Chapter 10
Design of Turbocharger Platforms

10.1 Introduction

After learning how to deal with rotordynamics, bearing systems, balancing of turbochargers, and tribology in the bearings, we will learn how to design a platform of automotive turbochargers so that most current and future customer requirements should be covered in the platform. Therefore, we do not need to design a specific turbocharger for each customer requirement. Instead, we just take a suitable CHRA (Center Housing and Rotating Assembly) and insert it into the required housings of compressor and turbine for the customer applications. Additionally, we cut short the development time, time to market, and save money as well.

Furthermore, the common parts of turbochargers, such as compressor and turbine wheels, seal rings, thrust bearings, etc., could be used in different type series in the platform. As a result, much money can be saved if we order a huge number of the same part instead of different parts with a small number at the suppliers.

All the above considered are the reasons why the platform is necessary and important. Note that it is very difficult, sometimes impossible and costs a lot of time and money to correct the already unsuitable platform of turbochargers.

10.2 Market Analyses of Combustion Engines

Before we design the platforms of turbochargers for passenger (PV) and commercial vehicles (CV), two questions need to be answered. First, which combustion engines are required for the worldwide market in the future? Second, how many type series of turbochargers in the platform could cover most applications of customers currently and in the future?

In the following section, a market analysis of diesel engines for passenger vehicles (PV) is chosen as an example of designing turbocharger platforms for

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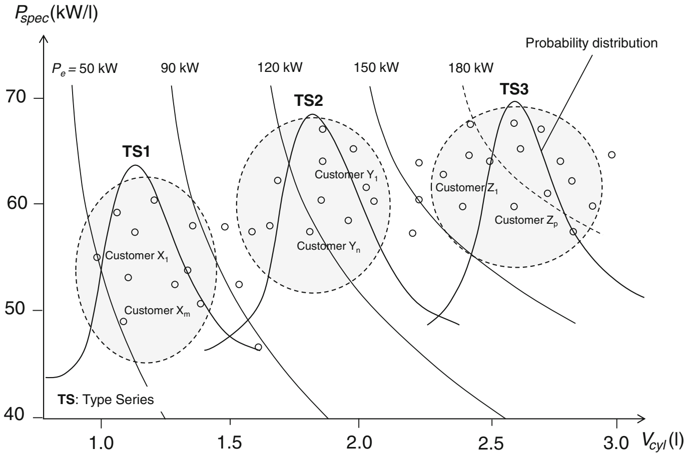

Fig. 10.1 Schematic market analysis for diesel passenger vehicles

passenger vehicles (PV). Using the same method, the turbocharger platform for commercial vehicles (CV) is designed.

The specific and total engine powers versus engine cylinder volume are displayed in Fig. 10.1. The density probability distribution of the engines in the market concentrates on three main areas of V_{cyl} from 0.9 to 1.5 l, 1.6 to 2.3 l and 2.5 to 3.0 l. The type series of the turbocharger platform should cover the required engines.

As a result, three *type series* (TS) of turbochargers TS1, TS2, and TS3 are designed for the platform of passenger vehicles respectively. Note that some engines in the requirement are intentionally not considered in the platform of turbochargers because they are only used in a small number of very special applications. Therefore, they are not commonly used in the platform. If they had been taken into account in the platform, it would lead to high cost for the platform development. Therefore, we concentrate on the essential thing and exclude such applications in order to save development time and cost.

The first type series TS1 for small engines with maximum engine powers from 45 to 90 kW, the second TS2 for middle engines (95 to 145 kW), and the third TS3 for large engines (155 to 200 kW) are summarized with maximum specific powers (Table 10.1). The engine specific power is defined as an engine power per unit of cylinder volume (kW/l).

Table 10.1 Type series for a diesel turbocharger platform of PV

Type series	Cylinder Volume ***V***cyl (l)	Maximum engine power P_e (kW)	Maximum specific power ***P***spec (kW/l)
TS1	0.9–1.5	45–90	50–60
TS2	1.6–2.3	95–145	60–63
TS3	2.5–3.0	155–200	62–67

Generally, the platform of turbochargers must be defined at the beginning of the development. Unfortunately, neither the geometrical sizes of the compressor and turbine wheels nor the shaft diameters of rotor of each type series are known in the platform.

