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EXPERIMENTAL STUDY OF THE SURGE BEHAVIOUR OF A CENTRIFUGAL COMPRESSOR

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ABSTRACT

Turbo charging is a way for car manufacturers to meet the major challenge of fuel economy and reduction of the CO2 emissions. The aim is to replace current engines with a downsized engine in order to improve its fuel consumption. This requires higher specific power and torque especially at very low engine speed.

The wide functioning area required for this technique leads to an operation of compressors at low air flow rate and high compression ratio characterised by the occurrence of surge phenomenon. On the engine, it can be intensified by the fuel flow rate pulses consecutive to air flow rate pulses leading to torque oscillations. In fact, surge is an aerodynamic instability that affects both the compressor and its associated network which results in strong pressure and mass flow rate oscillations describing a surge cycle in the compressor map. The surge limit of a centrifugal compressor and its frequency are highly dependant on the geometrical configuration of its associated network.

In this paper, the experimental work was focused on the study of the surge behaviour of a centrifugal compressor by means of a turbocharger test bench and the influence of the upstream and downstream circuit (volumes sizes and pipes lengths) and their interaction at the surge limit, the frequency and the unsteady fluctuations amplitude of this phenomenon. A comparative study of our experimental results with the Greitzer model was also done in terms of surge frequency and intensity prediction.

Keywords: Centrifugal compressor; Turbocharger; Surge; Compressor circuit; Greitzer model

INTRODUCTION

During the use of turbocharged Diesel engines at low flow, the compressor could operate into his prohibited surge area. This occurrence may induce dangerous instabilities which result in mechanical and heat loads that could damage the whole turbocharged engine [1]. In practice, avoiding this risk is done by reducing the boost pressure at low engine speeds and hence reducing the engine torque.

In fact, surge is an aerodynamic instability characterized by fluctuations in the average flow through the whole compression system. The mass flow fluctuations at surge can be so large that during a short part of the surge cycle the mass flow becomes negative. The oscillatory loading and unloading of the compressor in surge imposes transient loads on the bearings supporting the compressor shaft and can have disastrous effects on the whole compression system. Furthermore, surge produces mechanical vibrations and is usually clearly audible.

The surge behaviour of compressor is defined as a function of, not only the compressor design, but also of the geometry around of compressor [2-3]. It is well-known that a bad design of the compressor inlet may induce surge [4-5]. Depending on the pressure oscillation levels, the turbocharger can be damaged when operating in deep surge [6].

Therefore, compression system models were developed. The oscillations in the system are modelled in a manner analogous to those of a Helmholtz resonator [7]. This kind of modelling
has an analogy with electrical and mechanical systems. The assumptions of a Helmholtz resonator imply that all the kinetic energy of the oscillations is associated with the motion of the gas in the compressor and ducts. The potential energy of the gas is associated with the compression of the gas in the plenum. Furthermore, the assumptions are that the inlet Mach number is low and that the pressure rises are small compared to the ambient pressure.

Greitzer has developed a lumped parameter model for axial compressors, but Hansen et al. [9] showed that the model is also applicable to a centrifugal compression system. This publication was followed by more studies that focused on the analysis and modeling of centrifugal compression system dynamics. A simple, but relevant advancement in this evolving field was made by Fink [10] who included simple rotor dynamics in the Greitzer model to account for the effect of speed variations on system transients. The paper by Greitzer and Moore [11] marks another major development in the modeling of compression system dynamics. This last one presents the derivation and analysis of a nonlinear dynamic model that describes the growth and possible decay of a rotating stall cell during compressor transients, the development of surge, and the possible coupling between the two instabilities. While surge is considered as an unsteady axisymmetric oscillation, rotating stall is a steady flow variation in both axial and circumferential direction. To capture both phenomena the so-called Moore-Greitzer model is formed by coupling two-dimensional unsteady flow descriptions to a lumped-parameter system model.

**EXPERIMENTAL FACILITY**

The facility used in this work is based on a turbocharger test cell as presented in Fig. 1. The turbocharger used for these experiments is a Honeywell TurboTechnologies (GARRETT). The compressor is driven by the turbine which expands compressed air stored at 20 bars and ambient temperature in an important plenum. The test bench is instrumented as follows:

- Upstream and downstream compressor pressure: two piezo-electric transducers from Kistler (0-2.2 bar and 10 kHz).
- Upstream and downstream compressor temperature: two thermocouples.
- Air mass flow rate: a differential pressure gauge is used with a diaphragm (1-100 mbar 1 kHz).
- The compressor rotational speed is measured by an optical fiber.

