

Optimal Pressure Based Detection of Compressor Instabilities Using the Hurst Exponent

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ABSTRACT

The compressor surge line of automotive turbochargers can limit the low-end torque of an engine. In order to determine how close the compressor operates to its surge limit, the Hurst exponent of the pressure signal has recently been proposed as a criterion. The Hurst exponent quantifies the fractal properties of a time series and its long-term memory. This paper evaluates the outcome of applying Hurst exponent based criterion on time-resolved pressure signals, measured simultaneously at different locations in the compression system. Experiments were performed using a truck-sized turbocharger on a cold gas stand at the University of Cincinnati. The pressure sensors were flush-mounted at different circumferential positions at the inlet of the compressor, in the diffuser and volute, as well as downstream of the compressor. Results show that the previously identified threshold value distinguishing between surge and stable operation when the analysis was carried out for a different and smaller compressor can be used also for this much larger compressor. The investigation concerning the sensor locations reveals that pressure sensors at the outlet or shortly upstream the volute tongue give the clearest distinction between fully stable operation and operation close to the surge line. Further investigations show that as currently implemented, the criterion would need a minimum sampling duration of 500 ms and sampling frequency of 512 Hz. An extended algorithm based on distinguishing between a mono- and multifractal pressure signal is shown to have potential as an early warning indicator.

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INTRODUCTION AND BACKGROUND

Turbo compressors are limited in their operating range at low mass flows by the surge phenomenon. Surge occurs when the compressor can no longer steadily supply the outlet pressure needed to overcome the flow resistances downstream. Typical flow resistances are throttles on a gas stand, or the throttle or intake valves on a turbocharged internal combustion engine. The mass flow and pressure increase through the compressor start oscillating. If the global mass flow becomes negative, the operation conditions is often called deep surge, while mild surge denotes an operation with oscillating, but always positive mass flow through the compression system. Due to the associated noise, uncertainty of the resulting volumetric efficiency, and mechanical stress on the components in the intake system, surge should be avoided on turbocharged combustion engines. It therefore limits the low-end torque in boosted engines.

A common way to model surge is to model the compression system analogously to a mass-spring-damper system. This idea was first introduced by Greitzer [1, 2] for axial compressors, but has been shown to also apply to radial compressors in subsequent studies [3, 4]. In this analogy, the compression system consists of air mass oscillating in a pipe (the mass), connected to a compressible plenum

downstream (the spring). The compressor is represented by an actuator disk in the pipe (a damper with negative damping), and at the plenum outlet, there is a flow resistance (a damper with positive damping). A schematic is given in the top right corner of Figure 1. From this analogy, some predictions can be made: The surge frequency should approximately correspond to the eigenfrequency of the system, i.e. the Helmholtz frequency f_H in this case. It is given by

$$f_H = \frac{a}{2\pi} \sqrt{\frac{A_c}{L_c V_p}} \quad (1)$$

where a is the speed of sound, V_p is the plenum volume, L_c is the length of the compression system (including inlet and outlet pipes as well as the flow path through the compressor), and A_c is its cross-section area. Further, it can be estimated whether the system is stable or (dynamically) unstable using the criterion

$$\left(\frac{\partial \psi}{\partial \phi}\right)_c \leq \frac{1}{B^2} \left(\frac{\partial \psi}{\partial \phi}\right)_t^{-1}$$


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$$B = \frac{U}{2 \cdot 2\pi f_H \cdot L_c} \quad (3)$$

where $\phi = \dot{m}/(\rho A U)$ and $\psi = \Delta p/(0.5\rho U^2)$ are the mass flow and pressure coefficients, ρ is the density, U is the blade velocity, and indices C and T denote compressor and throttle, respectively. Since the throttle slope is relatively steep, and B is of the order of magnitude 1, one can expect some oscillations on the positively sloped parts of the compressor speedline. Figure 1 shows the distinction between a stable and a dynamically unstable system in a schematic compressor map.

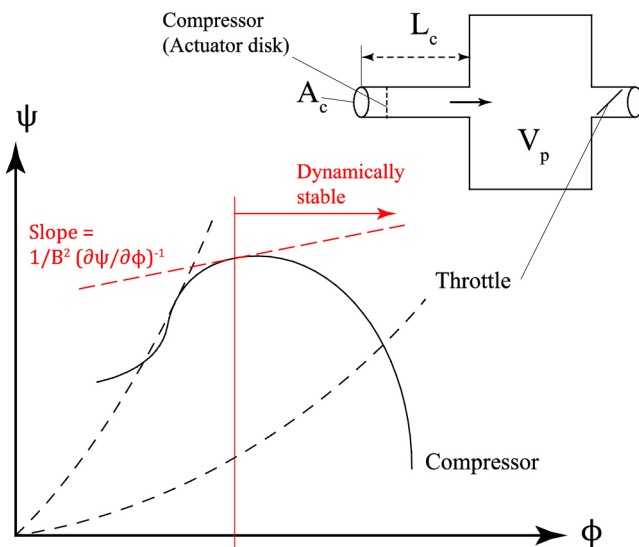


Figure 1. Compression system as a Helmholtz resonator (top right), and a schematic of stable operation and dynamically unstable operation (adapted from [1] and [5]).

