Modelling of surge phenomena in a centrifugal compressor: experimental analysis for control

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1. Introduction

The industrial plants that are using rotating systems, such as centrifugal compressors and rotating machinery, are recognised and proved to be less profitable, especially due to the instability surges, the axial movement or the vibration phenomena. These are considered practically to be the cause of 20–25% of emergency stops in registered facilities (Guemana, Aissani, & Hafaifa, 2011; Hafaifa, Daoudi, & Guemana, 2011; Hafaifa, Guemana, & Daoudi, 2012; Hafaifa, Rachid, & Mouloud, 2013; Marelli, Carraro, Marmorato, Zamboni, & Capobianco, 2014).

Several works have estimated the risk for the onset of surge and the consequences for the operation of the machine with more or less empirical analytical models (Galindo, Serrano, Climent, & Tiseira, 2008; Gravdahl, Egeland, & Vatland, 2002; Hafaifa, Djieddi, & Daoudi, 2013; Jiang, Khan, & Dougal, 2006; Pinarbasi, 2008; van Helvoirt & de Jager, 2007). However, the nature of this instability, that is, the basic mechanisms that are responsible, was not well identified. The study of these phenomena remains complicated due to the difficulties caused by the destructive instabilities. This work aims to understand the mechanisms involved in a centrifugal compressor during the unstable regime. The tool used is a model of compressor surge with the identification of precursory phenomena instabilities. In parallel with the numerical study, this work focuses on the experimental study of the surge phenomenon, in order to compare the simulation results together with experimental data obtained for the examined compression system. The proposed work offers advantageous performance in the modelling of the examined system.
2. Gas compression system

The compressor studied in this work is a centrifugal compressor of the SC3 Sonatrach station DJELFA, Algeria, fabricated by SOLAR turbine; the industrial plant using this compressor is shown in Figure 1. Each compressor stage consists of an impeller without a lid having main blades and spacers that are lying back, followed by a radial diffuser with an axial diffuser for both blades. Intercropping blades are located at the inter-blade mid-channel.

The SC3 Sonatrach station is equipped with a control room and computer equipment, which has enabled us to record the input and output measures of the examined compression system in order to model the system in real time using the implemented data station on-site.

The compression ratio, $P_{re}/P_{as}$, and efficiency, $\eta$, are the key measures of performance of a centrifugal compressor and must be submitted according to the considered operating point defined by the flow, $\dot{m}$, the rotation speed of the wheel, $N$, the suction conditions or power (total pressure, $P_{t1}$, and temperature, $T_{t1}$) and the gas nature ($R$, $\gamma$ and $\mu$). The results of the dimensional analysis, including the definition of dimensionless variables, allow for generalised comparisons between machines; such analyses have been widely discussed in the literature (Cunha, de Souza, Barreto, & de Souza, 2009; Galindo, Arnau, Tiseira, & Piqueras, 2010; Hafaifa, Guemana, & Daoudi, 2013a; Halimi, Hafaifa, Bouali, & Guemana, 2014; Utamura, Fukuda, & Aritomi, 2012). We simply retain the following dimensionless variables to describe the operating point:

$$\dot{m}_{red} = \frac{\dot{m}\sqrt{T_{t1}/T_{ref}}}{P_{t1}/P_{ref}}, \quad (1)$$

$$N_{red} = \frac{N}{\sqrt{T_{t1}/T_{ref}}} \quad (2)$$

Thus, these variables are defined for homogenous flow and the speed of rotation, respectively. These “reduced” quantities can be easily shown to correspond to the flow and speed that would yield the compression ratio and efficiency (the Reynolds meadows effect) for a machine operating at standard temperature and pressure (Paduano, Valvani, Epstein, Greitzer, & Guenette, 1994; Willems, Heemels, de Jager, & Stoorvogel, 2002). The evaluation of the performance of a compressor is generally not limited to only one point of operation, as demonstrated by the features in Figure 2. In this compressor field, the compression ratio is a function of the corrected speed constant and corrected rotation speed, on which the contours of performance are often superimposed.

