/h)	607.7
Mass Flow (kg/h)	1.759e+004
Std Ideal LiqVol Flow (m^3/h)	20.00
Molar Enthalpy (kJ/kgmol)	4286
Mass Enthalpy (kJ/kg)	148.0
Molar Entropy (kJ/kgmol-^{0}C)	118.2
Mass Entropy (kJ/kg-^{0}C)	4.083
Heat Flow (kJ/h)	2.604e+006
Molar Density (kgmole/m^3)	0.1111
Mass Density (kg/m^3)	3.215
Std Ideal Liq Mass Density (kg/m^3)	879.6
Molar Heat Capacity (kJ/kgmol-^{0}C)	29.60
Mass Heat Capacity (kJ/kg-^{0}C)	1.023
Thermal Conductivity (W/m-K)	3.421e-002
Viscosity (cP)	2.524e-002
Molecular Weight	28.95
Z Factor	1.000
Cp/Cv	1.395
Act. Vol. Flow (m^3/h)	5471

Table 7: Compressor Sizing Input Details

Unit operation type: Compressor
Equipment type: Gas Compressor – Centrifugal Horizontal
Compressor sizing input Details

Operating capacity	17591.48kg/h
Operating adiabatic head	7585.53m
Operating polytropic head	7852.30m
Adiabatic efficiency	71.7188
Polytropic efficiency	74.2551
Operating gas power	506.84 kW
Capacity overdesign factor	1,1000
Head overdesign factor	1,1000

Table 8: Compressor Sizing Data Output

Unit operation type: Compressor
Equipment type: Gas Compressor – Centrifugal Horizontal
Compressor sizing input Details

Design capacity	19350.63kg/h
Design adiabatic head	8344.1m
Design polytropic head	8637.5m
Gas power	613.16kW
Mechanical losses	10.93kW
Design power	624.09kW
Driver power	900hp

