Experimental investigation of upstream installation effects on the turbocharger compressor map

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ABSTRACT
This paper experimentally investigates the effects of an upstream bended pipe on the compressor speedline slopes and surge line. Different orientation angles for the incoming bended pipe relative to the compressor scroll are investigated. The tests were carried out on a cold gas stand on a passenger car sized turbocharger. A bended pipe upstream of the compressor leads to an increase of the surge margin. This effect does not depend on the orientation of the bend. Comparisons with a straight inlet with artificially generated pressure losses indicate that the increase in operating range is an effect of the pressure losses generated in the bend.

NOMENCLATURE

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<th>Latin letters</th>
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<td>A</td>
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<td>Greitzer B parameter</td>
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<td>Impeller speed [rpm]</td>
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<td>Temperature</td>
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<td>TPR</td>
<td>Total pressure ratio</td>
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<td>Total temperature ratio</td>
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<td>U</td>
<td>Impeller tip speed [m/s]</td>
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Indices

| 1 | Before compressor |
| 2 | After compressor |
| C | Compressor |
| c | corrected |
| p | Pressure signal |
| T | Throttle |
| t-t | Total-to-total |
| ref | Reference value |
INTRODUCTION AND BACKGROUND

The operating range of turbocharger compressors is an important factor in modern boosted internal combustion engines. The compressor must be able to deliver mass flow and boosting pressure as demanded by the engine operating point. The mass flow is approximately proportional to the engine speed for constant brake mean effective pressure (BMEP). A higher BMEP, on the other hand, requires both a higher boosting pressure and a higher mass flow for a given fuel-to-air ratio.

The range of a turbo-compressor is limited by three factors. At high mass flows, transonic flow conditions may be reached. The flow gets choked, and this limit is called the choke line. At high impeller speeds, the compressor operation is limited by mechanical stresses. Finally, at low mass flows, the compression system becomes unstable, with the flow periodically flowing from the high-pressure volume at the compressor outlet towards the low-pressure volume at the compressor inlet. This pumping behaviour is called surge. The surge line limits the engine operating range at low speeds and high boosting pressure demands, i.e. high BMEP.

The impeller and the diffuser of a centrifugal compressor are usually designed for a certain mass flow at a certain rotational speed, the so-called aerodynamic design point. Deviations from this operating point will alter the flow path, increasing the likelihood for wall adverse pressure gradients and flow separation to develop. Naturally, this decreases compressor’s efficiency. For very low mass flows, the losses can become so large that the pressure ratio decreases for decreasing mass flows, and the speedline slope becomes positive. As shown by Greitzer for axial compressors in (1; 2) and confirmed by Fink et al for centrifugal compressors (3), a positive compressor slope can lead to an unstable compression system, depending on the resistance of other elements in the flow loop, e.g. the throttle valve, and the energy that can be stored in the compressed volume between compressor and throttle. The pressure ratio and mass flow start oscillating, because the volume is filled and emptied with a characteristic low frequency. This pumping behaviour is called surge. A theoretical limit for the stability of a compression system derived in (1), using a static (Equation (1)) and a dynamic (Equation (2)) stability criterion, is

$$\frac{\partial \psi_C}{\partial \phi_C} \leq \frac{\partial \psi_T}{\partial \phi_T} \quad (1)$$

$$\frac{\partial \psi_C}{\partial \phi_C} \leq \frac{1}{B^2} \left(\frac{\partial \psi_T}{\partial \phi_T}\right)^{-1} \quad (2)$$

where $\psi$ is the dimensionless pressure parameter and $\phi$ is the dimensionless mass flow parameter. The indices $C$ and $T$ refer to the compressor and the throttle (or other flow resistance downstream the compressor). $B$ is a parameter that relates the pressure to the inertia forces in a duct, see (4). It is defined as

$$B = \frac{U}{2\omega_H L_c} \quad (3)$$

where $U$ is the impeller tip speed, $\omega_H$ is the Helmholtz frequency of a system with equivalent duct and plenum dimensions, and $L_c$ is the length of the compressor duct. A larger value of $B$ makes a system more likely to enter surge operation. This theoretical model has been incorporated into larger engine models e.g. by Theotokatos and Kyrtatos in (5) and by Galindo et al in (6).

