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Turbocharger Oil Sealing Design and Capability

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ABSTRACT: *turbocharger improves efficiency by using exhaust gas energy that would otherwise be lost and plays a huge role in meeting current & future fuel economy, performance goals and due to effective combustion reduces environmental pollution. Due to different duty cycles, there are many failures and one of sever and most frequent is oil leakages from compressor and turbine side. Major failure modes of oil sealing leads to clogged air filter, gas leak in exhaust manifold to engine join, damaged turbo, obstruct turbo oil drain line, bearing housing coked & worn engine rings. It is important to keep oil away from the compressor and turbine rotor seal areas, because it can contribute to emissions. Oil sealing in turbocharger play very important role in meet anti-pollution legislation standard. Leakage being the major issue for engines & turbochargers, oil sealing is considered as the critical while designing sealing. These failures occur because of the lack of understanding of the oil leakage from turbocharger. In this paper, different seal design & fea parameters have been analyzed for predicting the behavior of the systems very accurately and efficiently. Study for understanding performance of oil sealing capability of turbocharger from different research papers which help to understand assumption made, analysis carried and different ways for oil sealing capability. Ian sherrington, neil grice, and john clifford jackson (1993)[7], did research & able to identify oil leakage at the turbine end of a turbocharger.*

Keywords*: Turbocharger, bearing, lubrication, oil seal, sealing system, rotor seal*

I. INTRODUCTION

Fig 1 shows oil flow path of turbocharger. The turbocharger shaft is supported by journal and thrust bearings. Bearings are lubricated with engine oil. Bearings are located within a central bearing housing connected between the turbine and compressor wheel housings. It is well known that providing an effective sealing system to prevent oil leakage from the central bearing housing into the compressor or turbine housing is problematical. This is particularly the case at the compressor end of the turbocharger since at low boost pressures there can be a significant drop in pressure from the bearing housing to the compressor housing which encourages oil leakage into the compressor housing. It is important to keep oil away from the compressor and turbine rotor seal areas, because it can contribute to emissions.

