Condition Monitoring of A Diesel Engine Turbocharger for Effective Performance Analysis

By

*** Ezenwa A. Ogbonnaya** ezenwaogbonnaya@yahoo.com Corresponding Author
*** Kombo T. Johnson** kombojohnson@yahoo.com
****H. U. Ugwu** badyke2001@yahoo.com.au
***** C.A.N. Johnson** Canjohnson2000@yahoo.com

* Department of Marine Engineering, Rivers State University of Science and Technology, Port Harcourt – Nigeria.
** Department of Mechanical Engineering, Michael Okpara University of Agriculture Umudike, Umuahia – Nigeria.
*** Department of Marine Engineering, Niger Delta University Wilberforce Island, Bayelsa State – Nigeria.

ABSTRACT

The diesel engine consists of several components which enhance its efficient and effective performance. One of these components is the turbocharger unit - compressor and turbine. It was found from this work that the overall efficiency of this unit depends mainly on the performance of the compressor and turbine components of the turbocharger unit. The test engine was an MTU 12V 396 TC 32 diesel engine. In this work,component modeling was adopted as the best means and synosure for achieving this aim of effective engine performance applying CM to arvert failure and engine downtime, thus increasing the useful life of the equipment. It was carried out to determine the turbocharger boost pressure. A software code named "TOP B" written in Quick Basic programming language was developed for the analysis. The data obtained shows boost pressure and exhaust temperature increased at the range of 0.1 bar and 10°C respectively. The results revealed that the turbocharger is able to develop enough boost to scavenge the cylinder of the exhaust gas as further. It also shows that the model developed is capable of predicting any blockage the air intake filter may experience. In view of these, it is highly recommemded that Condition-based maintenance philosophy should be used to increase its reliability and flexibility.

Keywords:
Component modeling, condition monitoring, pressure ratio and turbocharger unit.

NOMENCLATURE AND ABBREVIATIONS

PDF	=	Pressure drop across air intake filter (bar)
P_{02}/P_{01}	=	Boost pressure ratio
P_{01}	=	Air Pressure before entry into the turbocharger compressor nozzle
P_{02}	=	Air pressure at compressor outlet (bar)
η_{tc}	=	Overall turbocharger Efficiency (%)
η_m	=	Mechanical efficiency of turbocharger
η_c	=	Compressor efficiency (%)
η_T	=	Turbine efficiency (%)
T_{01}	=	Air Temperature before entry into the turbocharger compressor (°C)
T_{02}	=	Air temperature at the compressor outlet (°C)
T_{03}	=	Exhaust gas temperature before entry into the turbocharger turbine (°C)
P_{03}	=	Exhaust gas pressure from cylinders before entry into the turbine (bar).
γ	=	Ratio of specific heats
CM	=	Condition monitoring
M	=	Mass flow rate (kg/min)
M_g	=	Gas mass flow rate
M_a	=	Air mass flow rate
RCF	=	Filter Resistance Coefficient
AMF	=	Air Mass Flow, (Kg/min)
W_c	=	Compressor work (kJ/kg)
W_T	=	Turbine work (KJ/kg)
T_2^1	=	Isentropic compressor exit temperature (°C)
EGR	=	Exhaust Gas Recirculation
Q_1	=	Heat input (KJ/kg)
Q_2	=	Heat output (KJ/kg)
C_{Pg}	=	Specific heat capacity of the gas (KJ/kg)
C_{pa}	=	Specific heat capacity of the air (KJ/kg)
PE_1	=	Initial potential energy (KJ)
PE_2	=	Final potential energy (KJ)
KE_1	=	Initial kinetic energy (KJ)
KE_2	=	Final kinetic energy (KJ)

1. INTRODUCTION

Turbocharging is another method of supercharging an internal combustion engine. In turbocharging, the exhaust gas energy which is otherwise wasted is utilized to drive a turbine which in turn drives a compressor and thus increase the density of the charge air delivered into the engine (Bergman, *et al.*, 1993; Ogbonnaya, 1998; Nag, 2001 and Rai, 2006), The turbocharger is self contained and comprises basically of gas turbine and compressor fitted on a common shaft. It is stated in Eastop and McConkey (2005) that the attraction of turbocharging is that the energy lost in this way is used to drive a turbine wheel integrated with a compressor wheel which delivers charged air to the cylinder (Raven,et al, 2005). An explored view of a typical turbocharger showing the internal details is shown in Fig. 1.

