An evaluation of 1D loss model collections for the off-design performance prediction of automotive turbocharger compressors

P Harley\textsuperscript{1}, S Spence\textsuperscript{1}, J Early\textsuperscript{1}, D Filsinger\textsuperscript{2} and M Dietrich\textsuperscript{2}

\textsuperscript{1}School of Mechanical & Aerospace Engineering, Queen’s University, Belfast BT9 5AH, UK
\textsuperscript{2}IHI Charging Systems International GmbH, Heidelberg 69126, Germany

E-mail: pharley01@qub.ac.uk

Abstract. Single-zone modelling is used to assess different collections of impeller 1D loss models. Three collections of loss models have been identified in literature, and the background to each of these collections is discussed. Each collection is evaluated using three modern automotive turbocharger style centrifugal compressors; comparisons of performance for each of the collections are made. An empirical data set taken from standard hot gas stand tests for each turbocharger is used as a baseline for comparison. Compressor range is predicted in this study; impeller diffusion ratio is shown to be a useful method of predicting compressor surge in 1D, and choke is predicted using basic compressible flow theory. The compressor designer can use this as a guide to identify the most compatible collection of losses for turbocharger compressor design applications. The analysis indicates the most appropriate collection for the design of automotive turbocharger centrifugal compressors.

1. Introduction
Modern turbocharger requirements are no longer orientated around maximum power output from the internal combustion (IC) engine. Designers are now focused on fuel efficiency and instant torque at low engine speeds. With the quantum shift from high power IC engine designs to small gasoline turbocharged engines, the design ethic is now completely different; map width is now the priority instead of maximum pressure ratio (PR).

Early centrifugal compressor design was focused around design point performance. In the automotive turbocharging application there is no single definitive operating condition, but instead there is an operating envelope. In an urban application the vehicle drive cycle could be described as inherently stop-start, in which case there is a need for high efficiency at low PR and low mass flow rates. Also crucial is the need for very fast response (instant PR) which leads to instabilities toward the surge side of the map. The compressor must also be able to deliver the maximum mass flow rate necessary to meet the engine requirements.

The design, and therefore performance, of modern automotive turbocharger compressor stages is very different to what one might refer to as classic industrial/process stages (with fixed operating points). The result is that the existing 1D off-design performance prediction methods have difficulty estimating state-of-the-art turbocharger compressor maps. Therefore, there is a need for further improvement of the current 1D methods with new performance targets in place.
2. Compressors

Comparisons made in this section are relative to typical automotive turbocharger compressor stages designed using the criterion described in the introduction; three centrifugal compressors are used and will henceforth be referred to as C-1, C-2, and C-3.

The Galvas [1] study had its origins at NASA in 1974. Galvas notes that the compressor is ‘built by a commercial engine manufacturer’, and gives an example of a turboshaft engine (it is reasonable to surmise that it is an aircraft engine centrifugal compressor). The compressor has a relatively narrow performance map with a peak efficiency of ~0.82 and peak PR of ~6.3. This suggests that unlike a turbocharger compressor, it has been designed for a particular operating condition. Geometry comparisons to small turbocharger compressors are presented in Table 1.

The Oh [2] study uses the three Eckardt impellers, and one referred to as the KIMM impeller. The Eckardt compressors were designed and manufactured in the mid 1970’s. As Eckardt’s Impeller ‘A’ is the backswept design, comparisons are made with this compressor in Table 1. Unfortunately the application of the KIMM impeller is unknown; however the same comparisons are drawn in Table 1.

From the Aungier [3] literature it is stated that the model has been validated using ‘more than a hundred different stages’, with a wide range of flow coefficients and pressure ratios (up to approximately 3.5). The applications of these compressors are reported to include ‘process compressors, air compressors, and turbochargers’.

2.1. Test Stages

In this study, three turbocharger centrifugal compressor stages are used for the comparison (Table 2). The chosen compressors represent some of the extremities of current design variations. The backsweep angles of C-2 and C-3 are expressed relative to C-1.

Variation between the compressor maps can be drawn from Figure 2. C-1 is the least backswept design, and has the highest pressure ratio. C-2 is the most backswept design and hence there is a penalty in maximum PR, however C-2 manages to shift the peak efficiency island to a lower PR and mass flow rate, and even though the impeller is 9% smaller in diameter, the same map width as C-1 is still achieved. C-3 is the largest of the three compressors with the highest \( \frac{D_{1s}}{D_2} \) ratio and therefore map width is much larger, but peak efficiency is sacrificed.