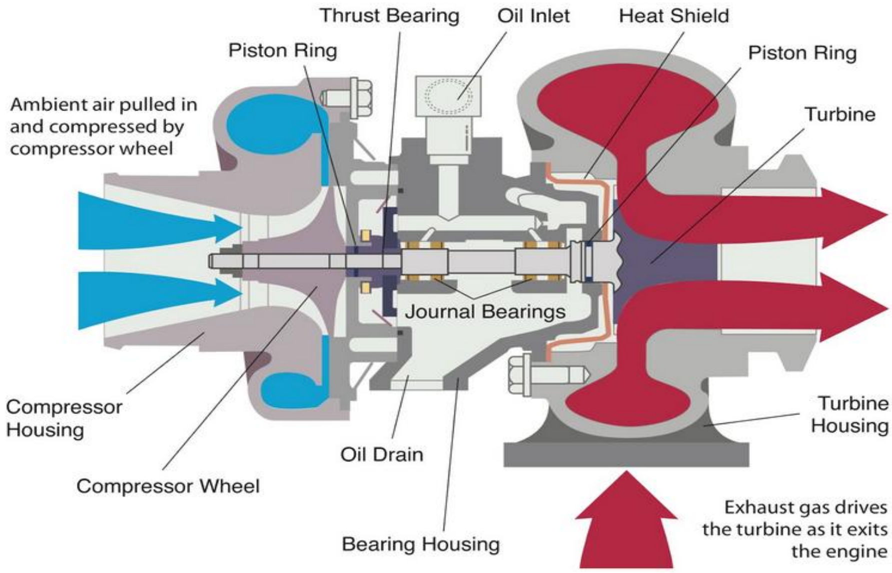


Fig 1: Components in Turbocharger and Oil Flow Path

In order to meet early anti-pollution legislation many heavy goods vehicle manufacturers have fitted turbochargers to vehicles and are now investigating how they can meet more exacting standards. It is apparent that although the use of turbochargers has a mostly beneficial effect on emissions, under certain operating conditions it is possible for the oil which lubricates them to leak past the seal which protects the clearance between the turbocharger shaft and its housing, contaminating the exhaust gases as a result. The seals at the compressor and turbine ends of the housing must not only keep oil from entering the compressor and turbine, but also must keep high pressure air and exhaust gases from entering the bearing housing. Oil is supplied to the bearing housing at the top through the oil inlet fitting and exits at the outlet connection. The oil drainage system must work even when the turbocharger is tilted on or about its axis.

Figure 2 shows piston ring position in turbocharger. The oil seal between the centre housing and the turbine housing & compressor cover is called a "piston-ring". It operates by virtue of the fact that the air pressure at the turbine end of the seal is normally greater than in the centre housing.

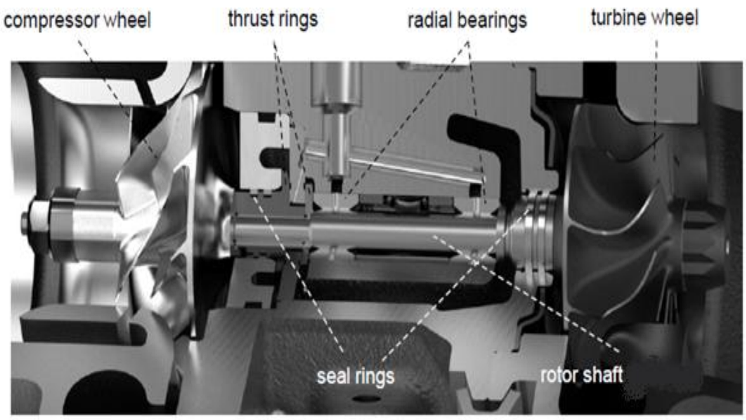


Fig 2: Seal Rings Position in Turbocharger

The oil seals at compressor and turbine ends are a difficult design problem, due to the need to keep frictional losses low, and to large movements due to large bearing clearance and adverse pressure gradients under some conditions. The compressor and turbine housings can usually be rotated, so that the compressor delivery and turbine inlet can be located in convenient positions. Fig 3 shows no oil leakage & oil leakage in turbine housing end oil sealing system.

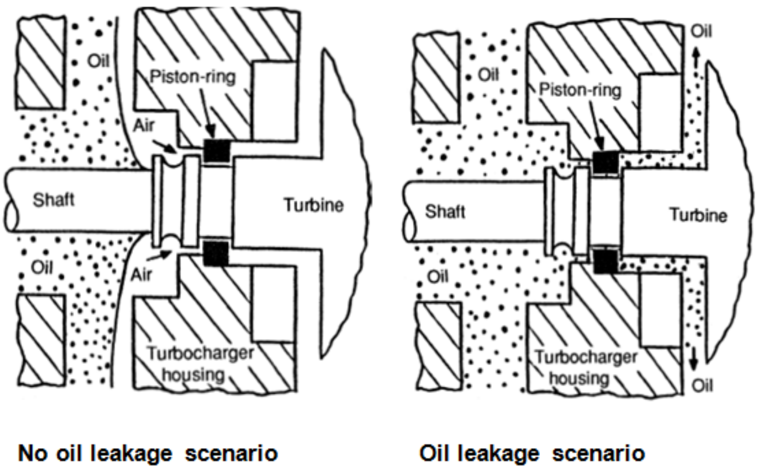


Fig 3: Turbine end no oil leakage & oil leakage scenario

One function of the sealing rings is to limit this flow of exhaust gas. The leakage gas is called “blow by” gas, and adds to the other gas leakages into the engine crankcase. This blow by gases mix with engine oil in the turbocharger or in the engine and are then are filtered and vented to the atmosphere, or can be routed to the compressor inlet with increasingly stringent exhaust emission regulations there are restrictions on venting such gases to atmosphere. There is a continuing need to improve the efficiency of the turbine & compressor end seal arrangement of turbocharger.