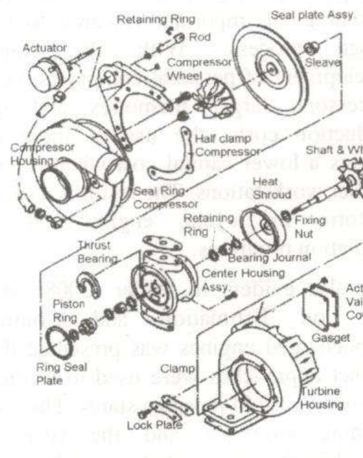

Fig. 1: An explored view of a turbocharger

The usual arrangement for a turbocharger is single-stage centrifugal compressor driven by a single-stage axial-flow turbine for the medium and large-size engines for industrial, railway and marine applications as shown in Fig. 2. Also this arrangement is applicable to a radial flow turbine for the smaller engine used in automotive applications, transport vehicles and cars (Eastop and McConkey, 2005). According to Luding and Sauer (2009), the design, construction, operation and maintenance of turbocharger is similar to what obtains for gas turbines. This therefore implies that the turbocharger of the diesel engine can be given the same considerations as the gas generator of a gas turbine.

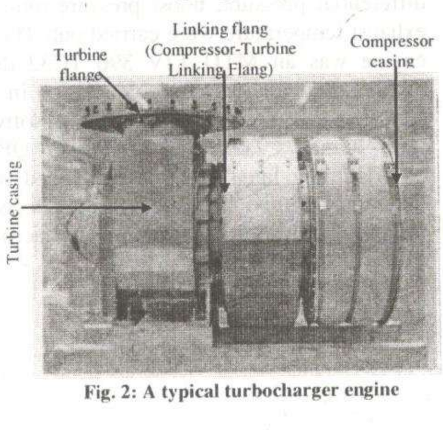

Fig. 2: A typical turbocharger engine

Consequently, failure and resulting downtime are always undesirable to the industry involved, because it leads to loss of production and costly repairs (Bell, 2003). However, Condition Monitoring (CM) is used to obtain early warning to prevent catastrophic downtime (Pussey, 2007; Ogbonnaya, 2009).

1.1 The Principle of Turbocharger Operation

The exhaust gases from the engine are directed to the turbine casing and expanded in the nozzle ring. The gases impart energy to the blades which are fixed to the turbine rotors and then passes via the outlet casing of the unit along an exhaust gas boiler. The charge air required by the engine is drawn through a suction pipe and filter to the compressor. It then flows through the diffusers and finally leaves the turbocharger via the scroll of the compressor (Ogbonnaya, 1998).

1.2 Methods of Turbocharging

In turbocharging an engine, the performance characteristics are chosen to suit the engine application. The attainment of higher engine efficiency, fuel economy and fast response in critical manoeuvring situation will have to be considered in selection of a method of turbocharging (Eastop and McConkey, 2005). The two systems commonly in the use are the constant pressure and pulse system of turbocharging.

1.2.1 Constant pressure system

It is stated in Eastop and McConkey (2005) that constant pressure is the simplest method of turbocharging system in terms of exhaust pipe arrangement and is easily matched to attain optimum efficiency. In this method, the exhaust gases from all the cylinders are collected in a common large exhaust manifold and fed to the turbine through a single entry. The available energy in the exhaust gases is converted to thermal energy by an enthalpy drop through the turbine.

The compressor and the turbine operate at near constant pressure or steady flow condition, because the fluctuating exhaust gases from the cylinders are damped to a steady flow in the manifold and at the turbine nozzle (Luding and Sauer, 2009). A high turbine efficiency will be achieved since the gases flowing through the turbine is full. The main advantage in this method manifests when there is a sudden increase in engine speed or load (Nag, 2001). Notably, when this occurs, the pressure in the large volume manifold will only rise slowly and the energy available at the turbine will also increase slowly giving rise to poor response of the turbine, compressor and the engine (Ogbonnaya, 1998).

