Applied Thermal Engineering 139 (2018) 334–340

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Research Paper

Lubricant thermo-viscosity effects on turbocharger performance at low engine load

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HIGHLIGHTS

- Two multi-grade API SN lubrication oils were tested at 50, 70 and 90 °C.
- The lubrication oil viscosity is an inverse nonlinear function of temperature.
- The hottest and thinnest oil gave highest compressor pressure ratio and efficiency.
- Turbocharger speed is directly dependent on turbine mass flow rate.
- The correlations of viscosity and turbocharger mechanical efficiency are proposed.

ARTICLE INFO

Keywords:
Compressor
Lubricant
Temperature
Turbine
Turbocharger
Viscosity

ABSTRACT

This work presents the performance and efficiency of a needle bearing type turbocharger under low load condition as affected by lubrication oil temperature and viscosity. The turbocharger on a test bench was propelled by known component exhaust gases with controllable mass flow and temperature at the turbine side. Two API SN multi-grade lubrication oils 0W-20 and 10W-40 were tested. The lubricant flow rates were kept constant by external supply at different inlet temperatures. The turbocharger efficiencies were determined by the measured inlet and outlet temperatures, pressures and flow rates for both turbine and compressor sides as well as the shaft rotational speeds. The experimental results were revealed that the higher inlet temperatures related to the thinner lubricant resulted in higher turbocharger efficiency. It suggests that the lubricant thermal manipulation is required for turbocharger under low load condition to operate within the design performance window.

1. Introduction

Turbocharger is currently gained more attention by vehicle designers and manufacturers as its key advantage is to enhance engine performance by increasing air mass density entering to the combustion chamber without expanding displaced volume. Additionally, some exhaust emissions from a typical internal combustion engine (ICE) have been found to reduce under certain operating conditions [1], particularly for diesel engines [2]. Furthermore, turbocharger has also been widely used in vehicles such as common rail fuel injection system diesel engines and gasoline direct injection (GDI) engines [3]. As only air and exhaust gas recirculation (EGR) are inducted into the combustion chamber and compressed with a trace of residual gas from the previous stroke without fuel presented [4], the engines can operate without limitation of spark knock [5], particularly for the GDI engines.

Thermal load is one of the factors affecting the turbocharger operating performance. As long as the engine is running, the hot exhaust gas must be persistently flowing through the turbine [6]. The resultant heat is then transferred to a shaft of the turbocharger that has to bear with a high rotational speed under high temperature. The shaft supported by bushings and bearings requires proper lubrication and cooling oil, flowing from the engine lubrication system and rinsing back into the oil pan [7]. Therefore, the exhaust gas temperature plays a vital role in determining a turbocharger output performance incorporated with lubrication oil.

Under low engine load where the exhaust gas temperature is at low temperature, less energy is transferred from the exhaust gas to the turbine blades, resulting in low turbocharger speed. Chun [8] compared lubricants in operation with and without bubbles under constant load at a speed of 20,000 rpm. It has been found that a friction from the oil with bubbles is higher due to the greater distribution of pressure. Pastor et al. [7] tried to cease the turbocharger lubrication system for few

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https://doi.org/10.1016/j.applthermaleng.2018.05.002
Received 9 February 2018; Received in revised form 3 April 2018; Accepted 1 May 2018
Available online 02 May 2018
1359-4311/

Nomenclature

A/R	diameter to radius ratio
BMEP	brake mean effective pressure (bar)
°C	celsius degree
$C_{p,e}$	specific heat at constant pressure of the gas flowing through turbine (J·kg^{-1}·K^{-1})
$C_{p,i}$	specific heat at constant pressure of the gas flowing through compressor (J·kg^{-1}·K^{-1})
EGR	exhaust gas recirculation
FS	full scale
GDI	gasoline direct injection
h	enthalpy (J·kg^{-1})
ICE	internal combustion engine
$\dot{m}_e$	mass flow rate of gas through the turbine (kg·s^{-1})
$\dot{m}_i$	mass flow rate of gas through the compressor (kg·s^{-1})
p	pressure (bar)
p_{02}/p_{01}	compressor pressure ratio
p_{03}/p_{04}	turbine pressure ratio
ppm	part per million
rpm	revolution per minute
rps	revolution per second
SAE	society of automotive engineers
T	temperature (K)
VDC	volt, direct current (V)
VGT	variable geometry turbocharger
VI	viscosity index
$\dot{W}_C$	power required to drive the compressor (W)
$\dot{W}_T$	power produced by the turbine (W)
γ	ratio of specific heat
η_{CTT}	compressor total-to-total isentropic efficiency (%)
η_m	turbocharger mechanical efficiency (%)
η_{TTT}	turbine total-to-total isentropic efficiency (%)
ν	kinematic viscosity (cSt)

seconds. By their trials, the shaft of the turbocharger was quickly worn out. The effects of temperature and pressure of the lubricating oil under no-heat-loss condition were studied [9]; the results have shown that the turbocharger efficiency was bound by 30% at a speed of 30,000 rpm. The outcomes of the test demonstrated the interaction between oil temperature and pressure on the performance of the compressor such as the compressor pressure ratio, power, compressor efficiency and mechanical efficiency of the turbocharger.