II. LITERATURE REVIEW

- There are various methods which have been used over time to eliminate turbocharger oil leakage (from the bearing housing to the turbine and compressor housings), as well as the elimination of exhaust gas leakage from the turbine housing into the bearing housing.
- The use of piston rings for sealing is the most common method used today
- The sealing on the turbine end is achieved relatively easy due to the positive pressure gradient which always occurs between the turbine and bearing housings
- The sealing on the compressor end is more complex than the sealing on the turbine end because of the low pressures that sometimes occur in the compressor housing
- Some of the reasons causing these low pressures are:

– Air slipstreams occur between compressor housing and compressor impeller back plate,
– Pressure drop occurring through the air filter
– The low pressure regime occurring at the compressor during non-boost periods, when the engine operates at low or idling speeds.

Authors	Methodology	Conclusion
Hong He, Siyou Xu, Ruiqian Yan and Jianbo Ji (2009)[3]	Hexahedral grid used in analysis	1. Side clearance is determining factor that affect leakage rate 2. Two piston ring reduced leakage rate
Tong Seop Kim, Kyu Sang Cha(2009)[4]	-k-e turbulence model	Leakage rate decreases with increase in numbers of cavities
Mber Dellis Polichronis, Retzios Evaggelos, Geralis Alcibiades, Gasparakis Elias, Pesiridis Apostolos,(2013)[5]	1. Dynamic oil seals which throws oil back to bearing housing 2. A number of holes drilled on oil thrower are used for oil separation from the gas 3. Use of graphite rings	1. Dynamic oil seal increase oil sealing turbocharger 2. Graphite rings improves oil sealing capacity but reduced performance due to increase in friction
Ian Sherrington, Neil Grice, and John Clifford Jackson(1993)[7]	Capacitance based on transducer is used to detect leakage through piston ring	Capacitance based on transducer is able to identify oil leakage at turbine end

Table 1: Summary of Literature review

Detection of Oil leakage: Fig 4 shows Oil leakage through Turbine housing & Compressor cover.

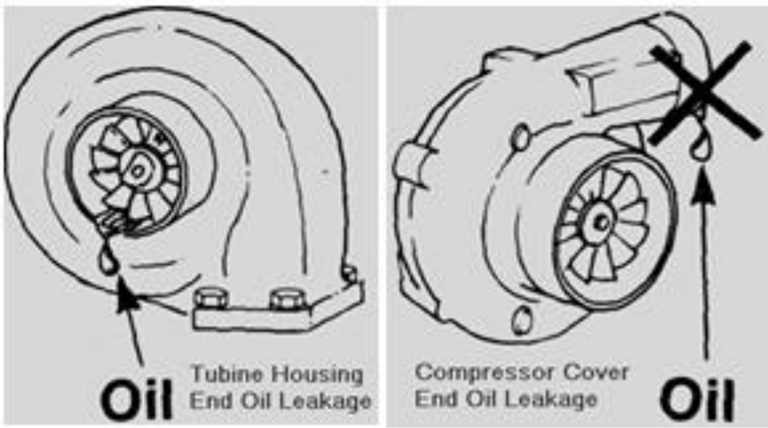


Fig 4: Leakage at Turbine housing & Compressor Cover

Ian Sherrington, Neil Grice, and John Clifford Jackson (1993)[7], did research on transducer which is able to identify oil leakage at the turbine end of a turbocharger. This has been developed for use in a bench test apparatus. Its principle of operation is based on that of the parallel plate capacitor which has been used extensively elsewhere to measure the thickness of thin lubricating films. The transducer has been successfully employed in this application to assess vacuum to leak conditions in turbocharger operating over a range of speeds on a bench test apparatus. Fig 5 shows Turbocharger Oil Leakage Detection Method.