The upstream geometry of the piping leading to the compressor has been shown to influence the stability of the compression system. Kim et al (7) showed
experimentally that a pipe bend upstream the impeller reduces the compressor efficiency, and concluded that this happens mainly due to deterioration of the impeller flow. More specifically for the surge limit, Galindo et al (8) showed that a tapered inlet, a bent pipe, and a volume at the inlet all improve the surge margin, especially at higher pressure ratios, and Serrano et al (9) investigated the effect of different elbows, with and without guide vanes, on pressure ratio and the surge limit.

In this paper the surge line dependency on a bended inlet pipe is further investigated by also changing the relative orientation of the bend to the compressor volute outlet. The objective is to determine whether this orientation has any effects on the lower mass flow half of the compressor map. In section 2, the experimental setup and the inlet geometries tested are explained. In section 3.1, the compressor maps measured for a straight and a bended inlet are given. The effects of the bend orientation relative to the compressor are investigated in section 3.2. Based on these preliminary results, a more detailed investigation into the surge line at high speeds using theoretical and experimental criteria is carried out in section 3.3. The results are concluded in section 4.

2 METHOD

2.1 Experimental setup

The tests were run on a Garrett GT1752 turbocharger, which was installed on a cold gas stand. The compressor impeller has an exducer diameter of 52 mm and a TRIM of

$$TRIM = \left(\frac{D_{\text{Inducer}}}{D_{\text{exducer}}}\right)^2 = 0.53.$$  \tag{4}

Its design mass flow is 115 g/sec. The six main blades and six splitter blades are backwards swept. It is followed by a vaneless diffuser and a volute. The turbine is fed with compressed air, which is heated to 65 °C in order to prevent icing after the expansion.

The air is induced by the compressor from the test chamber through a funnel when using the straight inlet pipe, or through a bellmouth for the bended pipe configurations. A schematic and sensor locations are shown in Figure 1. The straight pipe has a length of $L = 17D$. Total temperature and pressure were measured at the inlet. For the configurations with a bended inlet, static wall pressure was also measured downstream the bend at two other circumferential positions ($p_{2a}$, $p_{2b}$) using Kulite WCT-312M (0...1.7 bar abs) sensors.

![Figure 1: Schematic of the rest rig. Measurements are marked by white circles.](image-url)
The compressor mass flow is throttled by a pneumatic valve located 16\(D\) downstream the compressor. This valve position relatively close to the compressor was chosen in order to keep the volume small, \(V \approx 1\) L, and limit the amplitude of surge cycle pressures. Total pressure and temperature are measured 10\(D\) downstream the compressor; wall static pressure is measured both 3 and 10\(D\) downstream \((p_{2a}, p_{2b})\) using a piezoresistive Kistler 4045A5 (0...5 bar abs) sensor. The mass flow is measured with the help of an orifice downstream the pneumatic valve. An orifice was chosen to get very accurate mass flow measurements, at the cost of not being able to determine mass flow oscillations in surge.

The compressor maps were measured in accordance with the SAE testing standard for turbocharger maps (10), i.e. speedline wise from high to low mass flows. Since the current investigation is mainly concerned with surge, and a larger orifice would be needed for high mass flows, measurements were taken only at mass flows lower than the aerodynamic design mass flows. The last operating point displayed corresponds to the minimum mass flow before the compression system enters deep surge. Since the mass flow is not measured in a time-resolved manner, the common definition of deep surge as operation with an at times negative mass flow could not be used. A criterion based on the outlet pressure oscillation magnitude as described in section 3.1 is applied instead.