1.2.2 Pulse system

This method utilizes the high kinetic energy of the exhaust gases. When the exhaust valve or port is opened, the exhaust gases cause pressure wave to be propagated in short exhaust pipes. These pipes ensure that most of the kinetic energy in the gases is maintained up to the turbine entry. The exhaust gas flows in a pulsating manner with high pressure fluctuation to the turbine nozzle (Eastop and McConkey, 2005). The kinetic energy of the gas is partially transmitted to the turbine with sonic velocity in the form of a pressure wave. Notably, the advantage of this method is that the energy available for conversion to useful work in the turbine is greater. Also, it has far superior acceleration response of the system when quick load changes occur because the small volume of the exhaust manifold results in rapid transfer of energy by pressure waves to turbine (Ogbonnaya, 1998). The limitation of this method is the poor performance that results when one or two cylinders only are connected to a turbine inlet, particularly if the pressure ratio is high. Also poor gas exchange occurs if the pressure waves arrive at the exhaust valve at the wrong time.

1.3 Effective Turbocharger Performance

The performance of the modern marine diesel engines is critically dependent upon the performance of the turbocharger. It influences to a very significant extent, engine power output, specific fuel consumption, thermal loading and the ability of the engine to burn low grade of fuel successfully (Amos, 2009). The author states that the current advancement in aerodynamic design and material technology are aimed at achieving the following benefits: high boost pressure ratio, increased overall efficiency, low specific fuel consumption at all operating loading, a realistic increase in engine power for a reduction in engine size. Others are mechanical reliability, low noise, a realistic cost, resistance to corrosion and erosion. He concludes that with a good air filtration system, noise level can be controlled while pressure drop is minimized.

1.4 Approaches to Monitoring and Data Collection

Continuous and periodic monitoring are used for turbine generator monitoring and analysis. The process of continuous monitoring does not eliminate the need for periodic monitoring. The continuous monitoring system warns the operator about imminent problems. Periodic monitoring along with the collection of external data provides, a means for analysis and projection of potential long-term problem both with respect to maintenance and operation. Gas generator model data collections are capable of acquiring the necessary information to monitor and trend the state of engine health (Ogbonnaya, 1998).

1.5 Other Approaches / Techniques

a) Mesbahi, *et a*, (2004), presented a draft on Diesel engine fault diagnosis by means of Artificial Neural Networks. They stated that computer models of diesel engines have been used predominatly in the last two decades, thus becoming an important research tool in the design process. With the continuous development of personal computers, i.e. faster processors, larger memories and reduced production costs, the use of these models implies a lower capital cost when running on smaller workstations. The result is a successful platform for cycle, engine and system simulation programs.

b) It is also evident in Miller (2008), where a draft on Estimation and Control of Turbocharged engines was presented that two distinct approaches were used to estimate the exhaust manifold pressure status. These are the dynamic approach and the steady state turbocharger energy balance approach. He stated that a turbocharged engine has barometric pressure, boost pressure and boost air temperature sensors. It was observed that in the dynamic approach, the model was found to give good accuracy for some data sets, but sufficient accuracy was not achievable for all available data sets. This approach was abandoned in favour of a steady-state turbocharger energy balance approach that has stood test of time.

c) According to Ogbonnaya (1998), the use of component model to generate component output variables for the healthy condition is shown in Fig.3. Analyses of air intake differential pressure, boost pressure ratio and exhaust temperature were carried out. The test engine was an MTU 12V 396 TC32 diesel engine, used for electricity generation in Port Harcourt, Rivers State of Nigeria. A software code named "TOP B" written in QBasic programming language was developed using the engine component models as it affects the turbocharger boost system. The acronym

"TOP B" stands for "Turbocharger Optimizer Boost System". This method of model-based computer program is faster in diagnosing faults, because it signals the limit of operation through instrumentation in the form of alarm.

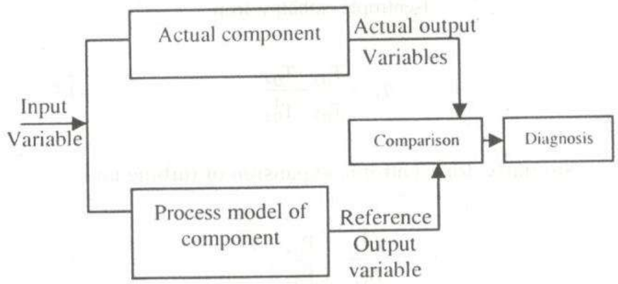

Fig. 3: Component model for monitoring and diagnosis

2. MATERIALS AND METHODS

The engine setup under investigation as Ogbonnaya (1998) reported is depicted in figure 3. The engine model is MTU 12V 396 TC 32 diesel engine. Also the engine characteristics are shown in Appendix A. The paper contributed solution to prevent the unexpected failure/reduction in performance of a turbocharger boost system by making its performance effective. This was achieved by comparing the field/operating data of thermodynamic/performance parameters to that found in the manufacturer's manual and adopt measures using CM to achieve the optimum performance. Pressure gauges were fitted on the downstream of the air intake filter. The pressure drop across the filter before entry into the compressor was monitored and compared with the load on the plant. The filter condition was proactively monitored using the model below.