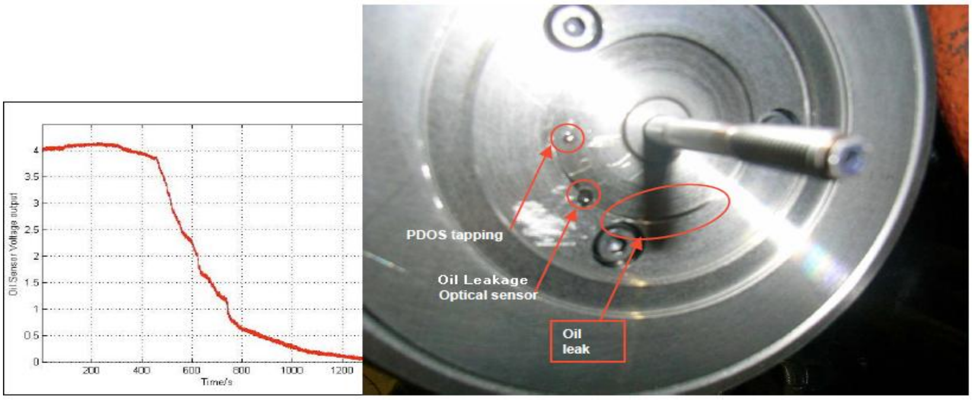


Fig 5: Turbocharger Oil Leakage Detection Method

Analysis of Turbocharger oil Leakage

Assumptions in CFD Analysis

- 3D geometry of oil passage near sealing area is only considered for analysis.
- Angularity/turbo tilt effect is considered as 0 degree turbo tilt angle.
- Oil flow is assumed as steady state flow.
- Temperature, density and viscosity are constant across the film thickness.
- Oil flow is assumed as incompressible flow with heat transfer (Total Energy equation).
- Effect of vibration on oil flow is not considered for CFD analysis.
- Pressure based coupled solver is used for analysis.
- Resizable k-e Turbulence Model is used for analysis.
- Turbulence Kinetic Energy & dissipation rate are set to first order upwind.
- Due to high shaft speed & mixing of oil & entered exhaust gases from turbine housing & or air from compressor cover, this results in oil frothing. Oil frothing can be modeled in CFD analysis with VOF (Volume of Fluid) Model. Due to complexity in analysis oil frothing is not considered in the analysis i.e. oil flow assumed without cavitation.
- Assumed oil inlet pressure at inlet region in cavity is same as that of pressure at turbocharge inlet. Effect of pressure change due journal bearing & mechanical friction is not considered.

Preparation of Geometry

Turbocharger assembly 3D model is opened in Creo. New part file in same assembly file is created. Piston seal ring surface & oil slinger surfaces are copied in newly created part file

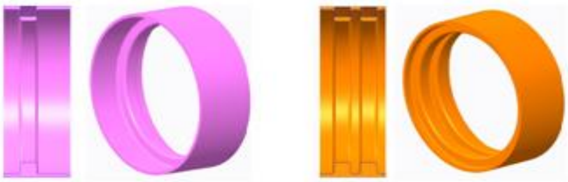


Fig 6: Oil cavity 3D geometry of Turbocharger under study

Meshing:

Oil cavity 3-D model is import to ICEM-CFD. Mesh is created at one section & revolved about the section. Fig 7 shows mesh setting for hybrid mesh Minimum orthogonal mesh quality for single piston seal cavity & two piston seal cavity is 0.65 & 0.58 respectively

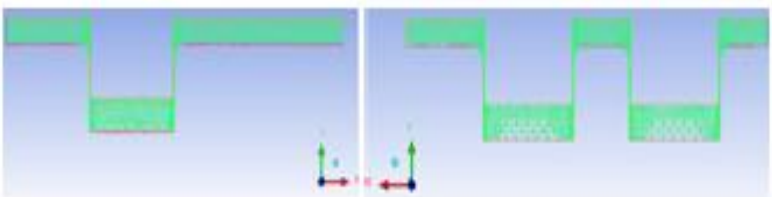


Fig 7: Mesh of one & two piston seal ring cavity

Oil domain material Properties

Engine oil SAE 15W40 is used in the turbocharger under study.