2.2 Inlet pipe geometries and orientation tested
The compressor inlet geometries tested are given in Figure 2.

![Figure 2: Inlet geometries that were tested. (a) shows the default straight inlet, while (b) and (c) show the bended inlet configuration and its clocking.](image)

The baseline case shown in Figure 2 (a) consists of a straight inlet with an inner diameter of \(D = 44\) mm, the same as the compressor inlet. For the next cases, a bent pipe is mounted 2\(D\) upstream the compressor inlet. The pipe has a ratio of inner radius to centreline radius of \(R_i/R_c = 1.04\). It has the possibility to be rotated around the compressor axis, as shown in Figure 2 (c) for verifying "clocking" effects. In this paper, the results for the orientations 3, 6, 9, 10, and 12 o'clock were tested. Since the volute is not axisymmetric, with the tongue at a position of 4 o'clock, a non-uniform pressure distribution is expected for mass flows different from the design mass flow. The aim is to investigate whether any optimal positioning between inlet angle and volute outlet exists.

2.3 Repeatability
A compressor operating point was determined as steady state if the isentropic efficiency as calculated acc. to Equation (8) fluctuated with a maximum amplitude of 0.3 percentage points around a constant mean value. The sampling time for each point was two minutes. In order to estimate the repeatability of the measurements, some operating points were sampled at least five times over two days. The operating points were reached both by throttling from a higher mass flow, i.e. according to the SAE standard (10), and by de-throttling from lower mass flows. The mass flow standard deviation of the samples for one operating point was \(s_{MFC} = 0.3...0.5\%\) of the mean (depending on the operating point), which is within the measurement accuracy of mass flow measurements using the pressure drop.
over an orifice. The total pressure ratio standard deviation of the samples was around $s_{TPR} \approx 0.15\%$, again within the measurement accuracy of the pressure transducer. There was no detectable difference between throttling and de-throttling towards a given operating point. The measurements are therefore deemed repeatable.

3 RESULTS

3.1 Compressor Maps
In this section, the compressor maps for the different inlet geometries are compared. The compressor maps are characterized by the corrected mass flow ($MF_c$), the corrected speed ($N_c$), the total pressure ratio from inlet to outlet ($TPR$), and the isentropic efficiency ($\eta_{is,t-t}$) defined by the relations (5), (6), (7), and (8), respectively.

$$MF_c = \frac{MF \cdot \sqrt{T_1/T_{ref}}}{p_1/p_{ref}}$$  \hspace{1cm} (5)

$$N_c = \frac{N}{\sqrt{T_1/T_{ref}}}$$  \hspace{1cm} (6)

$$TPR = \frac{p_{02}}{p_{01}}$$  \hspace{1cm} (7)

$$\eta_{is,t-t} = \frac{\frac{\kappa - 1}{\kappa} \cdot TPR^\kappa - 1}{TTR - 1}$$  \hspace{1cm} (8)

In the above relations, $MF$ is the physical mass flow; $T_1$ and $p_1$ are the static inlet temperature and pressure; $T_{ref}$ and $p_{ref}$ are a reference pressure and temperature; $N$ is the physical impeller rotational frequency; and $TTR$ is the total temperature ratio $T_{02}/T_{01}$. The correction of mass flow and impeller speed is made to ensure Mach similarity, i.e. the same axial and circumferential Mach number irrespective of inlet conditions. The isentropic efficiency compares the actual enthalpy increase with that of an isentropic compression reaching the same pressure ratio.

Figure 3 shows a comparison between the map measured by the supplier on a hot gas stand (in blue), and the map measured on the cold gas stand, for a straight inlet with inner diameter $D = 44$ mm (in red). The speedlines for a configuration using a bended inlet pipe with a bend towards 9 o'clock, see also Figure 2 (c), are also drawn in black.

Comparing the hot and cold gas stand measurements using a straight inlet pipe, one can see that the $TPR$ at high mass flows is higher on the cold gas stand than on the hot gas stand. This could be due to the lack of heat transfer from the turbine to the compressor, or even cooling of the compressor housing at higher speeds. Heat addition to a compressible flow reduces its total pressure, cp. a Rayleigh flow (11). The pressure difference is more pronounced for higher mass flows, which could be explained by the higher Nusselt numbers in both turbine and compressor, and thus higher heat transfer (12). The slope of the speedlines on the cold gas stand is close to zero or even positive, whereas the hot gas stand compressor map shows consistently negative slopes except at the very highest speedline. At speeds of $N_c \geq 160$ krpm, this leads to higher pressure ratios on the hot gas stand. A possible explanation could be that the acceleration of the flow due to the heat addition prevents local stall, since the flow is better aligned with the geometry designed for
higher flow rates. The surge line given by the supplier is at higher mass flows than measured, though this could be purely an effect of a safety margin towards surge.

