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Efficiency Estimation of the Turbocharger Compressor Wheel

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Abstract— The paper is to discuss on turbocharger compressor wheel at different blade angles to find out the maximum efficiency of a turbocharger at inlet blade angle β_1 = 65^0,45^0 and 35^0 and studied the analysis of fluid flow phenomena over a compressor wheel of the turbocharger with the help of computational fluid dynamics (ANSYS-CFX).

Index Terms— Blade angles, Compressor map, Compressor wheel, Velocityof Fluid flow, Pressure, Efficiency,Turbocharger.

1 Introduction

THE turbocharger main components are turbine, impeller/compressor wheel, housing and the center housing/hub rotating assembly. The housing is fitted around the compressor and turbine wheels. The turbine is spin with the help of exhaust gases of the engine and which is drive the compressor wheel to develop high pressure of inlet atmospheric air to the engine. The compressor blade angles will give direct impact to develop high pressure.

2 Design and Analysis

The turbocharger compressor wheel is modeled (Table 1) with help of Pro-E for different blade angles where as analysis is done using ANSYS–CFX for different blade angles (65^0,45^0,35^0) for same geometry.

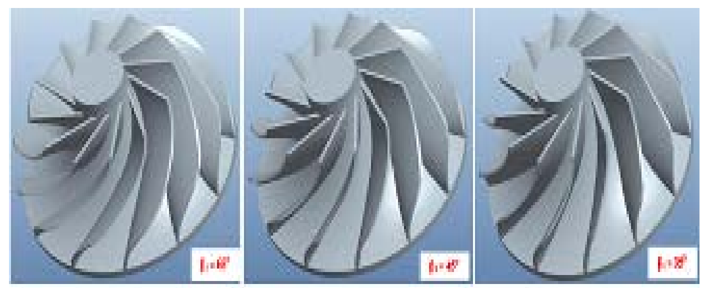

Fig.1. Turbocharger compressor wheel with different blade angle

TABLE 1

Turbocharger Compressor Wheel Specifications

S No.	Description	Dimensions
1	Inducer diameter (D_1)	90mm
2	Exducer diameter (D_2)	140mm
3	Trim	41.32
4	Tip height	20mm
5	Blade height (H)	58.38mm
6	No. of blades on impellor(N_b)	12
7	Thickness (t)	2mm

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2.1 Mesh Generation

The high-quality tetrahedral mesh is selected for inlet blade angle 65^0 (Nodes-24092 & Elements-108184), 45^0 (Nodes-23987 & Elements-108622) and 35^0 (Nodes-32661 & Elements-145299) and these meshes are used in the ANSYS-CFX to solve complex blade passage problems.

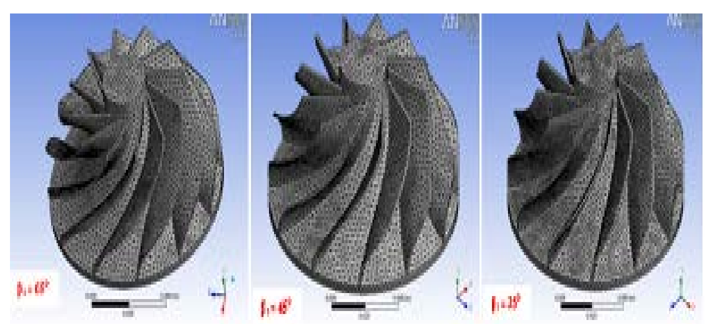

Fig.2. Mesh generation of turbocharger compressor wheel with different blade angle

2.2 Boundary Conditions

Boundary conditions are same for inlet blade angle 65^0,45^0 and 35^0 of a turbocharger compressor wheel.

- The working fluid is air at 25^0C and the pressure is 1 atm.
- The fluid domain is considered as stationary.
- Standard k-ε turbulence model with turbulence intensity of 5% is considered.

- The impellor domain is rotating and angular velocity is 15000 rpm.
- The speed of the fluid at inlet of the wheel is 150 kmph and the flow regime is subsonic.
- The wall boundaries are assumed as a smooth surface with adiabatic flow.

3 RESULTS AND DISCUSSION

The air (fluid) inlet velocity is 150km/h where as the turbo-charger compressor wheel spins at 15000 rpm for different inlet blade angles of 8.1 L engine at 7200 rpm. The analysis made based on velocity and pressure counters to estimate maximum efficiency.