Oil temp	Oil Density	Kinematic viscosity	Dynamic Viscoisity	Thermal conductivity
Degree C	kg/m^3	Cst	kg/m.s	W/m-K
100	800	16	0.0128	0.6069
65	800	43	0.0344	0.6069
30	800	191	0.1528	0.6069

Table 2: Material Properties for Engine oil SAE 15W40

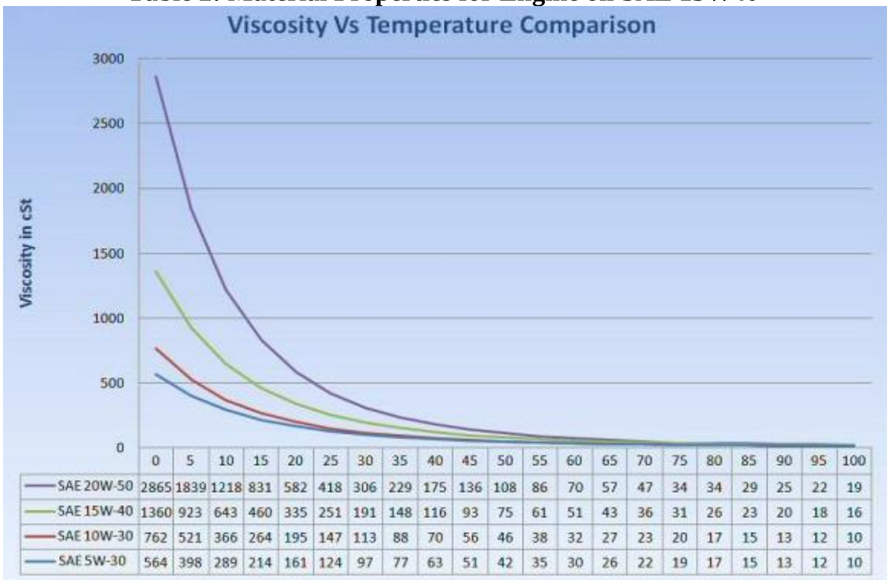


Fig 8: Viscosity Vs Temperature plot for engine oil

Seal Type	Iteration No.	Oil Inlet Temperature, °C	Oil Inlet Pressure, psi
Single piston seal split ring at 5000 rpm rotor speed	1	30	30
	2	65	30
	3	100	30
	4	30	45
	5	65	45
	6	100	45
	7	30	80
	8	65	80
	9	100	80
Two piston seal split ring at 5000 rpm rotor speed	10	30	30
	11	65	30
	12	100	30
	13	30	45
	14	65	45
	15	100	45
	16	30	80
	17	65	80
	18	100	80

Table 3- Oil Inlet Temperature & Oil Inlet Pressure setting at different Iterations

Solver: For Turbocharger oil leakage analysis k-e viscous Resizable Turbulence model is used. Solution Initialisation setting is kept to standard. Gauge pressure is kept to 0 psi & solution is initialised. iterations analysis is carried out

CFD Results: Fig 9 shows comparison of pressure contours for one seal ring design & two seal ring design