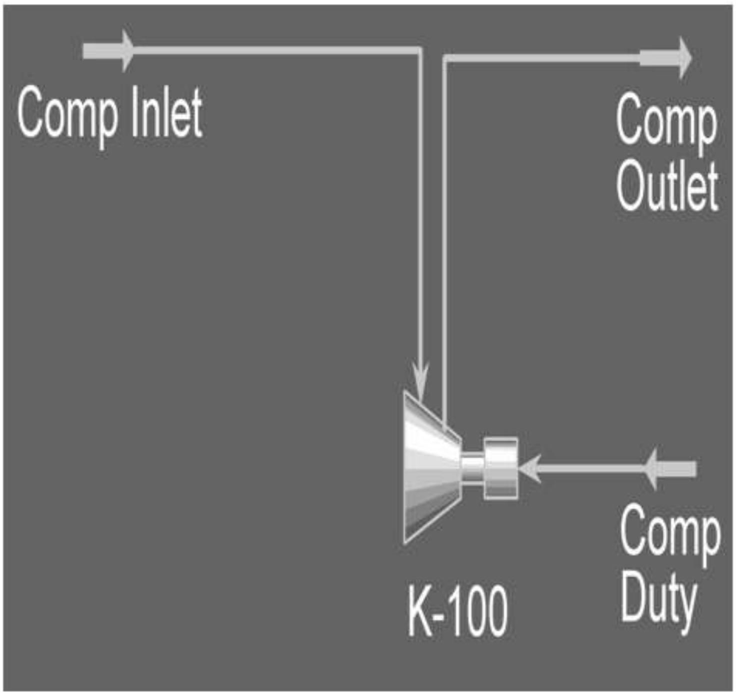

Figure .4: Compressor

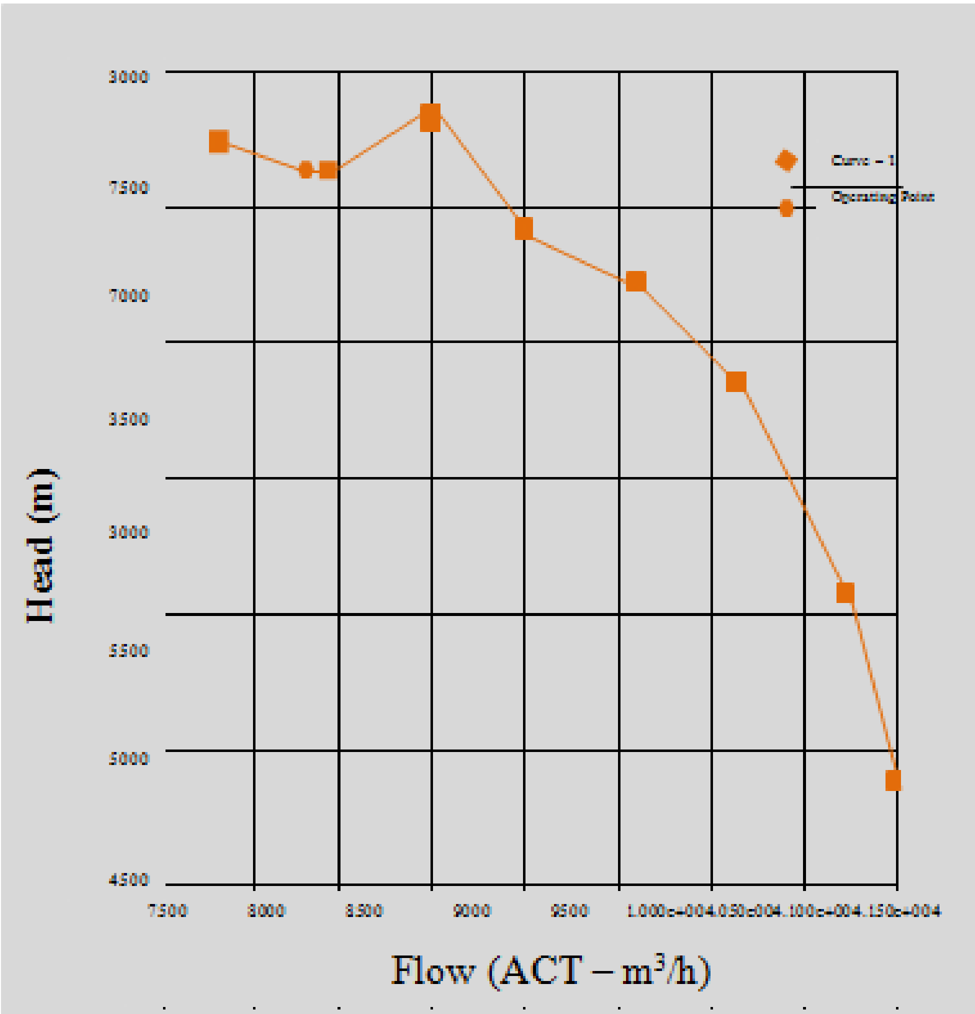

Figure 5: Head Curves of the Compressor

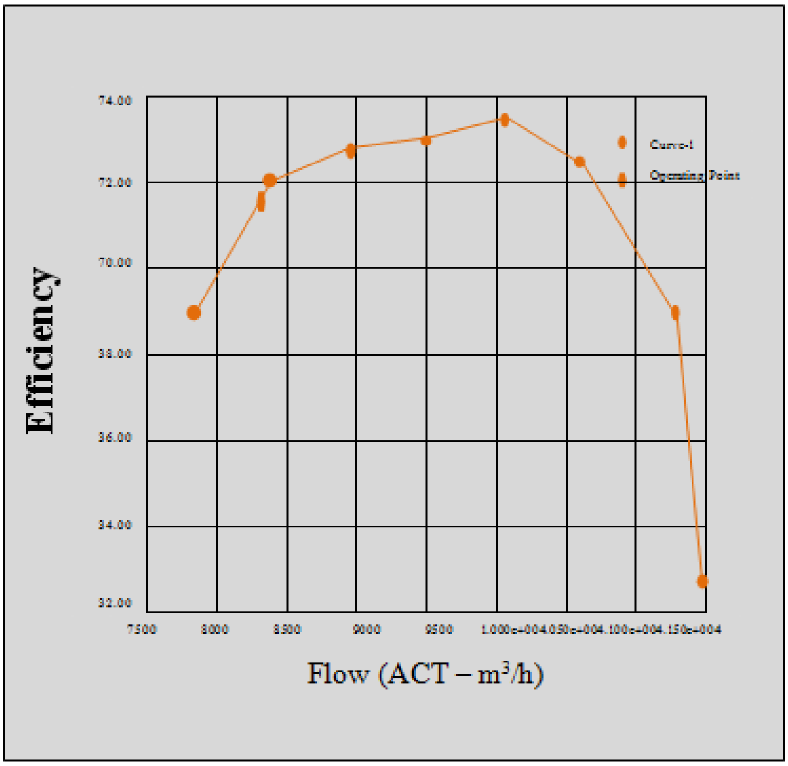

Fig. 6: Variation of Efficiency with Flow

IV. Discussion Of Results

Simulated Results

From figure 6 (efficiency curves of the compressor) the efficiency increases when the flow (m^3/h) is high and reduces when the level of the flow (m^3/h) decreases.

From table 2 the adiabatic efficiency was found to be 71.7% while the polytrophic efficiency was 74.3%.

Also from table 4 (compressor flow limits) the friction loss factor was found to be 3.00 rad/min.

From the head curves of the compressor (figure 5), the head decreases when the flow (m^3/h) increases.

The efficiency increases when the flow (m^3/h) increases as shown in figure 6.

V. Conclusion

This work set out to achieve the main goal of exploring ways for an effective energy saving which is expected to reduce energy cost, generate higher profit and increase capacity utilization. Energy loss which affects the output of the system was minimized. The empirical process heat loss, the actual values heat loss and the simulation model heat losss were found to be 2.56Kw, ,9.81kW and 10.93kW. The adiabatic efficiency and polytropic efficiency were found to be 71.7188% and 74.2551%. Moreover, the mechanical losses was found to be 10.93kW.

References

[1]. Emshoff, J. R (2001)."Design and use of computer simulation Models". New York: Macmillan,