In the following sections, we apply the physical similarity laws to designing the new platform. The physical similarity laws are based on many theories of combustion engines, turbomachinery, technical thermodynamics, rotordynamics, bearing dynamics, and classical mechanics.

10.3 Calculating Sizes of Compressor and Turbine Wheels

The geometrical sizes of the compressor and turbine wheels are calculated for each type series in the platform using the physical similarity law. It is obvious that the engine power is proportional to the mass flow rate of the charge air, as discussed in Chap. 1. In principle, the larger the mass flow rate of air, the higher the engine power will be.

The engine power is calculated according to Eq. (1.6a) as [1–3]

$$P_e = \eta_f Q_f \cdot \left(\eta_{vol} \rho_a V_{cyl} \frac{N}{n_R} \right) \cdot \frac{1}{AFR} \propto \dot{m}_C \tag{10.1}$$

where $\dot{m}_C$, the mass flow rate of the charge air for the engine, is defined as

$$\dot{m}_C = \eta_{vol} \rho_a V_{cyl} \frac{N}{n_R}.$$

Using thermodynamics of turbomachinery according to Eq. (2.14), the real compressor power is calculated [1, 3] as

$$P_C = \frac{\dot{m}_C c_{p,a} T_1}{\eta_C} \left[\left(\frac{p_2}{p_1} \right)^{\left(\frac{\kappa - 1}{\kappa} \right)_a} - 1 \right] \propto \dot{m}_C \tag{10.2}$$

The mass flow rate of the charge air from the compressor is proportional to squared inflow diameter D_1 of the compressor wheel (cf. Fig. 10.2).

$$\dot{m}_C \propto D_1^2 \tag{10.3a}$$

Using the definition of the trim ratio TR of the compressor wheel ($TR = D_1/D_2$), the inflow diameter D_1 (*inducer diameter*) of the compressor wheel is written in its outflow diameter D_2 (*exducer diameter*).

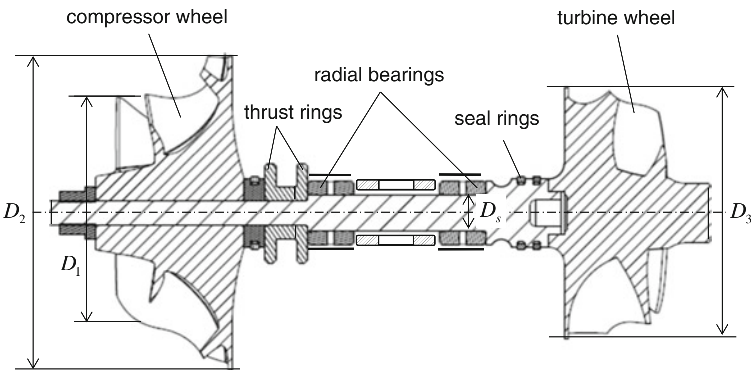

Fig. 10.2 Setup of the CHRA of a turbocharger

$$D_1 = D_2 \cdot TR \tag{10.3b}$$

Substituting Eq. (10.3b) into Eq. (10.3a), one obtains the mass flow rate of the charge air from the compressor.

$$\dot{m}_C \propto D_2^2 \cdot TR^2 \tag{10.4}$$

Using Eq. (10.1), the physical similarity law between the engine and compressor powers gives the relation of the engine power and outflow diameter of the compressor wheel [4].

$$\begin{aligned} P_e &\propto \dot{m}_C \propto D_2^2 \cdot TR^2 \\ \Rightarrow D_2 &\propto \frac{\sqrt{P_e}}{TR} = f \cdot \frac{\sqrt{P_e}}{TR} \end{aligned} \tag{10.5}$$

The result of Eq. (10.5) indicates that the exducer diameter D_2 of the compressor wheel is proportional to the square root of the engine power and inversion of the trim ratio of the compressor wheel. However, we do not know the proportional factor f for the turbocharger type series. Now, the question is how to determine D_2 corresponding to the engine power for the type series in Table 10.1.