![Fig.1 Turbocharger test bench](image)

The test procedure is to set the compressor speed, the downstream valve being initially wide open and it is gradually shut off to reduce the mass flow rate, until getting closer to the surge area. The surge limit is reached just before the apparition of a surge loop in substitution to the...
operating point shown, in real time, on an interface software for data acquisition. This technique is in good agreement with the characteristic noise emitted by the compressor whose frequency and level are highly altered during surge. For all tested configurations, are presented (Fig.2):
- The pressure ratio versus mass flow-rate characteristic for 3 positions of the downstream valve (out of surge, at the surge limit, in the surge area)
- The spectral analysis of the outlet pressure and of the differential pressure

![Graphs](image)

Fig.2- Compressor’s operating point and its spectral analysis (out of surge, on the surge limit, in the surge area)

The objectives of these experiments are to highlight the mechanisms important in the surge occurrence, and to characterize the surge cycles in terms of amplitude and frequency of the non steady pressure ratio and mass flow-rate fluctuations. We expect that the geometrical configuration of the downstream or upstream “line” has an effect on the compressor surge line position and on the deep surge loop size. The influence of the geometrical configuration is studied by a modification of the pipe length and variation of a plenum volume. This plenum may represent for example, the volume of the air filter (compressor upstream) or the air cooler (compressor downstream). Another specificity of this test line is that the pipes are straight, without any elbow, in order to minimize the aerodynamic phenomena that may influence the surge line characteristics. The mass flow-rate is regulated by means of the downstream valve. Experiments were done at rotational speeds of 60000 and 80000 RPM. These experiments are based on testing separately the effect of volume and ducts, on upstream and downstream line, by changing the length of duct (440, 700 and 1000 mm) or the plenum’s volume (2, 5 and 20 dm$^3$), and then making arrangements for the upstream and downstream resumed in Tab.1
### Arrangements of upstream and downstream line

<table>
<thead>
<tr>
<th></th>
<th>Upstream circuit</th>
<th>Downstream circuit</th>
</tr>
</thead>
<tbody>
<tr>
<td>AR1</td>
<td>Volume 5 dm$^3$</td>
<td>Volume 5 dm$^3$</td>
</tr>
<tr>
<td>AR2</td>
<td>Volume 5 dm$^3$</td>
<td>Conduit 700 mm</td>
</tr>
</tbody>
</table>

### INFLUENCE OF THE UPSTREAM AND DOWNSTREAM CIRCUIT ON THE SURGE CHARACTERISTICS

**PLENUM’S VOLUME EFFECT**

Fig. 3 treats the effect of the upstream and downstream plenum volume on the surge appearance. This figure shows the average mass flow rate at surge limit measured for the two compressor rotational speeds. The upstream volume of 2 dm$^3$ has the lowest mass flow rate compared to other volumes and is close to the surge limit mass flow of the first arrangement AR1. We can see, by comparing the surge limit mass flow of the two arrangements, that by adding a 700 mm duct length instead of 5 dm$^3$ volume how we can lose on the compressor operating range by increasing the surge limit mass flow rate.

In the case of the upstream circuit, the lowest volume remains the worst case on the gain on the surge limit with an advantage for the arrangement AR1.

![Fig. 3 Effect of the upstream (a) and downstream (b) plenum’s volume on surge limit](image)

Figure 4 brings surge frequencies extracted from the Fourier analysis of signals of pressure and flow rate. The upstream volume 5 dm$^3$ has the lowest surge frequency. This observation for this volume is also valid for the oscillations of the compression ratio and flow. We can observe that even if the surge with an upstream volume of 2 dm$^3$ appears later than the other volumes, it is more intense in terms of frequency and surge loop size. Surge frequencies decreased with the volume of downstream capacity. They are generally lower on tests at high speed. The surge frequency could be an indication on the number of surge cycles described by surge loops. Thus, the higher the frequency is, the most quickly we brought the compressor back into a new surge cycle.
Surge is obviously a phenomenon that occurs at low frequency. To judge its intensity for a test configuration, we should associate the frequency peak corresponding to the amplitude which gives an idea of the spectrum energy.

**Fig.4 Surge frequencies for the upstream (a) and downstream (b) plenum’s volume**

**DUCT LENGTH EFFECT**

The effect of the upstream and downstream duct of the compressor is presented on Fig.5. We don’t notice a significant difference in the appearance of surge with the test of the upstream duct, contrarily to the test of the downstream duct, marking an advantage for the 700 mm duct at 80000 rpm and for 1000 mm at 60000 rpm. The test with arrangement AR2 shows an occurrence of the phenomenon in a more important flow and so a loss on the compressor operating range out of the surge area. Fig.6 shows that the surge frequency tends to decrease with the size of the downstream or upstream duct according to the results of the literature. We remind that this analysis is directed in the sense to optimize the architecture of the circuits associated with the compressor in order to extend the operating range of the turbomachine and to improve the engine’s filling as well as the efficiency of the engine at low speed in the case of a supercharged engine by a turbocharger.

