Impact of Mixed Flow Turbines on the Efficiency of Automotive Turbocharger Applications

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Abstract— **Automotive turbocharger applications are built on the basis of radial flow or axial flow in order to extract energy. The ever growing demands for the enhanced fuel efficiency and low cost of fuel used in passenger cars, part load and low-end performance are the crucial areas. These areas should be focused more while designing the automotive turbocharger applications and turbines. A turbine that can mine more energy at less rotational speed and greater pressure ratios is considered more efficient in context of automotive drive phase. Reviewing various studies, it has been found that efficiency of radial flow lies at 0.7 speed ratios, but in the meantime, the less rotational speeds and greater pressure ratios lead towards the low blade speed ratio. Focusing on such findings, mixed flow turbines are considered a best option that offers significant benefits for the automotive turbocharger applications. In order to prove this consideration, the presented research was based on experimental valuation of blade speed ratio and efficiency of mixed flow turbines. For this purpose, semi-unsteady approach was used in addition to providing with the evidences of advantages of mixed flow behavior in turbocharger applications.**

Index Terms— **Axial flow turbines, Radial flow turbines, Mixed flow turbines, Blade speed ratio, rotational speed, turbocharger**

I. INTRODUCTION

Turbochargers, originally known as super turbochargers, are one of the most common means of forced induction used in Internal Combustion engines. Initially, the turbochargers had been used only with diesel internal combustion engines but later they became common in internal combustions engines running other fuels like petrol and biofuel also [1]. Among three main components of the turbocharger, the compressor turbine compresses the incoming air and sends it to the engine and the exhaust turbine extracts energy from the high-pressure gas coming from the combustion chamber and sends this energy to the compressor turbine which uses it to compress the air [2]. The compressed air contains more oxygen, eventually more energy being released during combustion in the engine resulting in a higher power output. In this regard, turbochargers are the most preferred form of forced induction because they are simple, have high efficiency and do not require any additional power from the engine to perform its specified functions.

According to Bell (1997), a turbocharger is an air pump driven by energy remaining in the exhaust gases as they exit the engine [3]. One-third of the energy, released in the combustion process, goes into the cooling system, one-third becomes power down of the crankshaft and one-third is dumped out the tailpipe as heat. Assuming the engine produces 200 bhp, approximately 70 bhp of raw heat is let out through the tailpipe completely unused; it has been assumed that the amount of wasted energy increases with increase in engine size [4]. However, the turbocharger consists of parts necessary to extract the energy from the exhaust gas and use it to compress incoming air and hence supply more oxygen to the engine. Today the design of turbochargers has advanced more and now there are various types like variable geometry turbocharger, variable vane turbocharger, waste-gate turbocharger, and power assist turbocharger, variable nozzle turbocharger but research on optimizing the design of compressor and exhaust turbine impellors has been given much attention recently [5]. Turbochargers can increase the pressure of the incoming air by up to 10-15 psi depending on the turbocharger size and its setup [6].

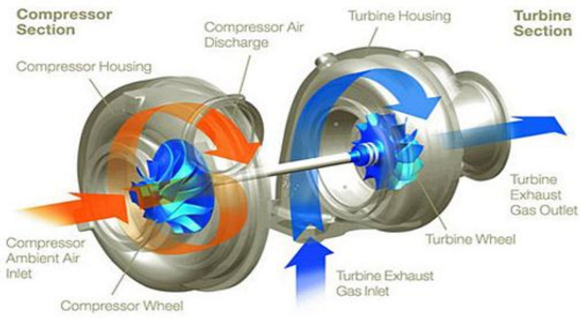

Figure 1. A turbocharger.

Manuscript received June 11, 2019; revised May 12, 2020.

doi: 10.18178/ijmerr.9.6.791-796

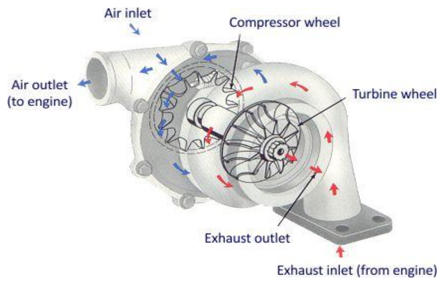

Figure 2. Flow of gasses in a Turbocharger.