$$PDF = RCF\,[\text{AMF}]^2 \qquad (1)$$

It was stated in the manufacturer's manual that the mass of air through the intake air filter is 169.2kg/min. This was used to determine the variable RCF, which together with PDF are measures of the health of the filter.

2.1 Turbocharger Performance Modeling

Modeling of a turbocharger is to present different characteristics of the unit by a set of mathematical equations, so that when properly programmed, they can provide computers with sufficient information to simulate the behaviour of the unit (Ogbonnaya, 1998; (Mesbahi, 2000; Mesbahi *et al.*, 2004). The main purpose of turbocharger CM is to detect any abnormalities in the behaviour of the unit and how it affects the overall performance of the diesel engine, make a comparison with a reference set of data which corresponds to the healthy turbocharger and identify the possible fault (Ogbonnaya, 1998).

Therefore, a reliable turbocharger CM must be able to automatically process the measured values, interpret the acquired data, make a logical analysis and arrive at a conclusion about where the fault could possibly be initiated. The CM is expected to:

i) Verify the state of health of the turbocharger.
ii) Predict developing fault
iii) Diagnose the cause and
iv) Make recommendation on operation and maintenance of the unit.

Fig. 4 represents the turbocharging system in an engine set up. Pressure, temperature and flow gauges were fixed on already provided spell out on the experimental test engine.

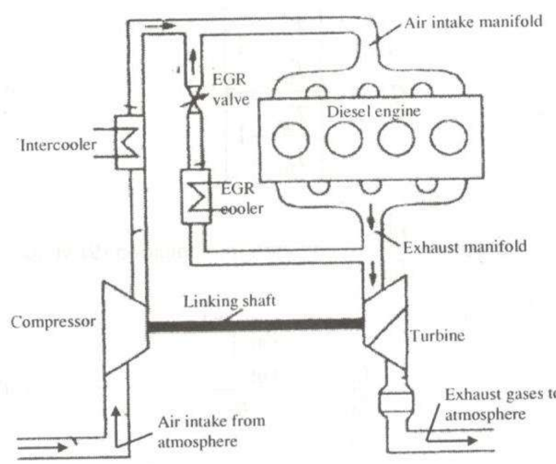

Fig. 4: Position of turbocharger in an engine setup

From steady flow energy equation

$$Q - W = m\,[(h_2+KE_2+PE_2)-(h_1+KE_1+PE_2)] \qquad (2)$$

By neglecting *KE*, *PE* and *Q*

$$-W = m\,[h_2 - h_1] \qquad (3)$$

But

$$dh = C_p dT \qquad (4)$$

By substituting (4) in (3) gives

$$-W = mC_p\,[T_{02} - T_{01}] \qquad (5)$$

The compressor work is given by

$$-W_c = m_a C_{pa} T_{01}\left[\frac{T_{02}}{T_{01}} - 1\right] \quad (6)$$

2.1.1 Efficiency of the compressor unit

$$\eta_c = \frac{\text{Isentropic enthalpy drop}}{\text{Actual enthalpy drop}}$$

$$\text{Thus } \eta_c = \frac{T^1_{02} - T_{01}}{T_{02} - T_{01}} \quad (7)$$

Also, Isentropic compression of the compressor was obtained as follows:

$$\frac{T^1_{02}}{T_{01}} = \left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}} \quad (8)$$

Dividing the numerator and denominator of equation (7) T_{01} gives:

$$\eta_c = \frac{\left[\frac{T^1_{02}}{T_{01}} - 1\right]}{\left[\frac{T_{02}}{T_{01}} - 1\right]} \quad (9)$$

Substitute $\frac{T^1_{02}}{T_{01}}$ of equation into equation (9) yields:

$$\frac{T_{02}}{T_{01}} - 1 = \frac{\left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\eta_c} \quad (10)$$