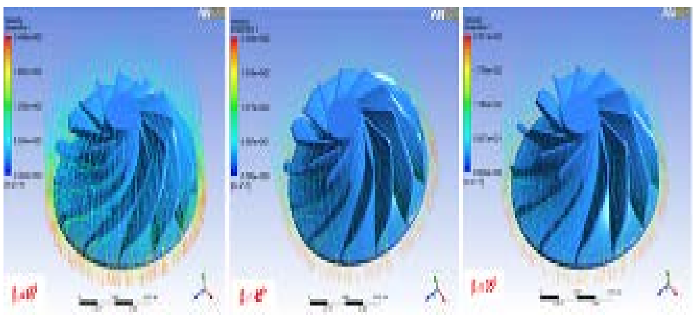

Fig.3. Velocity streamline of turbocharger compressor wheel with different blade angle

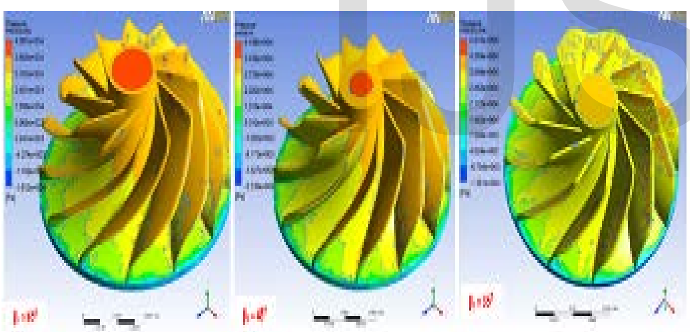

Fig.4.Pressure contour of turbocharger compressor wheel with different blade angle

3.1 Efficiency Estimation on a Compressor Map at β_1 = 65^0

Mass flow rate at outlet obtained from CFD calculations W_a = 0.982715 kg /s = (129.99 lb/min)

$$MAP_{req} = (W_a * R * (460+T_m)) / (VE * N/2 *Vd) \quad (1)$$

W_a = Airflow actual (lb/min or kg/s)
MAP = Manifold Absolute Pressure (psia or kpa).
R = Gas Constant = 639.6
T_m =Intake Manifold Temperature (^{0}F or ^{0}C) = 130 ^{0}F or 54.4 ^{0}C
VE = Volumetric Efficiency = 0.92
N = Engine speed (RPM) = 7200rpm
V = Engine displacement (8.1 liters * 61.02 = 494.262 CI or 8100CC)
[Conversion from Liters to Cubic Inches (CI) = No of Liters x 61.02]

$$MAP_{req} = (129.99 * 639.6 * (460 + 130)) / (0.92*(7200/2)*(8.1*61.02))$$

$$= 29.96 \text{ psia.} = 206.566 \text{ kpa.}$$

$$P_{2c} = MAP + \Delta P_{loss} \quad (2)$$

Where,
P_{2c} = Compressor Discharge Pressure (psia or kpa)
MAP = Manifold Absolute Pressure (psia or kpa)
ΔP_{loss} = Pressure Loss between the Compressor and the Manifold (psi or kpa)

P_{2c} = 29.96 + 2 = 31.96 psia = 220.3564 kpa.

$$P_{1c} = P_{amb} - \Delta P_{loss} \quad (3)$$

Where,
P_{1c} = Compressor Inlet Pressure (psia or kpa)
P_{amb} = Ambient Air pressure (psia or kpa).
ΔP_{loss} = Pressure Loss due to Air Filter/Piping (psi or kpa).

P_{1c} = 14.7 – 1 = 13.7 psia = 94.458174 kpa.

$$\text{Pressure ratio, } \pi_c = P_{2c} / P_{1c} \quad (4)$$

$$= 31.96 / 13.7 = 2.333$$

The efficiency η=80% is found (Fig 5) from GT6041 compressor map for 8.1 L engine at blade angle 65^0.

3.2 Efficiency Estimation on a Compressor Map at β_1 = 45^0

Mass flow rate at outlet obtained from CFD calculations W_a = 1.00756 kg/s = (133.2773 lb/min)

MAP_{req} = $(W_a * R * (460+T_m)) / (VE * N/2 *Vd)$

*MAP_{req} = (133.27737 * 639.6 * (460 + 130)) / (0.92*(7200/2)*(8.1*61.02))*

= 30.723 psia. = 211.830 kpa

P_{2c} = 30.723 + 2 = 32.723 psia = 225.6199 kpa.

P_{1c} = 14.7 – 1 = 13.7 psia = 94.458174 kpa.

Pressure ratio, $\pi_c = P_{2c} / P_{1c}$ = 32.723 / 13.7 = 2.388

The efficiency η=80% is found (Fig 5) from GT6041 compressor map for 8.1 L engine at blade angle 45^0.

3.3. Efficiency Estimation on a Compressor Map at β_1 =

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35^0

Mass flow rate at outlet obtained from CFD calculations

W_a = 0.98265 kg/s = (129.9835 lb/min)

$MAP_{req} = (W_a * R * (460+T_m)) / (VE * N/2 * Vd)$

$MAP_{req} = (129.9835 * 639.6 * (460 + 130)) / (0.92*(7200/2)*(8.1*61.02))$

$= 29.96$ psia. $= 206.566$ kpa

$P_{2c} = 29.96 + 2 = 31.96$ psia $= 220.3564$ kpa.

$P_{1c} = 14.7 - 1 = 13.7$ psia $= 94.458174$ kpa.

Pressure ratio, $\pi_c = P_{2c} / P_{1c} = 31.96 / 13.7 = 2.333$

The efficiency η=80% is found (Fig 5) from GT6041 compressor map for 8.1 L engine at blade angle 35^0.