The established automotive turbochargers were designed on the concept of radial flow turbines integrating with the centrifugal compressor [7]. In context of turbine flow, the radial concept is superior to the axial flow, more specifically for the turbines designed and deployed in various automotive applications [8] [9]. This is due to the fact that axial flow turbines face rotor clearance losses in higher ratio. Various studies depict the fact that radial design is more feasible as compared to axial design. However, looking for better approach and design, it has been found that efficiency at smaller blade speed ratio and larger pressure ratios is achieved by mixed flow turbine [10] [11]. In a nutshell, it can be stated that less axial flow leads towards radial flow and less radial flow gives mixed flow turbine as output to attain high efficiency when it is about automotive turbocharger applications [12].

Based on the aforementioned assumptions and surveys on turbochargers, the current research focuses primarily on the turbochargers used in diesel engines. After an analysis of working and technical aspects of turbochargers, the study is intended to optimize and improve their design. For this purpose, a model was proposed, based on general guidelines; the intention was to analyze the model in Computational Fluid Dynamics (CFD) software using the findings to develop better understanding of the turbocharger.

II. Components of a Turbocharger

There are various components of a turbocharger such as compressor, turbine, control system, bearing system and sealing; amongst them two of the most important are under discussion:

Compressor: It includes the impeller, the diffuser and the housing. Atmospheric air is sucked towards the impeller axially which supplies energy to it and increases its velocity, pressure and temperature. The air exits the impeller radially. The shape of the housing directs the compressed air towards the diffuser which is designed such that it decreases the air’s velocity and imparts more pressure to it so as to counteract for the pressure loss up to the combustion chamber.

Turbine: It is the combination of turbine wheel and the turbine housing. The turbine wheel is the impeller that absorbs the kinetic energy of the incoming exhaust gases coming from the combustion chamber. The extraction of energy from the exhaust gases decreases their pressure and temperature.

The compressor and turbine wheels have the greatest influence on the turbocharger's operational characteristics. These wheels are designed by means of computer programs which allow a three-dimensional calculation of the air and exhaust gas flows. The wheel strength is at the same time optimized by means of the finite-element method (FEM), and durability calculated on basis of realistic driving cycles [13]. Despite today's advanced computer technology, detailed calculation programs, it’s testing which finally decides the quality of new aerodynamic components. The fine alteration, checking of results is therefore carried out on turbocharger test stands.

Total-to-static turbine efficiency is defined as:

$$\eta_{ts} = \frac{h_{3t} - h_{4t}}{h_{3t} - h_{4s,\,is}} = \frac{\Delta h_{tt,\,stage}}{\Delta h_{ts,\,stage,\,is}} \qquad (1)$$

Where h= Enthalpy (J/(kg K)) and Δh: Enthalpy difference (J/kg).

III. Turbocharging in Diesel Engines

In diesel engines turbocharging is highly preferable because diesel engine blocks are generally made more robust than their petrol counterparts to handle the more explosive diesel. This goes hand in hand with the purpose for which the turbocharger was designed. With the passage of time, turbocharging has become common in truck diesel engines and modern cars in order to generate reduced emission levels, increased and improved power outputs and enhanced efficiency considering the equal capacity of the engine [14] [15]. It is commonly termed as turbo-diesel which is pointing towards any diesel engine with a turbocharger equipped in it. Since diesel engines run better on lean mixtures, turbochargers compliment that with more compressed air being sent into the combustion chamber for the amount of fuel supplied. They also help decrease the combustion noise, although mechanical noise does go up thanks to an increase in component load. Thus, the overall efficiency is now improved with reduced noise factor. For better understanding, pictorial representation of different turbo-diesels is given below:

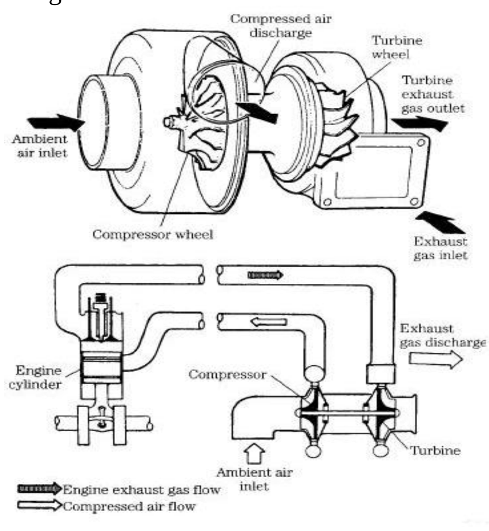

Figure 3. Diesel engine turbocharger

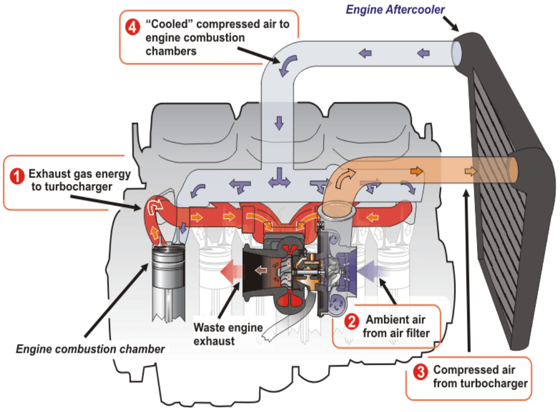

Figure 4. Diesel turbocharged engine system

IV. Turbomachinery Fundamentals

The figure given below (Fig. 5) shows the velocity triangles at the turbine wheel inlet and exit. The normal velocity triangle (including inlet swirl) at the impeller inlet is shown. Velocity triangles at the turbine exit with and without swirl are also shown [16]. The corresponding inlet diameter of a mixed flow turbine is expressed by the equation given below. This value is also used for circumferential velocity calculation of the mixed flow turbine wheel.

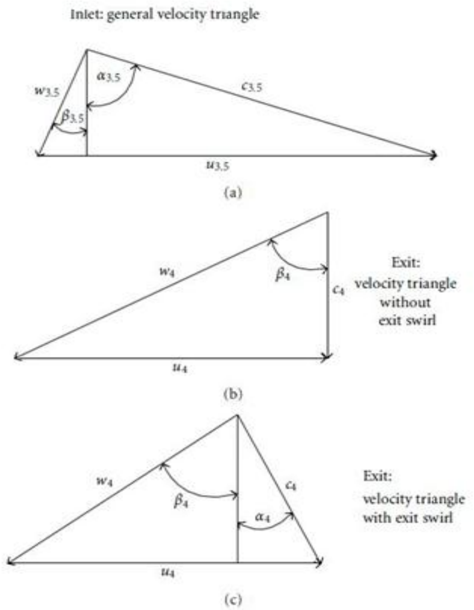

Figure 5. Velocity Triangles at the turbine wheel inlet and exit

$$D_{3,5} = \sqrt{(D^2{}_{3.5h} + D^2{}_{3.5s})/2} \tag{2}$$

Furthermore, the isentropic spouting velocity, which could be achieved if the available total-to-static enthalpy drop would be converted into kinetic energy by an isentropic process, is expressed as:

$$\frac{Cs^2}{2} = \Delta h_{ts,stage,is} \tag{3}$$

The blade loading factor is:

$$\psi = \frac{P_T}{m_T u^2{}_{3.5}} = \frac{\Delta h_{tt,stage}}{u^2{}_{3.5}} = \frac{C_{\phi 3.5} u_{3.5}}{u^2{}_{3.5}} - \frac{C\phi_4 u_4}{u^2{}_{3.5}} \tag{4}$$