Usually, a few designed turbochargers must be built to measure the proportional factor. This work would take a long time and cost a lot of money. Fortunately, the competitors had already done this task. However, we cannot use it one-to-one in the designed platform because the type series are different from the competitor type series. Notwithstanding, we can use their results with subtle skill to achieve the target in a very short time and at minimum cost.

In order to save the development time, time to market, and cost, we use the similarity between the designed turbocharger and the commercial turbochargers on the market (marked *) to find out the proportional factor f for calculating the exducer diameter D_2 of the designed turbocharger type series in the platform.

The similarity of the engine powers gives the relation of related parameters.

$$P_e \propto D_2^2 \cdot TR^2 \Leftrightarrow P_e^* \propto \left(D_2^*\right)^2 \cdot TR^{*2}$$

Thus, the exducer diameter D_2 of the designed compressor wheel results in [4]

$$\begin{aligned} \frac{P_e}{P_e^*} &= \frac{D_2^2 \cdot TR^2}{\left(D_2^*\right)^2 \cdot TR^{*2}} \\ \Rightarrow D_2 &= D_2^* \cdot \left(\frac{TR^*}{TR}\right)\sqrt{\frac{P_e}{P_e^*}} \end{aligned} \tag{10.6}$$

In this case, all starred parameters of the commercial turbochargers on the market have been known; they are used to calibrate the proportional factor for the platform. At the given engine power P_{e} and the trim ratio TR (normally, $TR \approx 0.76$ for compressor wheels), the exducer diameters D_2 of the compressor wheel are calculated for each type series of the platform using Eq. (10.6).

The wheel ratio of D_2/D_3 between the exducer diameter of compressor wheel and the inducer diameter of turbine wheel is chosen between 1.05 and 1.25 [4].

$$\frac{D_2}{D_3} = 1.05\text{–}1.25 \tag{10.7}$$

In Table 10.2, the wheel ratios of D_2/D_3 are given in the diagonal band of the platform matrix. Outside this range, the turbochargers are not suitable for automotive applications. If the wheel ratio is too small, the turbine wheel is larger than the compressor wheel. Thus, the mass center of the CHRA lies nearer to the turbine radial bearing on

Table 10.2 Ratios of compressor to turbine wheels in the platform

		Compressor wheel diameter D_2 (mm)								
		D_{21}	D_{22}	D_{23}	D_{24}	D_{25}	D_{26}	D_{27}	D_{28}	D_{29}
Turbine wheel diameter D_3 (mm)	D_{31}	1.10	1.15							
	D_{32}	1.05	1.10	1.25						
	D_{33}		1.05	1.15						
	D_{34}			1.10	1.20	1.25				
	D_{35}			1.06	1.15	1.20				
	D_{36}					1.10	1.20			
	D_{37}						1.12	1.20	1.25	
	D_{38}						1.06	1.15	1.20	
	D_{39}								1.08	1.20
CHRA Platform:		Type series 1 (45–90 kW)			Type series 2 (95–145 kW)			Type series 3 (155–200 kW)		

the RHS; it leads to a large load acting upon the turbine bearing. This could cause rotordynamic instability and bearing failure at the turbine side. Otherwise, the wheel ratio is too large; i.e., the turbine wheel is much smaller than the compressor wheel. It leads to large axial load on the thrust bearing that is between the thrust rings. This could cause a bad transient behavior of the rotor (large turbolag) and possibly failure of the thrust bearing due to overload in the thrust bearing.

By experience, the ratio should be chosen between 1.05 and 1.25 in order to compromise the acting loads on the turbine bearing and the thrust bearing. As a result, the inducer diameters D_3 of turbine wheels are calculated using Eq. (10.7) for each type series in the platform.

Until now, the rotor shaft diameter D_s has not been dealt with. In the following section, we will learn how to compute it for the designed platform.

10.4 Calculating Diameters of the Rotor Shaft

Besides the key function of connecting the compressor and turbine wheels, the rotor shaft has two additional functions: supporting the radial bearings and keeping the compressor wheel fixed on the rotor. This is against the aerodynamic torque M_{CW} acting on the compressor wheel and keeps the compressor wheel rotating.