As a surge criterion, the signal power of the static outlet pressure \( p_{2b} \) in the frequency interval \([f_{\text{surge}} - 10 \text{ Hz}; f_{\text{surge}} + 10 \text{ Hz}]\) was calculated by integrating the power spectral density of the outlet pressure signal \( p_{2b} \) \( S_{pp} \):

\[
P_{p2b} = \int_{f_{\text{surge}}-10\text{Hz}}^{f_{\text{surge}}+10\text{Hz}} S_{pp} df
\]  

This criterion thus is a measure of the outlet pressure oscillations in the frequency interval around the surge frequency. A reference pressure of 20 \( \mu \text{Pa} \) was used to convert to decibel. A similar criterion was also used e.g. by Dehner in (13). The points shown in Figure 3 all have a value \( P_{p2b} \leq 120 \text{ dB} \), while operating points in deep surge are above this threshold. The dashed lines, coloured in red for the straight inlet and in black for the bended inlet, are the iso-lines \( P_{p2cb} = 110 \text{ dB} \). This value was chosen as an exemplary lower threshold. \( P_{p2b} \) is not monotonically increasing with reduced mass flows at some higher impeller speeds, which means that the compressor in some regions becomes more stable by decreasing the mass flow. In this case, the lines connect the largest mass flow where the value of 110 dB is reached. For low impeller speeds of 100 krpm, there is no significant difference in pressure ratio, but a slightly improved surge margin for the bended pipe configuration. At higher speeds and mass flows, the pressure ratio of the bended inlet configuration drops compared to the straight inlet configuration. The \( P_{p2b} = 120 \text{ dB} \) surge line is moved towards lower mass flows by about 10% at low
impeller speeds. For the highest impeller speed, the shift becomes much larger at about $\Delta MF_{120\text{dB}} = 20 \text{ g/s}$, which amounts to 25% of the straight inlet surge line mass flow. Thus, the 110 dB surge line differs from the 120 dB line only at high impeller speeds. The difference between the straight and the bended inlet increases significantly from $N_c = 160 \text{ krpm}$ to $N_c = 175 \text{ krpm}$. This confirms what was found by other authors as well (8; 9).

The drop in pressure ratio shown here is entirely the result of losses in the bend. If one corrects for those, the differences in pressure ratio disappear. Figure 4 shows the measured static wall pressures at 2 circumferential locations, 1 o’clock and 6 o’clock, for five different orientations of the bended pipe, namely 3, 6, 9, 10, and 12 o’clock (see also Figure 2 (c) for orientation purposes).

![Figure 4: Static inlet pressure $p_{1c}$ relative to test room stagnation pressure $p_{01c}$ for $N_c = 175 \text{ krpm}$.

Also plotted are the theoretical static pressure for an isentropic acceleration from stagnation to the bulk inlet velocity (which in turn is calculated from the measured mass flow), and a theoretical loss correction for bends using the equation proposed by Crawford et al in (14). For a given bend ratio $R/R_c = 1.04$, the equivalent pipe length may be estimated as

$$l_e/D = 1.25 \cdot \left( \frac{R_l}{R_c} \right)^{0.5} \cdot \text{Re}_D^{0.35}$$  \hspace{1cm} (10)$$

where $\text{Re}_D$ is the Reynolds number based on pipe diameter. The pressure losses can then be calculated as

$$\Delta p = 4f \cdot \frac{l_e}{D} \cdot \rho \cdot \frac{v^2}{2}$$  \hspace{1cm} (11)
where \( \rho \) is the inlet density, \( v \) is the bulk velocity, and \( f \) is the Fanning friction factor, estimated here using the equation from Haaland in (15) for turbulent circular pipe flow:

\[
\frac{1}{\sqrt{4f}} = -1.8 \cdot \log_{10} \left( \left( \frac{\varepsilon}{3.7D} \right)^{1.11} + \frac{6.9}{Re} \right)
\]  