The relationship between the real blade loading factor and isentropic blade loading factor is derived by:

$$\Psi = \frac{\Delta h_{tt,stage}}{u^2{}_{3,5}} \frac{\Delta h_{ts,stage,is}}{\Delta h_{tt,stage,is}} \tag{5}$$

$$= \frac{\Delta h_{ts,stage,is}}{u^2{}_{3,5}} - \frac{\Delta h_{tt,stage}}{\Delta h_{ts,stage,is}} = \psi_s \eta_{T,ts} \tag{6}$$

If the circumferential blade velocity at rotor inlet is comparably higher than the velocity at rotor exit, and the exit swirl of a single radial/mixed flow is small, the last term in above equation can be neglected, resulting in:

$$\psi \approx \psi^* = \frac{C_{\phi 3.5}}{u_{3.5}} \tag{7}$$

$$C_{\phi 3.5} = u_{3.5} \tan(\beta_{3.5}) \tag{8}$$

The relationship between velocity triangle at rotor inlet and stage loading is:

$$\eta_{T,ts} = \frac{\Delta h_{tt,stage}}{\Delta h_{ts,stage,is}} = \frac{\Psi u^2{}_{3,5}}{Cs^2/2} = 2\Psi\left(\frac{u_{3.5}}{Cs}\right)^2 \tag{9}$$

Therefore the total static turbine efficiency can be derived as:

$$\eta_{ts} = \frac{h_{3t} - h_{4t}}{h_{3t} - h_{4s,is}} = \frac{\Delta h_{tt,stage}}{\Delta h_{ts,stage,is}} \tag{10}$$

Where c= Velocity in stationary frame (m/s), u=Blade speed (m/s), Δh=Enthalpy difference (J/kg), C=Compressor, $u3.5/cs$=Blade speed ratio, h=Enthalpy (J/(kg K)), D=Diameter, s= Entropy (J/(kg K)) and cs=Isentropic spouting velocity (m/s).

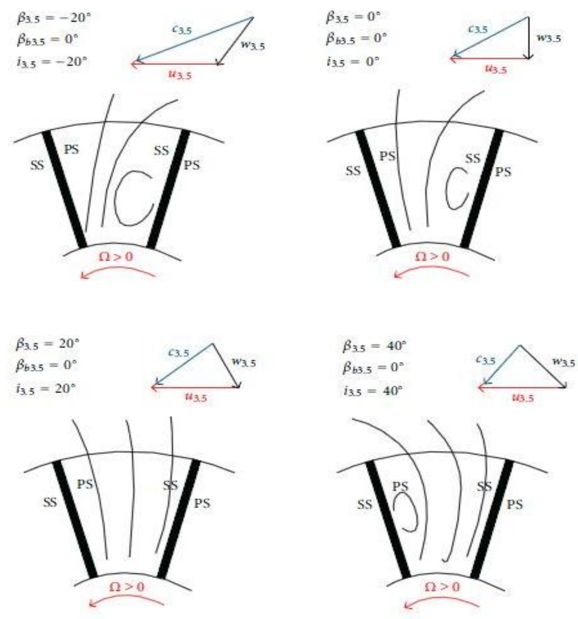

Figure 6. . Velocity diagrams at different relative flow angles.

In ideal conditions where there is no loss, no incidence, negligible swirl, the turbine efficiency becomes unity, the blade loading factor would also become equal to unity; thus, the blade speed ratio is given by:

$$\left(\frac{u_{3.5}}{Cs}\right)_{opt} = \sqrt{\frac{1}{2}} = 0.707 \tag{11}$$

The above value of 0.707 is commonly quoted as radial turbine optimum efficiency blade speed ratio. From previous equations we can see that it is itself a function of maximum turbine efficiency, even though the blade loading factor is constant. We know that the optimal efficiency for radial flow type is at loading factors less than 1, which means that there is rotor blades are approached with positive incidence by the flow, Incidence is given by:

Assuming a constant ratio of cm3.5/u3.5 of 0.5, by varying the incidence angle between 0 to 25 degrees, blade loading factor optimal incidence reduces to 0.77. Now upon assuming a maximum efficiency of 0.7, the optimal blade speed ratio is found to be approximately 0.67. Thus, for these parameters the radial flow type has its optimum efficiency at blade speed ratio below 0.707.