The friction torque of the compressor wheel on the rotor shaft is used to determine the rotor shaft diameter. The rotor shaft diameter D_s is defined as the journal diameter inside the radial bearing, as shown in Figs. 10.2 and 10.3.

The aerodynamic torque acting on the compressor wheel (compressor wheel torque) is calculated using the theory of turbomachinery at a rotating speed ω [1, 2].

$$\begin{aligned} P_C &= M_{CW}\omega \propto (D_2)^2 \cdot TR^2 \\ \Rightarrow M_{CW} &= \frac{P_C}{\omega} \propto \frac{(D_2)^2 \cdot TR^2}{\omega} \end{aligned} \tag{10.8}$$

The maximum circumferential velocity of the compressor wheel is assumed as constant due to stability of the compressor wheel.

$$U_2 = \frac{\omega D_2}{2} = const. \tag{10.9a}$$

Using Eq. (10.9a), the rotor speed is proportional to the inversion of the compressor wheel diameter.

$$\omega \propto \frac{1}{D_2} \tag{10.9b}$$

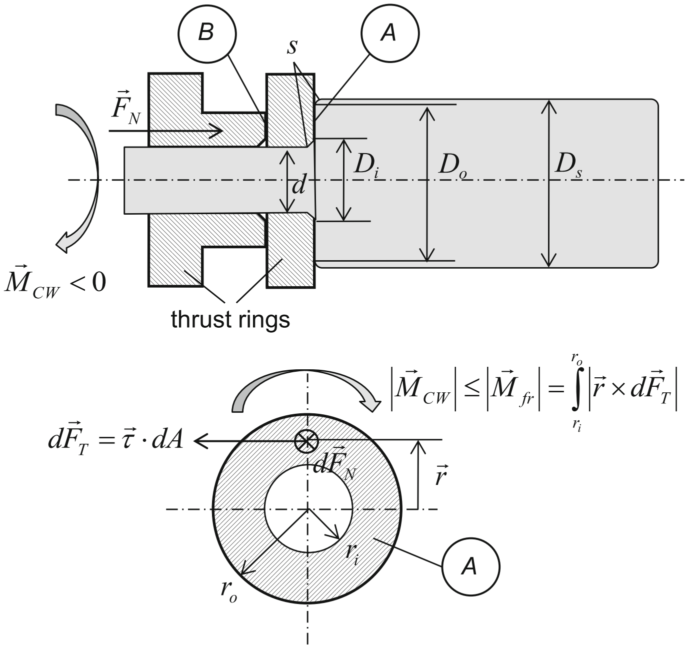

Fig. 10.3 Friction torque versus compressor wheel torque

Substituting Eq. (10.9b) into Eq. (10.9a), one obtains the compressor wheel torque

$$M_{CW} \propto \frac{(D_2)^2 \cdot TR^2}{\omega} = (D_2)^3 \cdot TR^2 \tag{10.10}$$

The friction torque acting on the contact areas A and B between the rotor shaft and thrust rings must be at least larger than the compressor wheel torque in order to keep the compressor wheel fixed on the rotor.

At first, integrating the tangential force F_T over the rotor radius r, the friction torque is calculated at the contact area A.

$$\begin{aligned} M_{fr} &= \int r dF_T = \int_A r\tau dA \\ &= \int_{r_i}^{r_o} r(2\pi r\tau)dr = \frac{\pi}{12}\tau\left(D_o^3 - D_i^3\right) \\ &\propto \left(D_o^3 - D_i^3\right) \end{aligned} \tag{10.11}$$

where τ is the shear stress in the contact area A.