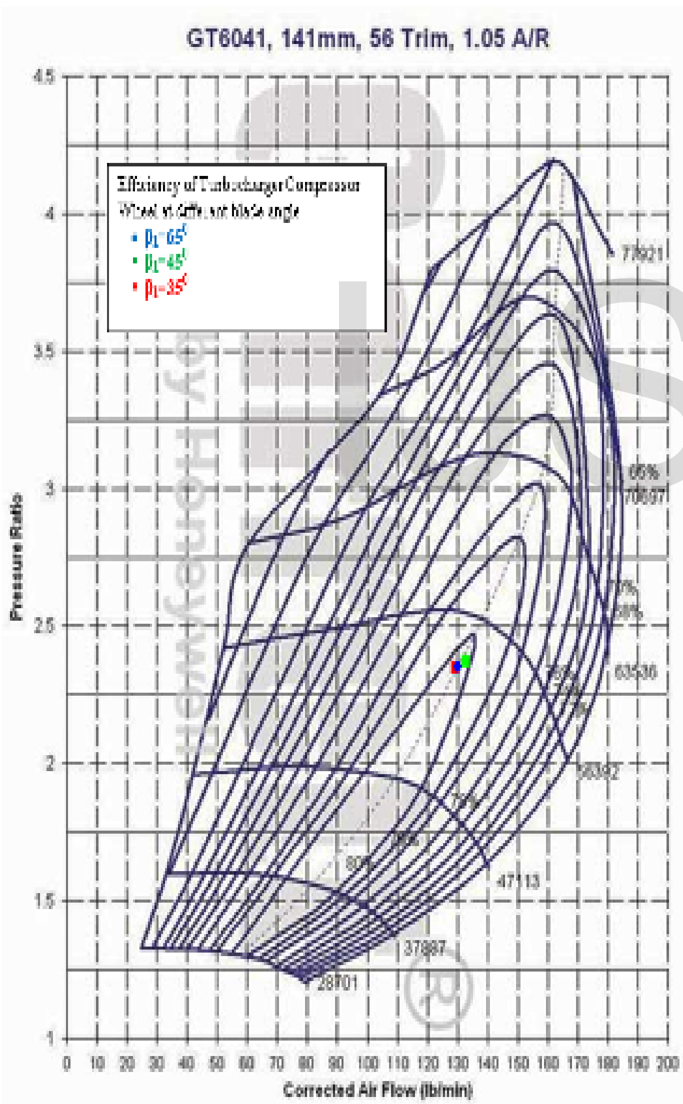
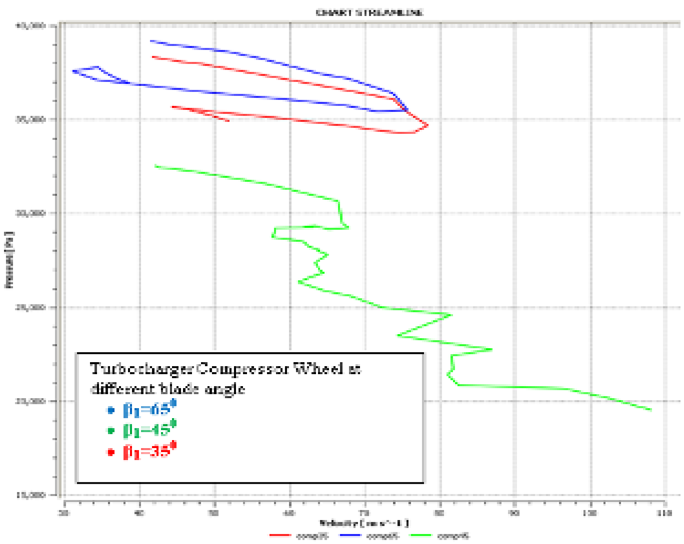


Fig.5.Compressor Map GT6041

At $\beta_1=65^0$, $\beta_1=45^0$ and $\beta_1=35^0$ the outlet mass flows (W_a = 129.99 lb/min, 133.2773 lb/min and 129.9835 lb/min) and pressure ratios (π_c = 2.33, 2.38 and 2.33) obtained and these values are marked on compressor map and found the efficiency η=80% for 8.1 L engine

Fig.6.Velocity versus Pressure for different blade angle



4 CONCLUSIONS

- It is observed that inlet blade angle $\beta_1 = 65^0$ is more efficient than $\beta_1 = 35^0$ and 45^0 it is observed from pressure contour.
- From velocity streamline it is observed that flow field in the blade passage is quit smooth in inlet blade angle, $\beta_1 = 65^0$ than other blade angles.

i.)It is observed at inlet blade angle $\beta_1 = 35^0$ blade passage the flow field is smooth but the developing pressure is low as compared to $\beta_1 = 65^0$ (Fig 6) this may not be efficient.

ii.) It is found that at inlet blade angle $\beta_1 = 45^0$ fluid particles in the flow field are mixing with subsequent blade passages that can be observe in velocity streamline and due to this pressure and velocity is fluctuating (Fig 6).

- Therefore it is recommended that inlet blade angle $\beta_1 = 65^0$ is more efficient than the other blade angles.

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