V SIMULATION AND RESULTS

A. Findings with Steady and Unsteady Flow Conditions

Steady flow conditions were assumed where the flow of air to exhaust gas turbine was continued and not in pulses. To determine the performance in steady flow conditions of the mixed flow turbine in comparison to the radial flow turbine used in normal automobiles, the same dimensions of the turbocharger were used, as previously discussed. Calculating the performance on the mixed flow turbocharger, the angle at exducer root mean was -52 degrees and the velocity ratio had been taken as 0.616. These dimensions were used to compare the normal turbocharger and the mixed flow turbocharger. The turbine volute inlet diameter is 38mm and the mass flow rate 0.678 kg/s. These dimensions resulted in the mixed flow turbine giving a pressure ratio of 2.91 at the same RPM as the radial flow type which gave a pressure ratio of 1.4. The total-to-static efficiency goes up with rotational speed only up to the 80% condition, beyond which it becomes constant. The total-to-static peak efficiency value (0.75 at 80% speed) takes place at a velocity ratio of 0.61, now this differs from a more known result that is the same takes place for radial flow turbines at a velocity ratio of 0.7. This feature where the turbine peak efficiency shifts towards a lower velocity ratio raises the overall turbine efficiency in the low velocity ratio region. This confirms that mixed flow turbines are better than there radial flow counterparts.

Initially calculations were started in radial turbines where the relative flow angle at the turbine wheel inlet was varied from -20 to 40 degrees; the same variations were made for the incidence angle. Since it has been previously proven that optimum flow within backswept rotor in case of radial flow turbine is mixed flow achieved at negative incidence angles, this leads to a relative flow angle at the turbine wheel inlet being approximately zero.

Considering mixed flow turbines to be radial flow turbines with back sweep, the calculations were started by taking values for the blade angle at the turbine wheel inlet to be 5 degrees and then gradually increased until 20 where the maximum pressure ratio was achieved. Here the incidence angle was kept negative since it had already been cited by many authors that a negative incidence angle is more favorable for optimum loading. Increasing the relative flow angle more than 40 degrees showed a great decrease in the efficiency below the optimum value which had been previously calculated to be near 0.707 theoretically.

Calculations were carried out at different RPM's. The result was that the total-to- static efficiency was seen to increase till 80 percent of the designed turbocharger speed and then seemed to stay constant. Here the efficiency was 75 percent which was already higher than the 60 percent efficiency obtained by traditional radial flow turbochargers. The other result was that the velocity ratio was still 0.61 rather than 0.7 which was different from what had been theoretically proven before. This contrast to the well-known result, shifting turbine peak efficiency towards lower velocity ratios, is rather more welcome since it increases the turbochargers efficiency in the low velocity area, which is where the turbocharger has to do most of its work in the case of normal city driving. This contrast was also good for unsteady state performance where the maximum exhaust gas energy occurs during the pulse peak area when the pressure ratio is high. It is worth noting that the design was also done keeping in mind that the flow would be vortex free in the housing even though that may not be possible in the real world, but the rest of the parameters were kept as close to the real world operating conditions as possible.