3. Modelling
Off-design performance can be modelled using several different techniques. These range from the different 1D modelling methods to 3D solving of the Navier-Stokes equations. Common 1D modelling techniques are Zero-Zone (statistical, Casey and Robinson [4]), Single-Zone (mean-line), and Two-Zone (Japikse [5]). 3D Computational Fluid Dynamics (CFD) techniques exist, most commonly solving a simplified set of Navier-Stokes equations coupled with a proven turbulence model.

3.1. 1D Modelling
The purpose of this study is to provide the compressor designer with knowledge of the limitations surrounding 1D Single-Zone modelling. Many authors/developers have amassed a collection of loss models to suit particular applications, and typically this works quite well. Empirical data is required to verify any model, and also to develop any improved empirical loss models.

3.1.1. Single-zone Modelling. The single-zone code used in this study is based upon fundamental fluid flow equations [5]. Impeller overall dimensions are input; inlet and exit blade angles, blade thickness (at the RMS diameter where relevant), and the number of main and splitter blades. This is supplemented with basic vaneless diffuser and volute geometry. The meridional shape of the impeller is developed using Bezier curves for the hub and shroud, and by inputting a blade sweep angle, the impeller flow length along the mean-line is estimated. The streamwise starting position of the splitter blades is also input, and is used to calculate an equivalent number of main blades using a method adapted from Aungier [3] (Eqn. 1):

$$Z_{eq} = Z_m + Z_{sp}(1 - X_{sp})$$  \hspace{1cm} (1)

where $0 < X_{sp} < 1$, and 0 represents the leading edge of the main blades.

The impeller solver iterates impeller outlet total temperature until mass flow is conserved. The vaneless diffuser is simulated using the Herbert [6] method, and the scroll volute is calculated using the Weber and Koronowski [7] method.

Galvas and Oh both use the Wiesner [8] slip factor, while Aungier [3] uses his own modified version of the Wiesner slip factor.

Gas properties and inlet flow conditions must be specified and the values used are typical ambient conditions as shown in Table 3. Relevant gas properties are assumed to be constant at all stages in the compressor and the inlet flow angle is always assumed to be axial.

| Table 3. 1D Model Inlet Flow Conditions. |
|------------------------------------------|
| **Flow Properties** | **Unit** |
| Inlet Total Temperature | 300 K |
| Inlet Total Pressure | 101325 Pa |
| Ratio of Specific Heats | 1.395 - |
| Gas Constant | 287 J/kgK |
| Inlet Flow Angle | 0 deg |
| **Viscosity Data** | |
| Reference Viscosity at $T_0$ | 18.27 µPas |
| Reference Temperature | 291.15 K |
| Sutherland’s Constant | 120 K |

| Table 4. Impeller Loss Model Collections. |
|------------------------------------------|
| **Loss** | **Galvas** | **Oh** | **Aungier** |
| **External** | | | |
| Recirculation | • | • | • |
| Disk Friction | • | • | • |
| Leakage | • | • | • |
| **Internal** | | | |
| Incidence | • | • | • |
| Skin Friction | • | • | • |
| Blade Loading | • | • | • |
| Clearance | • | • | • |
| Exit Mixing | • | • | • |
| Shock | • | • | • |
| Choking | • | • | • |
| Distortion | • | • | • |

3.1.2. Impeller Loss Models. Three complete collections of loss models from three different authors have been chosen (Table 4). Loss models are divided into two groups; internal and external. Internal
losses are those which contribute to a reduction of the ideal total pressure rise predicted when using the Euler work equation. External losses contribute to a rise in total temperature over and above the ideal rise suggested by the Euler work equation but do not contribute to the desired outcome of pressure rise.

Three individual loss model collections are used. The loss models contained in each of these are detailed in Table 4. All of the losses noted here and used in the current Single-Zone model are taken directly from the literature. Although all three loss collections may make estimations of the same overall loss, they do not necessarily use the same correlation.

3.1.3. Compressor Surge. Oh [2] does not report on how surge and choke are predicted. Galvas [1] suggests using the Lieblein diffusion factor as an impeller stall criterion, while Aungier [3] suggests his own diffusion ratio based on the impeller relative throat velocity and exit relative velocity. Galvas also provides an empirically derived relationship which defines compressor surge based on vaned diffuser performance. However, the majority of automotive stages operate with vaneless diffusers, meaning this correlation is not applicable in this study.