In order to classify compressor operation as surge, different indicators have been proposed in literature. Many are based on the compressor outlet pressure signal. Subjective noise as experienced by the operator is one of the criteria used in [6]. The standard deviation of the signal was used e.g. in [6], [7] and [8]. It can be calculated based on the raw signal or based on a lowpass filtered signal that removes higher frequency oscillations; further, one can normalize the standard deviation, e.g. with the impeller dynamic pressure $0.5\rho U^2$, in order to obtain similar results for different speedlines. The height of the peak in the linear frequency spectrum, obtained by a Discrete Fourier Transform (DFT), was proposed in [9]. The signal power in a certain frequency band around the surge frequency - essentially the variance of the signal at these frequencies - has also been used, e.g. in [10], [11], and [12]. It was shown that the acoustic sound amplification of the compressor at low frequencies increases strongly as the compressor approaches surge in [13]. Recently, the Hurst exponent has also been proposed as a criterion in [12]. It is an indicator about the long-term memory for statistically similar time series data; more explanation will be given in the subsection *Detrended fluctuation analysis* in the *Method* section.

As an alternative to outlet pressure based indicators, the oscillations of the compressor speed can also be used, see e.g. [14]. For a compressor rig with electric motor instead of a turbine, the authors

in [14] also show that the power input to the electric motor has a steep increase at the onset of oscillations. Due to the backflow of hot gas from the impeller operating close to the surge line, the temperature increase at the inlet is also a commonly used surge indicator [6, 14].

This paper extends the work done in [12] on the applicability of the Hurst exponent as a surge criterion. The aims are threefold: First, to apply this methodology to a different compression system in order to gain confidence in the method; second, to investigate requirements of the pressure signal with respect to sensor position, sampling time, and sampling frequency; and third, to introduce another, more sensitive surge indicator based on the fractal properties of the signal.

The paper is based on experiments run on a truck-sized turbocharger compressor at the Gas Dynamic and Propulsion Laboratory at the University of Cincinnati. The section *Method* describes the theoretical background and the experiments. In its first subsection, the estimation of the Hurst exponent is shortly revisited. Then, the experimental setup is described. Results are presented in section *Results and Discussion*: First, the applicability of Detrended Fluctuation Analysis (DFA) for this compressor is established. The Hurst exponent as a surge indicator is compared to standard deviation for different pressure sensor locations along one speedline. Sensitivity studies are carried out with respect to the sampling duration and the sampling frequency. An additional Hurst exponent based surge indicator is introduced, which allows distinguishing between large amplitude oscillations and small amplitude oscillations. Finally, the investigation is extended to two other speedlines.

METHOD

Detrended Fluctuation Analysis

The Hurst exponent can be understood as a scaling relation for statistically self-similar data [15]. Given for example a random walk $x(t)$ and magnifying it by a factor c , i.e. rescaling the time axis by $1/c$, the data axis needs to be rescaled by a factor

$$x(t) \rightarrow \left(\frac{1}{c}\right)^H x\left(\frac{1}{c} \cdot t\right) \quad (4)$$

where H is the Hurst exponent. H depends on the underlying distribution of the step size of the random walker. If the step sizes are uncorrelated (white noise), the Hurst exponent is $H = 0.5$; this is another way of saying that the expected displacement of the random walker scales with the square root of length of the walk. For a random walker with a long-term positively autocorrelated step size, e.g. pink noise, $H > 0.5$ [15]. Turbulent pressure fluctuations exhibit a long-term positive autocorrelation.

The principal idea in applying this methodology to compressor surge is that the pressure signal consists of turbulent pressure fluctuations and signal noise at stable compressor operation, while it is more wave-like near surge, with a frequency corresponding to the eigenfrequency of the compression system. In this case, one can expect the pressure oscillations of the system natural frequency to be

governed by a scaling law with $H \geq 0.5$ in stable operation. In pure mild or deep surge, on the other hand, $H \approx 0$, because the pressure signal is repeating periodically. Thus, the signal variance is independent of the time scale as long as the time scale is larger than one wave period. In between these two extremes, one can expect a decline of the Hurst exponent from the stable regime $H \geq 0.5$ downwards.