[2]. Chilton, C. H.(2002)"Cost Engineering in the Process Industries". New York: McGraw – Hill,

[3]. Engeda, Abraham (1999). "From the Crystal Palace to the pump room".Mechanical Engineering.ASME.

[4]. Elliott Company. "Past, Present, Future, 1910-2010"(PDF). Elliott. Retrieved 1 May 2011.

[5]. API (2002). Std 673-2002 Centrifugal Fans for Petroleum, Chemical and Gas Industry Services. New York: API.

[6]. Whitfield, A. & Baines, N. C. (1990).Design of Radial Turbomachinery.Longman Scientific and Technical.ISBN 0-470-21667-0.

[7]. Aungier, Ronald H. (2000). Centrifugal Compressors, A Strategy for Aerodynamic Design and Analysis. ASME Press. ISBN 0-7918-0093-8.

[8]. Saravanamuttoo, H.I.H., Rogers, G.F.C. and Cohen, H. (2001).Gas Turbine Theory.Prentice-Hall.ISBN 0-13-015847-X.

[9]. Baines, Nicholas C. (2005). Fundamentals of Turbocharging. Concepts ETI .

[10]. . SAE/standards/power and propulsion/engines.SAE International.Retrieved 23 April 2011.

[11]. API (2002). Std 617-2002 Axial and Centrifugal Compressors and Expander-compressors for Petroleum, Chemical and Gas Industry Services. New York: API.

[12]. ASHRAE, American Society of Heating, Refrigeration and Air-Conditioning Engineers."Standards & Guidelines"ASHRAE.Retrieved 23 April 2011.

[13]. API (2007). Std 672-2007 Packaged, Integrally Geared Centrifugal Air Compressors for Petroleum, Chemical, and Gas Industry Services. New York: API.