For the current work, compressor surge is assumed to be driven by the impeller diffusion ratio as defined by Eqn. 2.

\[ DR = \frac{W_{1\xi}}{W_2} \]  

Figure 1. Impeller DR at Surge for each Speedline for C-1.

In order to define a value for \( DR \) at which surge occurred, a single-passage CFD model was produced to model C-1. A RANS simulation was carried out using ANSYS CFX and the SST turbulence model provided with it. The model consisted of stationary inlet and vaneless diffuser domains and a rotating domain for the impeller with frozen rotor interfaces between them. It contained approximately 1.6e6 cells and had a \( y^+ \) of less than 2 in the majority of the domains. Convergence was defined as the RMS residuals falling below 1e-4; imbalances in mass, momentum and energy reducing below 0.01%, and the efficiency fluctuating within \( \pm 0.05\% \). Surge was defined as the lowest mass flow rate for a given rotational speed where the convergence criterion could no longer be met.

As seen in Figure 1, the impeller \( DR \) at which surge occurred remained relatively constant at approximately 2.4 until higher tip speeds. At \( Mu_2 = 0.9 \) a transition region is seen, indicating that another component other than the impeller is the limiting factor. At the two highest tip speeds the limiting impeller \( DR \approx 1.5 \). This can also be identified in Figure 2 since the surge flow rate for C-1 is considerably higher at the higher tip speeds. For this study the impeller is assumed to be the only component driving surge conditions and so a value of \( DR = 2.4 \) has been selected.

3.1.4. Compressor Choke. Galvas [1] suggests predicting choke using a correlation proposed by Rodgers [9], in which the choking mass flow rate is a function of relative Mach number and blade
angle at the inlet RMS diameter. Aungier [3] has a choking loss (Table 4) applied which inevitably means that the efficiency will decrease rapidly, leading to numerical instability.

In this study, choke was predicted using a combination of two methods. At relatively low tip speeds the compressor is never truly choked before a pressure ratio of unity is achieved, whereas at relatively high tip speeds, the compressor is definitely choked as no further increase in mass can be passed. The maximum passable mass flow rate appears to increase with tip speed; this is explained by Dixon [10]. Dixon’s method was used to calculate the theoretical area at which choking occurs (Eqn. 3). The choked area was then compared to a throat area as estimated by Eqn. 4.

\[
A^* = \frac{\dot{m}}{\sqrt{\frac{2 + (\gamma - 1) \frac{U_k^2}{d_{01}^2}}{(\gamma + 1)}}} \left( \frac{\gamma+1}{2(\gamma-1)} \right) \tag{3}
\]

where \( a_{01} = \sqrt{\gamma R T_{01}} \)

\[
A_{th} = \frac{\pi}{4} \left( D_{1s}^2 - D_{1h}^2 \right) \cos \beta_{1b} - 0.5 \tau_{1b} Z_m (D_{1s} - D_{1h}) \tag{4}
\]

If the calculated choking area is less than the throat area, the stage is assumed choked. Also if the total-total isentropic efficiency drops below 0.4, the stage is assumed choked. Typically the efficiency is used at low tip speeds, and the throat check is used at high tip speeds.

4. Data Comparison

Predicted maps are compared here with the corresponding hot gas stand test data which has been corrected for the effect of heat transfer using the method of Casey and Fesich [11]. The correction procedure of C-1, C-2, and C-3 is described by Harley et al. [12]. All test data has been provided by IHI Charging Systems International; it has been collected in accordance with SAE J1826 [13] using a standard 2-loop hot gas stand.

Comparisons with the efficiency can contain errors due to the inevitable heat transfer involved in turbocharger gas stand data along with the assumptions of the heat transfer correction method used. It is still reasonable to compare the efficiency trends, however the absolute error between the corrected efficiency and the predicted efficiency should be treated with some care.

All data is non-dimensionalised using the maximum values (\(PR, \dot{m}, \dot{E}_t\)) found from the test data and the corresponding predicted data. Each compressor is assessed in turn, and the different loss model collections benchmarked against each other. It is important to note that flow range is being predicted in this study, as described previously. For all predictions, a value of \(DR = 2.4\) is used to predict the surge point as described earlier. The \(PR\) presented is total-total across the stage, and the efficiency is corrected isentropic total-total efficiency.