By substituting $\frac{T_{02}}{T_{01}} - 1$ of equation (10) into equation (6) gives

$$-W_c = m_a C_p T_{01}\left[\frac{\left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\eta_c}\right] \quad (11)$$

Hence equation (11) is the modified compressor work using the boost system

2.1.2 Efficiency of the turbine unit

$$W_T = Mg\, C_{p_g}\, [T_{03} - T_{04}] \quad (12)$$

By dividing with T_{03}, we have:

$$W_T = Mg\, C_{p_g}\, T_{03}\, [1 - \frac{T_{04}}{T_{03}}] \quad (13)$$

But Isentropic efficiency of turbine is given as:

$$\eta_T = \frac{\text{Actual enthalpy drop}}{\text{Isentropic enthalpy drop}}$$

$$\eta_T = \frac{T_{03} - T_{04}}{T_{03} - T^1_{04}} \quad (14)$$

Similarly, for Isentropic expansion of turbine unit

$$\frac{T^1_{04}}{T_{03}} = \left(\frac{P_{04}}{P_{03}}\right)^{\frac{\gamma-1}{\gamma}} \quad (15)$$

By dividing the numerator and denominator of equation (14) by T_3 yeilds:

$$\eta_T = \frac{\left[1 - \frac{T_{04}}{T_{03}}\right]}{\left[1 - \frac{T^1_{04}}{T_{03}}\right]} \quad (16)$$

$$\text{Hence} \quad 1 - \frac{T_{04}}{T_{03}} = \eta_T\left[1 - \frac{T^1_{04}}{T_{03}}\right] \quad (17)$$

By substituting equations (15) and (17) into (13) and rearranging becomes

$$W_T = m_g C_{p_g} T_{03} \eta_T \left[1 - \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}}\right] \quad (18)$$

Equation (18) is the modified turbine work using the boost system.

2.1.3 Overall efficiency of turbocharger for energy balance

$$-Wc = \eta_m W_T \quad (19)$$

Subsituting equations (11) and (18) into equations (19) yeilds:

$$m_a C_{p_g} T_{01} \frac{\left[\left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{\eta_c} = \eta_m m_g C_{P_g} T_{03} \eta_T \left[1 - \left(\frac{P_{04}}{P_{03}}\right)^{\frac{\gamma-1}{\gamma}}\right] \quad (20)$$

Further simplification of equation (20) yeilds:

$$\eta_m \eta_T \eta_c = \frac{m_a}{m_g} \cdot \frac{C_{P_a}}{C_{P_g}} \cdot \frac{T_{01}}{T_{03}} \left[\frac{\left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1}{1 - \left(\frac{P_{04}}{P_{03}}\right)^{\frac{\gamma-1}{\gamma}}} \right] \tag{21}$$

Where the product: $\eta_m \, \eta_T \, \eta_c = \eta_{tc}$, called the overall turbocharger efficiency.

Assuming $m_a = m_g$; $C_{pa} = C_{pg}$, then:

$$\eta_{tc} = \frac{T_{01}}{T_{03}} \left[\frac{\left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1}{1 - \left(\frac{P_{04}}{P_{03}}\right)^{\frac{\gamma-1}{\gamma}}} \right] \tag{22}$$

By simplifying further, the boost pressure ratio was obtained as:

$$\frac{P_{02}}{P_{01}} = \left\{ 1 + \eta_{tc} \frac{T_{03}}{T_{01}} \left[1 - \left(\frac{P_{04}}{P_{03}}\right)^{\frac{\gamma-1}{\gamma}} \right] \right\}^{\frac{\gamma}{\gamma-1}} \tag{23}$$

3. RESULTS AND DISCUSSIONS

The values of exhaust temperature, boost pressure and air intake differential pressure obtained from the CM session conducted for differential pressure drop, across the air intake filter Pdf are presented in Table (1).

The measured pdf was observed to remain at 0.85 bar for a very long time. Figure (5) shows the flow chart for "TOP B". While Table 2 as presented, was used to plot a combined graph of the variation of the boost pressure ratio and differential pressure versus exhaust temperature into the turbine nozzle as shown in figure (6).