The aerodynamic torque on the compressor wheel must be smaller than the friction torque in order to keep the compressor wheel from not slipping over the rotor shaft. Hence, the shaft diameter D_s of the rotor results from the necessary condition [4].

$$\begin{aligned} M_{CW} \propto (D_2)^3 \cdot TR^2 \leq M_{fr} \propto \left(D_o^3 - D_i^3\right) \\ \Rightarrow (D_2)^3 \cdot TR^2 \leq \left(D_o^3 - D_i^3\right) \end{aligned} \tag{10.12}$$

The inner and outer diameters in Eq. (10.12) are calculated as

$$\begin{aligned} D_i &= d + 2s; \\ D_o &= D_s - 2s \end{aligned} \tag{10.13}$$

where
d is the bore diameter of the compressor wheel;
D_s is the rotor shaft diameter;
s is the fillet radius ($s \approx 0.300$–0.350 mm).

Substituting Eqs. (10.12 and 10.13), one obtains the function of the rotor shaft diameter in other parameters [4].

$$M_{CW} \propto (D_2)^3 \cdot TR^2 \leq M_{fr} \propto (D_s - 2s)^3 - (d + 2s)^3 \tag{10.14}$$

The similarity law between the designed and commercial turbochargers on the market (marked *) gives the required rotor shaft diameter D_s [4].

$$\begin{aligned} M_{CW} : (D_2)^3 \cdot TR^2 \leq (D_s - 2s)^3 - (d + 2s)^3 \\ \Leftrightarrow M^*_{CW} : (D^*_2)^3 \cdot TR^{*2} \leq (D^*_s - 2s^*)^3 - (d^* + 2s^*)^3 \\ \Rightarrow \frac{(D_s - 2s)^3 - (d + 2s)^3}{(D^*_s - 2s^*)^3 - (d^* + 2s^*)^3} \geq \frac{(D_2)^3 \cdot TR^2}{(D^*_2)^3 \cdot TR^{*2}} \end{aligned}$$

Therefore, the rotor shaft diameter results as

$$D_s \geq 2s + \left[(d + 2s)^3 + \left(\frac{D_2}{D^*_2}\right)^3 \left(\frac{TR}{TR^*}\right)^2 \left[(D^*_s - 2s^*)^3 - (d^* + 2s^*)^3\right]\right]^{\frac{1}{3}} \tag{10.15}$$

Likewise, the required starred parameters in Eq. (10.15) of the commercial turbochargers have been already known. Therefore, it is easy to calculate the rotor shaft diameter D_s for the platform. To determine the rotor shaft diameter of the type

Table 10.3 An examplary platform of automotive turbochargers (Diesel PV)

Turbine wheel diameter D_3 (mm)	Compressor wheel diameter D_2 (mm) →								
	D_{21}	D_{22}	D_{23}	D_{24}	D_{25}	D_{26}	D_{27}	D_{28}	D_{29}
D_{31}	1.10	1.15		Shaft diameter $D_{s2} > D_{s1}$			Shaft diameter $D_{s3} > D_{s2}$		
D_{32}	1.05	1.10	1.25						
D_{33}		1.05	1.15						
D_{34}			1.10	1.20	1.25				
D_{35}			1.06	1.15	1.20				
D_{36}					1.10	1.20			
D_{37}						1.12	1.20	1.25	
D_{38}	Shaft diameter D_{s1}					1.06	1.15	1.20	
D_{39}								1.08	1.20
CHRA Platform:	Type series 1 (45–90 kW) 			Type series 2 (95–145 kW) 			Type series 3 (155–200 kW) 		

series, its high-end diameter D_2 (i.e., the maximum diameter) of the compressor wheel should be employed in Eq. (10.15) because the largest compressor wheel induces the highest aerodynamic torque acting on the compressor wheel.

The rotor shaft diameter D_{si} in the platform should be at least equal to the RHS of Eq. (10.15). Note that if the fillet radius s is larger due to some production processes, the RHS also increases with the fillet radius s. The rotor shaft diameter D_{si} should be checked for whether the condition for D_s in Eq. (10.15) is satisfied for the application. If it is not the case, some measures of surface coating between the thrust rings and rotor shaft should be additionally used to increase the surface friction coefficient. As a result, the condition in Eq. (10.12) of $M_{fr} \geq M_{CW}$ is always satisfied.

In general, the higher the engine power, the larger the rotor shaft diameter will be, as shown in Table 10.3. It is obvious that the higher engine power requires more mass flow rate of the charge air. Therefore, the compressor wheel must be larger (increased D_2) to provide the engine with more mass flow rate of the charge air. This requires a larger shaft diameter according to Eq. (10.15).