B. Calculations

First the blade speed ratio was calculated and then the blade rotor inlet angle and the absolute flow angle at the turbine rotor inlet were calculated. In the formula given below the term of the left side is the blade speed ratio which is the ratio of blade velocity and spouting velocity which was 0.616. Here the blade rotor inlet angle was 20 degrees which gave an absolute flow angle of 56 degrees. The spouting velocity was given to be 550 m/s.

$$\frac{u_2}{C_{is}} = \frac{1}{\sqrt{2}}\sqrt{1-\frac{\tan\beta_B}{\tan\alpha_2}} \tag{12}$$

Since assumed optimum conditions were assumed with no swirl and optimum incidence, the blade loading factor as mentioned in the previous chapter was thus 1. It is known that the blade speed ratio and the blade loading factor the total-to-static efficiency of the turbine was calculated 0.75 with the help of the following equation.

$$\eta_{T,ts} = \frac{\Delta h_{tt,stage}}{\Delta h_{tt,stage,is}} = \frac{\Psi u^2_{3.5}}{Cs^2/2} = 2\Psi\left(\frac{u_{3.5}}{Cs}\right)^2 \tag{13}$$

$$\psi = \frac{P_T}{m_T u^2_{3.5}} = \frac{\Delta h_{tt,stage}}{u^2_{3.5}} = \frac{C_{\phi 3.5} u_{3.5}}{u^2_{3.5}} - \frac{C\phi_4 u_4}{u^2_{3.5}} \quad (14)$$

The blade loading factor is unity, the mass flow rate and the flow velocity is already known, using the given equation we find the turbine power to be 77827 Joules. Furthermore, it is also known that the entire variables except the pressure ratio in the above equation, upon rearranging it the PR (pressure ratio) as 2.91 was calculated.

$$Power = \frac{1}{\eta} * m * \overline{C_p} * T_1 * \left(PR^{\frac{k-1}{k}} - 1 \right) \quad (15)$$

VI. Conclusions

Analyzing the components and design of turbochargers in diesel engine, the presented research has explained the detailed modeling and simulations of the turbocharger.

The mixed flow turbocharger was introduced and its fundamentals were discussed deeply. Conclusively, it has been seen that mixed flow turbines provide far greater efficiency than traditional turbochargers at normal operating speeds and in extreme conditions. They are slowly being investigated into more and more by automotive companies. Currently the mixed flow turbocharger is being used by Volkswagen in their 1.4 TSI engines which won the engine of the year award in 2012. Based on the development of mixed flow turbine, it is recommended to continue further the design and optimizations of the mixed flow turbocharger to improve its efficiency.

Appendix Nomenclature

p: Static pressure (Pa)
ptot: Stagnation pressure (Pa)
u3.5/cs: Blade speed ratio (—)
PI (also π): Pressure ratio (—)
c: Velocity in stationary frame (m/s)
w: Velocity in rotating, relative frame (m/s)
u: Blade speed (m/s)
Δh: Enthalpy difference (J/kg)
C: Compressor
D: Diameter
cs: Isentropic spouting velocity (m/s)
R: Degree of reaction (-)
i: Incidence (deg,◦)
$\dot{m}$: Mass flow rate (kg/s)
n: Rotational velocity (1/s)
Δt: Time difference (s)
t: Time (s)
P: Power (W)
T: Torque (Nm)
h: Enthalpy (J/(kg K))
s: Entropy (J/(kg K)).
Cp: Average constant specific heat = 804 J/kg*k
K: ratio of specific heats = 1.4

Greek Symbols

Γ : Cone angle (deg)
φ : Rake or camber angle (deg)
βb: Blade angle (deg)
β : Relative flow angle (deg)
α: Absolute flow angle(deg)
Ψ : Loading coefficient (—)
Ψ*: Simplified loading coefficient (—)
ω : Rotational speed (rad/s).

Indices

opt: Optimum
s: Static
t: Total
z: Axial,inz-direction
r: Radial
is: Isentropic
ss: Static to static
tt: Total to total
ts: Total to static
3: Turbine stage inlet
3.5: Turbine wheel inlet
4: Turbine (wheel) exit
m: Meridional
T: Turbine
ϕ : Circumferential

Conflict of Interest

The authors declare no conflict of interest.

Author Contributions

This work is based on the thesis work done by the second author Yasho Vinay Pipada under the supervision of the first author R.Udayakumar. Both the authors had approved the final version.

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