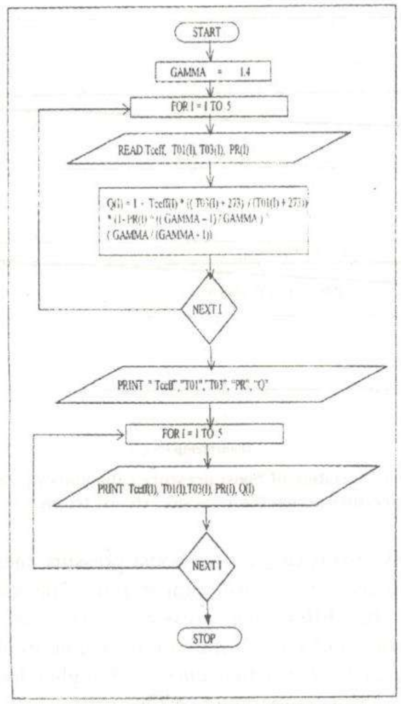

Fig. 5: Flowchart for "TOP B"

Table 2: Values of Exhaust Temperature, Boost Pressure and Air Intake Filter Differential Pressure.

T_3 (°C)	Q (bar)	Pdf (bar)	
exhaust	Boost	Cal.	actual
425	1.768	0.71	0.85
440	1.802	0.76	0.85
455	1.836	0.81	0.85
470	1.892	0.86	0.85
485	1.909	0.90	0.85

The trends of the calculated and measured pressure drops are almost same as depicted by the plot in figure 5. This shows that there is an approximate relationship between the pressure ratio and differential pressure as shown in the graph.

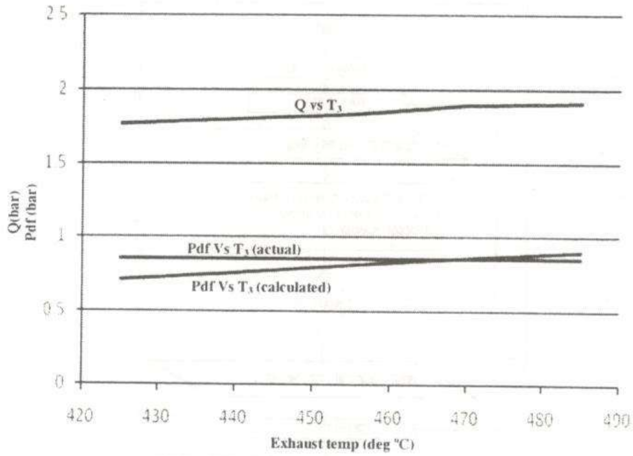

Fig.6: Variation of boost pressure ratio and differential pressure across filter versus exhaust temperature

The plots also revealed that boost pressure increases with increase in exhaust temperature. The same is true of the differential pressure across the filter though the latter takes a much longer time to change as depicted by the actual curve. At higher load (or higher exhaust temperature at the turbine nozzle), it was noticed that the compressor could develop enough boost irrespective of the state of the air intake filter. Finally, it was observed that the higher the η_{tc}, the greater the boost pressure ratio; the higher the expansion ratio across the turbine, the greater the boost pressure.

CONCLUSION

A diagnostic method of trend performance monitoring using component model which can proactively monitor the health of diesel engine turbocharger has evolved. The difficulties encountered in the analysis of data taken from air intake differential pressure, boost pressure ratio and exhaust temperature have been simplified. This was made possible using the software code named "TOP B", which enabled imminent failure to be detected.
The results obtained from the component models were programmed using QBasic programming language. It revealed that the approach is handy to trend performance monitoring of the turbocharger.
These models were validated using performance data obtained from an operational diesel engine. It was possible to experimentally establish that the differential pressure, boost pressure ratio and exhaust temperature are high enough to clear any blockage the filter may experience over a long period.

ACKNOWLEDGEMENTS:

The authors are highly grateful to the contributions of the following final year Marine Engineering students: **MR. EBUNUOHA, CHIGOZIE** and **MR. WOJI, JOHN NDUZINUAKACHI** for the effort they displayed during this work. They also worked tirelessly to supervise the typing of the manuscript which brought the work to this successful point. Moreso, we are indebted to the superintendent of the Nigerian Naval Ship Yard, Port Harcourt for allowing the use of their MTU engine.

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Table 1: Characteristics of an MTU 12V 396 TC 32 diesel engine

Indicated power	945kw
Mean indicated pressure	13.6 bar
Brake power	920kw
Brake mean effective pressure	23.1 bar
Engine speed	1800 rpm
Engine rated torque	1050Nm
Number of cycle	4