There are different ways to estimate the Hurst exponent from data. In this paper, DFA is used due to its flexibility. It was first introduced by Peng [16]. Alternatives include the original rescaled range approach [17], or via a Fourier transform [18]. A step-by-step explanation of how the Hurst exponent can be calculated for some artificial signals is given e.g. in [15, 12]. The main steps will be summarized in the following.

The first step is to calculate a cumulative time series of the signal in question,

$$Y_j = \sum_{i=1}^j x_i \quad (5)$$

The cumulative time series is then divided into windows of length t_s , and in each window w , the data is detrended,

$$y_w = Y_w - v_w \quad (6)$$

where v_w is a least square fit of the data in the window. In this case, detrending of order 1 (DFA1) is applied, meaning that the mean and the linear trend in each window are removed. Higher order DFA is also possible, but not used in this analysis. The fluctuation function F^2 is then the root mean of all window variances after detrending,

$$F^2(t_s) = \sqrt{\text{mean}(\text{var}[y_w])} \quad (7)$$

and a function of the window length t_s . This procedure is performed for many different t_s , which are logarithmically increasing. The Hurst exponent is then the slope of the fluctuation function F^2 over the window length t_s in a log-log diagram. It is important to note here that this slope is not generally constant for all window lengths, and thus, the Hurst exponent is dependent on the time scale over which it is calculated.

Besides the “standard” Hurst exponent, other orders of the Hurst exponent can be calculated by using a different order root mean instead of the square root mean as given in Eq. (7), see e.g. [15]. The fluctuation function of order q is then defined as

$$F^q(t_s) = \left(\text{mean} \left[(\text{var}[y_w])^{\frac{q}{2}} \right] \right)^{\frac{1}{q}} \quad (8)$$

The Hurst exponent of order q , H^q , is the slope of F^q in a logarithmic plot against window size t_s . For $q = 2$, Eq. (8) results in Eq. (7), and the “standard” Hurst exponent will henceforth be referred to as H^2 . For high values of q , the windows with a high variance are weighted heavily in the averaging procedure, and thus dominate the estimate of the Hurst exponent. For negative values of q , windows with small variance are weighted heavily. This method allows treating large amplitude and small amplitude fluctuations differently. A time series is then defined as “monofractal” if the Hurst exponent is independent of its order, $H^q \neq f(q)$. If $H^q = f(q)$, on the other hand, the time series is called “multifractal” [15].

Experimental Setup

The turbocharger used in this study is a Honeywell GT40, a truck-sized turbocharger. In the following, a short summary of the relevant features for this investigation will be given; for a detailed description of the turbocharger and the experimental setup, the reader is referred to the works by Guillou [8] and Gancedo [7].

The compressor has 10 full blades and an exducer diameter of $D^2 = 88$ mm. It inducts air via a bell mouth from the test chamber. Ports in the shroud serve as a passive flow device to stabilize compressor operation near the surge line. The impeller is followed by a vaneless diffuser and a volute. A pipe of length $L = 1780$ mm, whose diameter is $D = 63.5$ mm in its first half, and $D = 50.3$ mm in its second half, connects the compressor outlet to the pneumatic valve downstream. The flow is then guided to an orifice plate, which is instrumented for mass flow measurements.

This compression system has no compressible plenum between the compressor and the valve downstream that can clearly be distinguished from the outlet pipes. The system shows typical mild surge behavior in certain parts of the operating range. It does not, however, exhibit the classic deep surge cycle of quiescence, instability growth, blowdown, and recovery as described by Fink [3]. Instead, at a low enough mass flow, the (time-averaged) total pressure ratio drops significantly (see also Figure 10 later in the paper), and the oscillation amplitude increases. Guillou ([8], p. 237 ff.) estimated the instantaneous mass flow to be negative for parts of the surge cycle from the stereo particle image velocimetry data. This operating regime will therefore in the following be denoted as deep surge, and the term surge line denotes the mass flows where the switch between the mild surge like and deep surge behavior occurs.

The turbine that drives the compressor uses electrically heated pressurized air. The maximum air mass flow supplied by the facility is 0.26 kg/s at a gauge pressure of 8.3 bar. The air inlet temperature to the turbine was chosen to be around 200 degree C.

Since the compressor inducts air from the test chamber through a bellmouth, inlet total temperature and pressure are assumed to be equal to the respective test chamber values. The total temperature is measured by a type T thermocouple ca. 300 mm upstream the bellmouth, and the total pressure in the test chamber is measured with an atmospheric barometer. Compressor outlet conditions are measured by a Druck PDCR 130/W/C pressure transducer and a type K thermocouple. The compressor speed is obtained using a center