**Fig.5 Effect of the upstream (a) and downstream (b) duct’s length on surge limit**
EXPERIMENTAL INVESTIGATION OF THE GREITZER MODEL

The Greitzer model, shown in Fig.7, defined the so-called Greitzer parameter ($B$) that relates the Helmholtz frequency and the surge dynamics \[8\]. The $B$ parameter and Helmholtz frequency $\omega_h$ are defined as:

\[
B = \frac{U}{2a} \sqrt{\frac{A_c}{V_p L_c}}
\]

\[
\omega_h = a \sqrt{\frac{A_c}{V_p L_c}}
\]

In this section, we evaluated the developed compressor model by comparing its results with data from actual surge measurements (dynamic pressure and mass flow oscillations with its FFT analysis). This procedure was carried out for different values of $V_p$ and we selected the smallest compressor characteristic duct length that gave satisfactory results.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor duct length $L_c$</td>
<td>1.03</td>
<td>m</td>
</tr>
<tr>
<td>Compressor duct area $A_c$</td>
<td>0.00212</td>
<td>m$^2$</td>
</tr>
<tr>
<td>Plenum’s volume $V_p$</td>
<td>2, 5 and 20</td>
<td>dm$^3$</td>
</tr>
</tbody>
</table>

Tab. 2 Experimental parameters of the compressor test rig

Fig.8 summarizes the evolution of the Greitzer parameter depending on the tested volumes and on the compressor rotational speed. This evolution marks that the surge phenomenon is the most favored instability for these different volume tested ($B > 1$).

In accordance with the model, by increasing $B$, we observe a modification in the shape of the surge loops and the frequency of the oscillations for these geometries.
On the other hand, Greitzer considers that surge will be more violent as far as the volume of the plenum is bigger, which is not the case according to the experiments on our test bench, where the oscillations of the flow and the pressure are not necessarily more important for the big volumes (Fig.9).

![Graph](https://via.placeholder.com/150)

**Fig.8 Evolution of expeimental lumped parameter B**

Also according to Greitzer [8], the surge cycle takes the shape of a kind of ellipse. The frequency of the phenomenon of classical surge seems to correspond to the frequency of the oscillator of Helmholtz. The result of this model saying that the most the volume of the tank is important in front of the compressor, the most the surge is privileged versus to the rotating stall and more the surge frequency is low is verified for most of configurations of the tested volumes. But, Fig.9 shows that the surge frequencies extracted from the FFT analysis (Fast Fourier Tranformation) don’t coincide with the frequency of Helmholtz.

![Graph](https://via.placeholder.com/150)

**Fig.8 Mass flow (a) Pressure ratio (b) oscillations for the tested volumes**

The experimental surge frequencies are slightly underestimated with regard to the frequency of Helmholtz associated with the receiving circuit, 24 and 22 Hz for respectively 60 000 and 80 000 RPM against 55,14 Hz for the Helmholtz frequency with a 2 dm³ plenum volume. The results of this model seem more uncertain in the neighborhood of null and negative flow
rates. Hansen [9] asserts that these disparities are due to more important effects of compressibility of the flow in this dysfunction zone.

Indeed, the fact that the model does not take into account characteristics of the system of compression (interactions rotor / stator, aerodynamic slowness and geometry of the line associated to the compressor) limits its application field. In practice, it is impossible with this simple model to exactly determine the maximum of the pressure ratio and thus the point of surge limit like its frequency. Moreover, we have shown that by modifying the geometry of the upstream compressor’s line and by adding a downstream geometry like it was done for the arrangement AR1, this model does not predict the surge characteristics.

The lumped parameter models are not suitable for describing phenomena associated with the distributed nature of fluid flows like acoustic waves and flow pulsations. In fact, the damped pressure oscillations, occurring after each flow reversal, are the result of acoustic waves traveling back and forth in the discharge piping [12]. In particular for large industrial systems these phenomena can have a significant influence on the dynamic surge behavior for centrifugal compressors.

CONCLUSION
In this paper, we present a parametric study dedicated to the surge of centrifugal compressor for Diesel automotive application. The experiments carried out underline the effect of the geometrical characteristics of upstream and downstream compressor line on surge occurrence as well as on the characteristics of the deep surge. The parametric variation of the geometric characteristics was conducted either by modifying the length of the pipes or the plenum volumes. An experimental investigation of the Greitzer model was carried out showing that this lumped parameter model is unable to predict accurately the surge characteristics since it ignores the acoustic effects of the piping system, impedance and resonance frequencies of the compression system, and also the aerodynamic behaviour of the flow.

Based on the above findings, we conclude that the lumped parameter model should be extended to take into account the acoustic phenomena in the piping system and even though the compressor dynamics.
REFERENCES