(12)

with a surface roughness \( \varepsilon \). Figure 4 shows that this estimate gives a good fit to the measured pressures, although it slightly underestimates the losses for this bend at higher mass flows. Close to surge, the inlet static pressure is higher than the stagnation pressure in the test chamber. This is an indication of backflow from the impeller. Also noticeable is the fact that when this pressure increase is measured, the inlet pressure is higher at the circumferential position where the volute pressure is also expected to be higher. This points to a possible circumferentially uneven backflow from the compressor operating regimes close to surge, since the backflow has a higher potential in the 1 o’clock position.

If the pressure ratio is corrected for the reduced inlet pressure using the equations above, the difference in TPR can be explained entirely.

3.2 Inlet bend orientation effects

Since the largest differences between a straight inlet and a bent inlet were detected at the higher impeller speeds, these speedlines were chosen to investigate the bend clocking (orientation) effects. Figure 5 shows the highest speedline \( N_c = 175 \text{ krpm} \) for all five different bend orientations, 3, 6, 9, 10, and 12 o’clock. It shows that there is no difference outside the experimental uncertainty between the tested orientations, except at the peak TPR. Here, the orientation 9 o’clock results in a lower TPR, with differences up to \( \Delta TPR = 0.02 \), and a drop in efficiency by \( \Delta \eta_t-t = 1\% \). These results could also be confirmed for the lower speedline of \( N_c = 160 \text{ krpm} \) (not shown here). This drop in TPR could be related to the fact that in the 9 o’clock orientation, the highest velocities at the compressor inlet are at the same position where the highest pressures in the volute are expected. There is no discernible difference in the surge line as defined by the \( P_{p2b} = 120 \text{ dB} \) threshold between the different orientations of the bended inlet.

This result suggests that axisymmetric effects of the bend on the inlet velocity field are at least partly responsible for the shift in surge margin. Those could be a static pressure drop, or a corresponding increase in the axial inlet velocity. The increase in the inlet velocity is almost completely accounted for, however, by the correction of the mass flow, which ensures Mach similarity. The pressure drop also leads to a small drop in the static temperature and thus lower speed of sound. If one assumes an adiabatic pressure drop in the bend, the difference in axial inlet velocity is of the order of magnitude of 1%, for the same corrected mass flow. This can be shown using Fanno flow relations. It is thus unlikely that the increase in axial velocity is the cause for this shift in surge margin.

Another possibility is the stabilizing effects of a pressure drop in the bend on the compression system as given by the static and dynamic stability criteria, Eq. (1) and (2). The pressure losses occur upstream of the compressor. They increase approximately quadratic with the inlet axial velocity, and thus the (physical) mass flow. Therefore, they decrease the positive slope of the combined bend and compressor system, and have a stabilizing effect on the compression system.
3.3 Compressor speedline slope effects

In order to test this hypothesis, the highest speedlines $N_c = 160$ krpm and $N_c = 175$ krpm were tested with the straight inlet where a pressure loss was created in the inlet by a diffuser with flow separation. The diffuser was located 12D upstream of the compressor, so that the inlet flow is fully developed. It was decided to design the diffuser to generate pressure losses that result in an approximately flat speedline near surge for the combination of pressure loss device and compressor, in order to test whether an upstream device can in fact dampen the oscillations that occur in a dynamically unstable system.