In terms of efficiency, Serrano et al. [10] studied the loss of mechanical energy through the turbocharger's performance analysis. During a high speed operation, the maximum mechanical efficiency was observed. In the other hand, the low rotational speed of the turbocharger resulted in higher mechanical losses. In the test of three turbochargers without inlet variable geometry vane at the entrance of the turbine [11], the efficiency of up to 80% at a speed of 43 rps can be obtained. Meanwhile, the variable geometry turbochargers yielded similar efficiencies: 77% and 79%, respectively for a single and double inlet channels at a speed of 28 rps during 60 degrees vane adjusted. Recently, a variable geometry turbocharger (VGT) have been gained more attention as it enhances transient operating performance of a vehicle [12]. However, the VGT is yet crucially required working at light engine load.

To enhance performance of a turbocharger, Chiong et al. [13] modeled a twin entry turbine turbocharger using one-dimensional prediction. In the simulation, the five presented models were predicted for unsteady turbine flow performance to varying degrees of accuracy. A turbocharger performance testing was described in [14], conforming to the Turbocharger Gas Stand Test Code (SAE J1826) that have to be in consideration for the parameters concerned. In addition, the turbocharger performance as influenced by heat transfer was evaluated [15]. The test data and theoretical analysis showed the significance of the heat transfer from the turbine to the compressor on the component performance maps on a small turbocharger.

Heat transfer is one of the main issues associated to the reduction in turbocharger performance and efficiency. The insulation of a turbocharger can prevent radiative heat transfer from hot surfaces and reduce convective heat transfer between turbocharger components and the test cell [14]. The SAE J1826 code specifies that the turbocharger should be insulated for testing. The heat losses from a turbocharger was analyzed under stationary and transient engine operating conditions using a lumped capacity heat transfer model [16]. The model was verified with a turbocharger operating on a 2.2L diesel engine under steady and transient conditions ranging from 1000 to 3000 rpm and 2 to 17 bar BMEP (brake mean effective pressure). The turbine outlet temperature prediction could be reduced by 29 °C maximum for steady state conditions. The modelling and experiment of the external heat losses in small turbochargers were accomplished and reported in [17]. Both radiative and convective mechanisms were presented for two different turbochargers; later on the model was validated against experimental measurements on an engine test bench. The behavior of small turbochargers is deeply affected by heat transfer phenomena in which external heat transfer in turbochargers cannot be neglected.

For a flow and loss interaction, the exhaust gas at high flow and high temperature can make the blades of the turbine and compressor spinning at revolution speed in some conditions beyond a hundred thousand rpm [18], due to combustion hot gas expansion and exhaust gas velocity. However, the turbocharger operating at high speed was prone to cause the greater extent of friction on the bearing shaft [19], and in the same time it caused the greater flow rate of the lubrication oil. The flow instability in a turbocharger ported shroud compressor can occur in certain circumstance [20]. Stability improvement due to the ported shroud was by removing swirling backflow from the impeller inducer tip and recirculating it into the impeller inlet. In a vaneless diffuser of a scaled-up turbocharger compressor, the flow at low speed operating conditions was simulated using the steady-state viscous calculations [21]. The simulation was mainly emphasized on fluid streamlines and the flow coefficient over the specimens.

In effects of lubrication oil viscosity – temperature relationships, there are some findings associated to turbocharger parts such as bearings, valve train, and lubrication system. The performance of plain journal bearings affected by lubricant viscosity – temperature characteristics was quantified [22]. The bearing temperature, power loss, and minimum oil thickness were found to increase with rotational speed of the turbocharger. A model, with an experimental verification, was developed by [23] to simulate the working conditions of journal bearings influenced by lubricating oil viscosity in vehicle turbocharged diesel engine crankpin journals. The analyzed parameters were: an oil temperature in the bearing, a change of the maximum value of the average pressure in the oil, the minimum carrying oil layer and the share of mixed friction, all in the function of oil viscosity at the bearing inlet, with different clearances between bearing and crankpin journal. In a valve train system, the friction losses were as a result of lubricating oil characteristics [24]. The experimental results accomplished in a tractor D-110 diesel engine showed a 0.16 kW difference between the total friction losses in valve train system using SAE10 and SAE40 class lubricating oils. The higher SAE viscosity class greatly reduced the friction of the valve train system but this negatively influenced the engine crankshaft lubrication. For a turbocharger lubrication system, a hydraulic accumulator and a braking device were used to reduce the thermal factor of turbocharger elements and the running-out durability [25]. The sole use of hydraulic accumulator in the lubrication system of the ICE turbocharger ensured the regular lubricating and cooling of the