4.1. Compressor C-1

The PR predictions from the Oh and Galvas models are impressive for C-1 (Figure 3). As tip speed increases, some accuracy is lost in the Aungier and Galvas (Figure 3) predictions, but the Oh method predicts the PR very well at \(N_{max}\). The Aungier prediction for C-1 performs well in two regions, low tip speeds toward surge, and higher tip speeds toward choke. Accuracy is lost toward the choked side of the map at higher tip speeds in the Oh and Galvas models as there is no contraction of the throat area simulated; thus the choked mass flow is overestimated. The Aungier model predicts the choking flow rate at \(N_{max}\) with good accuracy. The flattening of the PR characteristic curve toward surge is not predicted by any of the models.
The efficiency is underestimated by all models although the predicted peak efficiency trends of all predictions are similar to C-1 by predicting peak efficiency to be at the highest tip speed. The Aungier prediction of efficiency for C-1 is the most accurate.

It is clear that $DR = 2.4$ across the impeller is a good predictor of compressor surge up to approximately $(N/N_{max}) = 0.81$ in this case, beyond which a different instability in the compressor causes the surge point to deviate.

### Figure 2. Test Compressor Maps.

### Figure 3. 1D Model Predictions for C-1.

#### 4.2. Compressor C-2

The PR prediction toward the surge side of the map has poor accuracy for all three models. The Oh model (Figure 4) has good accuracy at the low tip speeds, but this is quickly diminished toward higher tip speeds. The Galvas model (Figure 4) has impressive accuracy up until $N_{nax}$ when some deviation occurs. The Aungier model (Figure 4) again shows poor PR prediction accuracy at low tip speeds toward choke, and high tip speeds toward surge. Similarly the Aungier PR prediction at $N_{max}$ is over-predicted.

The peak efficiency is underestimated by all three models. The peak efficiency in C-2 is in the middle of the performance map, and none of the models predicts this accurately. Such erroneous predictions of modern compressor efficiency trends highlight the weaknesses of the existing 1D loss models.

Using a $DR = 2.4$ still appears to be a reasonable estimation of the compressor surge point. However surge is now predicted at a higher mass flow than the empirical data. As with C-1, the surge line deviates toward high tip speeds, suggesting further compressor instabilities.

#### 4.3. Compressor C-3

All three models have predicted PR with impressive accuracy. The Oh model (Figure 5) has better accuracy throughout the map; however some accuracy is lost at the high tip speeds. The Galvas model (Figure 5) however has an accurate PR prediction at the high tip speeds, but sacrifices some accuracy through the rest of the map. The low tip speed PR predictions in the Aungier model (Figure 5) are good; however the high tip speed prediction is poor when compared with the others.
The peak efficiency trend of C-3 resembles that of C-2 with a peak in the centre of the map. Again, all three models under-predict the peak efficiency, and none of the models accurately predict the location peak efficiency.

A $DR = 2.4$ does not suffice in this case; C-3 can handle more diffusion in the impeller as would be expected by it having the highest $(D_1/D_2)$ ratio. Compressor choking for C-3 is still overestimated but potentially a more accurate prediction for both models than for C-1 & C-2. It appears from the test data that C-3 could have passed more mass flow as neither the efficiency nor PR characteristic curves drop off.

![Figure 4. 1D Model Predictions for C-2.](image)

![Figure 5. 1D Model Predictions for C-3.](image)

5. Conclusions

Three different loss model collections have been used to simulate the performance maps of three modern centrifugal compressors found in automotive turbochargers. The limitations of these loss model collections have been displayed relative to modern turbocharger compressors. The Galvas model performed best at high tip speeds for the compressors with a higher $(D_1/D_2)$ ratio although the Oh model performs best across the majority of the PR map for all three compressors. Trends in the loss of accuracy were noted, specifically:

- Flattening of the PR characteristic curve toward the surge margin;
- Changing gradient of the surge line at relatively high tip speeds;
- Peak efficiency in modern automotive compressors is in the center of the map.

The most robust loss model collection for use with centrifugal compressors typically found in automotive turbocharger stages is the Galvas collection; it has the best performance prediction at high tip speeds and has acceptable performance prediction across the remainder of the map, whereas the loss in accuracy when using the Oh collection at high tip speeds is less acceptable. Furthermore the Galvas collection uses the least number of loss models and hence equations, which also helps with the robustness of performance prediction program.