Analogously, the same computation at the contact surface B is carried out to design the geometry of the thrust rings.

Some applications overlap at the boundary between the type series (See Table 10.3). For this case, the condition in Eq. (10.12) between the compressor wheel and friction torques must be checked by FEM computations.

10.5 Design of CHRA Geometry for the Platform

In this section, the overall geometry of CHRA is calculated for each type series in the platform. Normally, the geometrical computation begins from inside to outside of the rotor center in both axial and radial directions (cf. Fig. 10.4).

At first, the bearing distance l_0 between two radial bearing centers is chosen by experience so that the ratio of l_0 to the shaft diameter D_s lies between about 3.2 and 3.5 for turbochargers of passenger vehicles (PV) and between approximately 3.7 and 4.0 for turbochargers of commercial vehicles (CV) [4].

$$\frac{l_0}{D_s} = \begin{cases} 3.2-3.5 \text{ for } PV \\ 3.7-4.0 \text{ for } CV \end{cases} \tag{10.16}$$

If the ratio is too small, the rotor mass center is nearer to the turbine bearing leading to a large radial load acting on the turbine bearing and possibly to the rotor instability. Otherwise, the ratio is too large, the total length of the CHRA increases so that the turbocharger total length does not fit in the given space under the car hood. Note that the longer and slender the rotor is, the more easily unstable the rotor is with conical vibration mode.

The next step is to design the radial bearings discussed in the earlier chapters. Both bearings at the compressor and turbine sides have the same geometry that avoids confusing different radial bearings at the assembly line.

The length between the bearing ends l_b results from the bearing width W and bearing distance l_0. It is written as a function of D_s and W using Eq. (10.16).

$$l_b = l_0 + W = f(D_s, W) \tag{10.17}$$

Furthermore, the thrust bearing and seal rings are designed, as discussed in Chap. 6. Using aerodynamics of the turbomachinery, both compressor and turbine

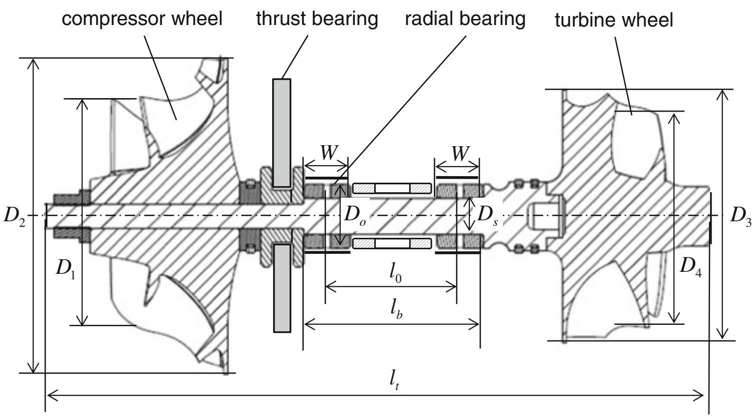

Fig. 10.4 Geometry of the CHRA of a type series

geometries are also computed. Finally, the total length l_t of the CHRA results from l_b and all related lengths l_i.

$$l_t = l_b + \sum_i l_i \tag{10.18}$$

Likewise, the CHRA geometry in radial direction begins from the shaft diameter, the outer diameters of the radial bearing, thrust rings, thrust bearing, and the outflow diameters of the compressor and turbine wheels.

Having known the CHRA geometry in both axial and radial directions, the bearing housing and the housings of the compressor and turbine wheels with all connections of oil, cooling water, exhaust gas, and charge air to the engine could be designed in the given space under the car hood.

References

1. Baines, N.C.: Fundamentals of turbocharging. Concepts ETI, Inc, Boston (2005)
2. Heywood, J.B.: Internal combustion engine fundamentals. McGraw-Hill, New York (1988)
3. Golloch, R.: Downsizing bei verbrennungsmotoren. Springer, Heidelberg (2005)
4. Nguyen-Schäfer, H.: Design of turbocharger platforms, Personal documents (2012)