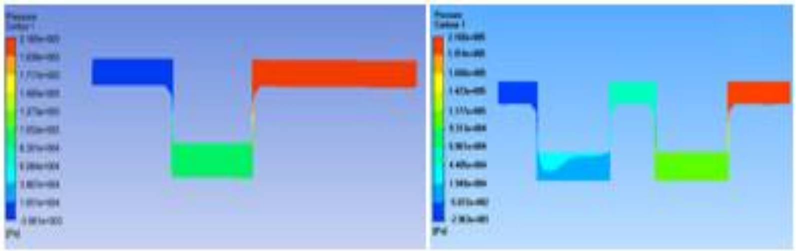


Fig 9 CFD results for 30 psi pressure at 30 degree C with Single & Double seal

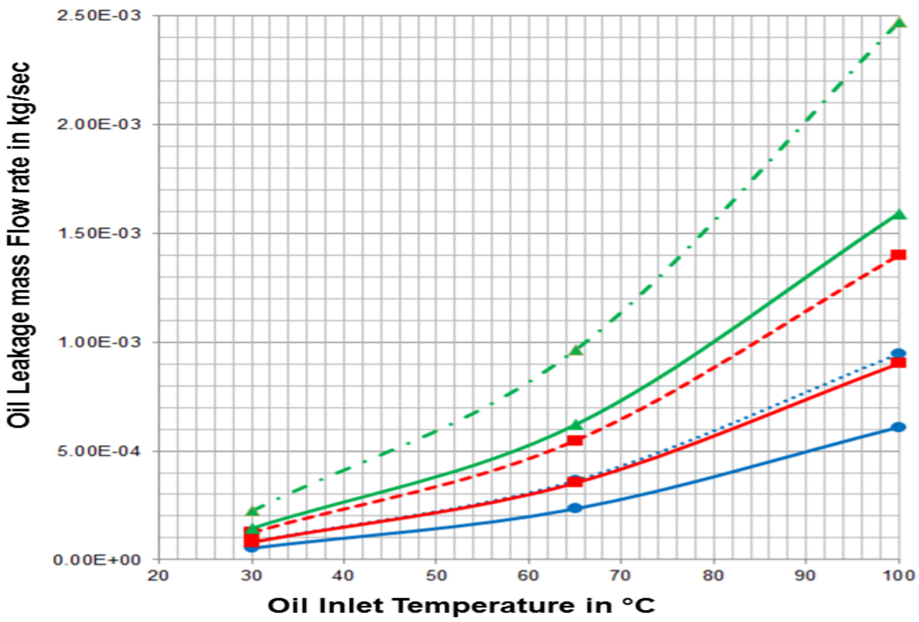
Similar analysis conducted for following combinations 45 psi & 65 deg & 80 psi & 100 deg also

Seal Type	Iteration No.	Oil Inlet Temperature, °C	Oil Inlet Pressure, psi	Oil leakage Mass flow rate, kg/sec
Single piston seal split ring at 5000 rpm speed	1	30	30	8.25E-05
	2	65	30	3.64E-04
	3	100	30	9.46E-04
	4	30	45	1.25E-04
	5	65	45	5.48E-04
	6	100	45	1.40E-03
	7	30	80	2.24E-04
	8	65	80	9.63E-04
	9	100	80	2.47E-03
Two piston seal split ring at 5000 rpm speed	10	30	30	5.31E-05
	11	65	30	2.34E-04
	12	100	30	6.09E-04
	13	30	45	8.05E-05
	14	65	45	3.53E-04
	15	100	45	9.02E-04
	16	30	80	1.44E-04
	17	65	80	6.20E-04
	18	100	80	1.59E-03

Table 4 shows oil leakage mass flow rate from CFD analysis for iteration 1 to 18

Summary of CFD results for oil leakage mass flow rate shows following trend,

- Two seal design reduces inlet pressure by 35 % when compared with one seal design when compared the result pressure contour.
- Oil leakage mass flow rate increases with increase in oil Inlet pressure at constant oil temperature.
- Oil leakage mass flow rate increases with increase in oil inlet temperature at constant oil inlet pressure
- Oil leakage rate reduced in two seal design by 31.6 % when compared with one seal ring design.