The loss models evaluated do not fully represent a loss in total pressure as seen in the test stages when the PR characteristic curve flattens toward surge. It could be suggested from this characteristic that an internal loss is not being properly accounted for, particularly in the region where one expects to find recirculation in the impeller passage.
The impeller diffusion ratio is shown to represent a good prediction of compressor surge at low tip speeds; however at higher tip speeds the surge point shifts to higher mass flow rates suggesting further component instability within the compressor. The impeller diffusion ratio of 2.4 is shown to be a good representation of compressor surge for stages C-1 and C-2, however C-3 can handle more diffusion and therefore a higher limiting diffusion ratio would be required. As for variation of the diffusion ratio from stage to stage, it appears to be related to the level of backsweep applied, and the $\left( D_{ls}/D_{2} \right)$ ratio.

Acknowledgements
The authors would like to thank IHI Charging Systems International for their technical support and provision of the required compressor geometry and test data. Thanks are also given to ANSYS Inc for the use of their software for numerical modelling in this research programme.

Nomenclature

| Symbol | Description         |
|--------|---------------------|
| $a$    | Acoustic Velocity (m/s) |
| $A$    | Area (m$^2$)          |
| $D$    | Diameter (m)         |
| $DR$   | Diffusion Ratio (-)  |
| $\dot{m}$ | Mass Flow Rate (kg/s) |
| $Mu$   | Tip Mach Number (-), $Mu_{2} = \frac{u_{2}}{\sqrt{PR_{2}}}$ |
| $N$    | Rotational Speed (rev/min) |
| $R$    | Gas Constant (J/kgK) |
| $t$    | Blade Thickness (m)  |
| $T$    | Temperature (K)      |
| $U$    | Blade Velocity (m/s) |
| $W$    | Relative Velocity (m/s) |
| $X$    | Meridional Position (-) |
| $Z$    | Number of Blades (-) |
| $\beta$ | Flow/Blade Angle Relative to Meridional (deg) |
| $\gamma$ | Ratio of Specific Heats (-) |
| $\eta$ | Total-Total Isentropic Efficiency (-) |
| $\pi$ | Total-Total Pressure Ratio (-) |
| $\rho$ | Density (kg/m$^3$)    |
| PR     | Pressure Ratio (-)   |

Abbreviations

| Abbreviation | Description |
|--------------|-------------|
| IC           | Internal Combustion |

Subscripts

| Subscript | Description |
|-----------|-------------|
| 0         | Total Conditions |
| 1         | Impeller Inlet |
| 2         | Impeller Exit |
| b         | Blade |
| eq        | Equivalent Number |
| h         | Hub |
| m         | Main Blades |
| max       | Maximum |
| p         | Peak |
| s         | Shroud |
| sp        | Splitter Blades |
| th        | Throat |

Superscripts

| Superscript | Description |
|-------------|-------------|
| *           | Choked Condition |

References

[1] Galvas M R 1974 *Fortran Program for Predicting Off-design Performance of Centrifugal Compressors* (Cleveland: NASA Lewis Research Center)
[2] Oh H W, Yoon E S and Chung M K 1997 *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy* 211(4) 331-8.
[3] Aungier R H 2000 *Centrifugal Compressors: A Strategy for Aerodynamic Design and Analysis* (New York: ASME Press)
[4] Casey M V and Robinson C J 2011 A Method to Estimate the Performance of a Centrifugal Compressor Stage *Proceedings of ASME Turbo Expo 2011: Power for Land, Sea and Air, (Vancouver, Canada, 6-10 June 2011)*
[5] Japikse D 1988 *Centrifugal compressor design and performance* (Vermont: Concepts NREC)
[6] Herbert M V 1980 *A Method of Performance Prediction for Centrifugal Compressors* (London: HMSO)
[7] Weber C R and Koronowski M E 1986 Meanline performance prediction of volutes in centrifugal compressors ASME, *International Gas Turbine Conference and Exhibit, 31st (Duesseldorf, West Germany, June 8-12, 1986)*
[8] Wiesner F J 1967 *Journal of Engineering for Power* 89 558
[9] Rodgers C 1964 *Journal of Engineering for Power* 86 161
[10] Dixon S L 1998 *Fluid Mechanics and Thermodynamics of Turbomachinery* (Oxford:
Butterworth-Heinemann)

[11] Casey M V and Fesich T M 2010 Journal of engineering for gas turbines and power 132(7) 072302

[12] Harley P, Spence S, Filsinger D, Dietrich M. and Early J 2012 An Evaluation of 1D Design Methods for the Off-Design Performance Prediction of Automotive Turbocharger Compressors Proceedings of ASME Turbo Expo 2012: Power for Land, Sea and Air, (Copenhagen, Denmark, 11-15 June 2012)

[13] SAE 1995 Turbocharger Gas Stand Test Code (Pennsylvania, USA: Society of Automotive Engineers)