Figure 6 shows the highest measured speedline $N_c = 175$ krpm in a non-dimensionalized $\phi-\psi$ compressor map for the straight inlet configuration (red stars), the bended inlet 9 o’clock configuration (black triangles), and the straight inlet configuration with the aforementioned pressure loss in the inlet pipe (blue circles). For each of these configurations, the limit where the outlet pressure fluctuation power $P_{p2b}$ reaches 110 dB and 120 dB respectively is marked. For the straight inlet and the bended inlet, the results correspond to those shown in Figure 3. There are some pressure oscillations at surge frequencies for the positively sloped part of the speedline, and the leftmost shown point is the last point with $P_{p2b} \leq 120$ dB. For the straight inlet with a pressure loss device, the speedline is negatively sloped except at very low mass flows. This means that the system is theoretically stable for most parts of the measured range. The threshold of $P_{p2b} = 110$ dB is not reached at all, except for values of $\phi < 0.089$, i.e. to the left of the last shown point. The results for $N_c = 160$ krpm show similar behaviour and are therefore not shown here.
The static stability criterion, Equation (1), is fulfilled for all points shown in the figure, since the throttle slope is very steep. Figure 6 shows theoretical estimates of the surge line using the dynamical stability criterion from Equations (2) and (3). The Helmholtz frequency for the calculation of the $B$ parameter was estimated using (see also Dehner et al (16))

$$\omega_H = a \sqrt{\frac{A}{L_c V_p}}, \quad \frac{A}{L_c} = \sum_i \frac{1}{\frac{L_{c,i}}{A_i}}$$  \hspace{1cm} (13)$$

where $a$ is the speed of sound at the compressor outlet, $A$ is the cross-sectional area and $L_c$ the length in meridional direction of the component, and $i$ counts through the inlet duct, the compressor and volute, and the outlet duct. The volume of the pipe between compressor outlet and throttle was used as plenum volume $V_p$. The frequencies estimated this way are within 10% of the frequencies measured in mild surge. The value of $B$ at 175 krpm is 0.53 for the straight inlet configurations and 0.55 for the bended inlet configurations (because the inlet duct is slightly shorter). The throttle slope $\partial \psi_r/\partial \phi_r$ was estimated assuming that only the pressure after the throttle and the temperature before the throttle are constant close to the measured operating point, so that only the pressure before the throttle changes. In Figure 6, the critical slope of the dynamic stability criterion is drawn at the point where it is first breached for each speedline. This happens when the speedline slope turns positive in all configurations. Thus, using the strict model criteria, the surge limit at the highest speedline would be located at a mass flow with a local $TPR$ peak. Due to the large losses created by the pressure loss device, the peak is shifted towards low mass flows, and the outlet pressure oscillations are much weaker.
Setting the threshold at $P_{p2b} = 120$ dB, one can see that while the pressure loss device in a straight inlet increases the operating range, the increase is less than that for the bended inlet. Therefore one can conclude that the speedline slope explains the milder oscillations of the compressor outlet pressure and a part of the shift of the onset of deep surge. It does not fully explain the increased operating range, however. The axis-asymmetric flow field following a bend is likely also partly responsible for this effect.

4 CONCLUSIONS

The left half of the compressor map was compared for two inlet configurations: A straight inlet pipe and a bended pipe. Bended pipe orientation effects relative to the compressor axis were investigated. The losses in the bend lead to a reduced total pressure ratio for high mass flows. The reduction in peak $TPR$ is almost independent of the clocking position, only the 9 o’clock position consistently shows slightly lower $TPR$ than other positions. Considering the surge line, the effects of the bend are strongly dependent on the definition of surge. With a theoretical criterion, the compression system becomes unstable as soon as the speedline slope turns positive. This gives an increase in the surge margin for a bended inlet as compared to the straight pipe inlet. When using a measure of the compressor outlet pressure signal power, $P_{p2b}$, the result is also an increase in the surge margin; the exact value of the increase depends on the chosen threshold.

A comparison with a straight inlet with a pressure drop device indicates that the increase in surge margin for the bended inlet is mainly a result of the stabilization of the compression system through the pressure losses in the bend. Since this also results in significantly lower boost pressures at high mass flows, however, it would likely not be beneficial to install such a system from an engine perspective. The results suggest that in SI engines with a throttle upstream the compressor, the throttle could be used to sacrifice boost pressure in order to suppress intake manifold pressure oscillations.

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REFERENCE LIST