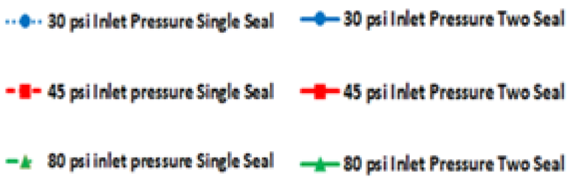


Fig 10 CFD result for oil leakage mass flow rate at different temperature & different pressure

Test Results

Table 4 shows oil sealing capability test results. Testing is completed at same shaft speed of 5000 rpm. Pressure difference of seal is maintained as 0 for all points.

In first part of testing single piston seal ring is tested at 9 test points. Each point is varied with change in oil inlet pressure & temperature. Results are useful to understand the current oil sealing capability of turbocharger. Similarly two design. Results are useful to understand improvement in oil sealing capability over current oil sealing capability. Also determine which design concept is more effective for improvement in oil sealing.

Results summary as follows

- Two piston seal ring shows improvement in oil sealing capability over one seal ring.
- Modified baffle shows more advantage on oil sealing compared to two seal rings design & one seal ring design.
- Modified baffle & Two piston ring combination can improve oil sealing capability
- At constant oil inlet pressure with change in oil inlet temperature, it shows leakage rate increases with increase in oil temperature but this is not true for all cases.
- At constant oil inlet temperature with change in oil inlet pressure, inconsistent trend seen for oil leakage.

Concept Description	Test Point No.	Oil Inlet Temperature, °C	Oil Inlet Pressure, psi	Test Result
Single piston seal split ring at 5000 rpm speed at 0 PDOS	1	30	30	Leakage
	2	65	30	No leakage
	3	100	30	Leakage
	4	30	45	Leakage
	5	65	45	Leakage
	6	100	45	No leakage
	7	30	80	No leakage
	8	65	80	Leakage
	9	100	80	No leakage
Two piston seal split ring concept at 5000 rpm speed at 0 PDOS	10	30	30	No leakage
	11	65	30	No leakage
	12	100	30	No leakage
	13	30	45	Leakage
	14	65	45	Leakage
	15	100	45	No leakage
	16	30	80	No leakage
	17	65	80	Leakage
	18	100	80	No leakage
Oil baffle modification concept at 5000 rpm speed at 0 PDOS	19	30	30	No leakage
	20	65	30	No leakage
	21	100	30	Leakage
	22	30	45	No leakage
	23	65	45	No leakage
	24	100	45	No leakage
	25	30	80	No leakage
	26	65	80	No leakage
	27	100	80	No leakage

Table 5 - Oil sealing capability test results

Comparison of Test results Vs CFD results

- As per CFD analysis two seal ring concept improves oil sealing capability at different oil inlet pressure & temperatures but at 3 cases test results shows leakage.
- As per CFD oil leakage mass flow rate increases with increase in temperature/pressure but test results shows inconsistent trend about this.

III. CONCLUSIONS

CFD analysis was performed to investigate improvement in oil sealing capability of compressor end of turbocharger. Two seal ring concepts is analysed through CFD analysis. CFD results are validated through the actual test using oil sealing capability test bench. Within scope & limitations of this project following conclusions can be drawn.

- Combination of two piston seal ring design concept & oil baffle modification can be potential solution to improve oil sealing capability of compressor end of the turbocharger.
- Turbocharger oil leakage is depending on oil inlet temperature & pressure. Oil leakage directly proportional to oil inlet pressure & temperature in addition to other operating parameters of turbocharger.
- Two piston seal ring design is structurally safe for implementation.

Future Scope

- Further study for formulation of two seal ring designs as well as modified oil baffle design with Multiphase turbulence model & correlate the test results with analytical CFD results.
- Formulate CFD analysis at different turbocharger tilt angle or angularity.
- Validate design improvement concepts through testing this on engine.
- Understand analytically & by test change in performance because of two seal split ring design.

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