The limit of the low flow is the surge, which is characterised by a flow instability (sometimes up to a reversal of the flow) together with large amplitude pressure oscillations, which may eventually damage the machinery. At high speeds, the limit is the block that corresponds to the appearance of a sonic section of the compressor stage. Both boundaries for incorporating an important measure of the performance of a centrifugal compressor are defined as the operating range

$$\text{Surge band} = \frac{\dot{m}_{blocking} - \dot{m}_{surge}}{\dot{m}_{normal}} \quad (3)$$

The flow blockage is sometimes used in the denominator of this expression; note that the range of a compressor decreases as the compression ratio increases.
2.1. Surge phenomenon

A rotating stall can suddenly evolve into a pumping mode. Characterised by an axial flow oscillation low frequency, its hysteresis is large, and escaping this regime is difficult (Robertson & Cameron, 1996). The mechanical stresses generated are very violent and quickly destroy the machine. This situation occurs when the flow is insufficient for a given system. The area over which this phenomenon occurs is clearly defined in the characteristic diagram shown in Figure 2; the operation of the machine must remain below a "surge bond" specified by the manufacturer. However, the machines are equipped with anti-surge devices to provide for possible entries into this area at the time of regime change. A sudden increase in the downstream flow of the machine allows out of this particular regime. The surge is a single axial dimensional instability that affects the compression system as a whole. Figure 3 shows the onset of the surge phenomenon; if the flow decreases, the losses tend to increase in the compressor. When the operating point crosses the surge line, the machine can no longer provide sufficient power to counteract the adverse pressure gradients, and a low-frequency pressure wave that moves from the downstream to the upstream affects the flow. The frequency of the pressure wave generally varies between 2 and 50 Hz for compressors.

Large fluctuations of the mean flow can be observed when the machine is operating in such a regime, and these fluctuations can sometimes result in a total reversal of the flow. This phenomenon is very dangerous to the compressor due to the extreme fatigue of the supporting blades. This instability also showed a strong hysteresis. Often the only solution to escape a system surge is to dramatically increase the flow in the machine (or restart when possible). In the literature, three types of surge have been described depending on the characteristics of the compression system: the conventional surge, the deep surge and the modified surge.

2.2. Surge line

The surge line represents the limit beyond which the compressor cannot operate in a stable regime. It passes near the point where the pressure ratio is maximised; the shape of this curve can be determined using the coefficients of total pressure and flow:

$$\psi = \frac{\Delta P}{(1/2) \rho U_m^2},$$  \hspace{1cm} (4)

$$\varphi = \frac{V_z}{U_m},$$  \hspace{1cm} (5)

where $\psi$ is the coefficient of total pressure, $\Delta P$ is the variation of the total pressure through the compressor, $U_m$ is the peripheral rotor speed, $V_z$ is the axial component of the velocity and $\varphi$ is the flow coefficient.

The relative magnitudes of the point of onset of pumping, conservation of flow and pressure changes in the rate can be written as follows:

$$\frac{Q}{\varphi_s \cdot S} = \rho \cdot U_m,$$

$$\left(\frac{P_2}{P_1}\right) = \left(\frac{P_1 + \Delta P_s}{P_1}\right),$$  \hspace{1cm} (6)

where $P_1$ and $P_2$ are the total pressures at the inlet and output of the compressor, respectively, $Q$ is the flow compressor suction and $S$ is the inlet section of the machine.

By defining

$$B = \frac{1}{2S^2 \cdot \rho P_1} \left(\frac{\psi_s}{\varphi_s^2}\right),$$  \hspace{1cm} (7)
Figure 4. Surge line shape.

and replacing $\Delta P$, in Equation (6) with expressions (4) and (5), the following can be demonstrated:

$$\left(\frac{P_2}{P_1}\right)_S - 1 = B Q_s^2.$$  \hspace{1cm} (8)

The surge line is parabolic, as shown in Figure 4, when the rotational speed of the machine varies. Unfortunately, the flow, $\phi_s$, is difficult to estimate. Therefore, manufacturers have focused on defining pumping margins. The surge margin can be defined using the following equation:

$$M = \left(\frac{P_2}{P_1}\right)_S - \left(\frac{P_2}{P_1}\right)_{DP},$$  \hspace{1cm} (9)

where $S$ (surge) is the relative magnitude at the point of onset of the surge and DP (design point) is the variable related to the nominal operating point.

According to this definition, the surge margin may reach 20% for axial flow compressors of the turbojet and 10% for centrifugal machines. A good understanding of the physical mechanisms related to aerodynamic instabilities is an essential step in order to reduce the margins and improve the performance of compressors.

### 3. Surge modelling

The model studied by Moore and Greitzer shown in Figure 5 identifies the origin of the surge phenomenon; this model presents the dimensions of the tank, which are assumed to be large compared to those of the compression system (Greitzer, 1976; Hafaifa, Daoudi, & Laroussi, 2011; Moore, 1984; Moore & Greitzer, 1986; Paduano et al., 1994; Seralathan & Roy Chowdhury, 2013).

In the tank, the compressible flow is assumed to have zero velocity and uniform pressure, $P$. The valve located downstream controls the flow. In the compressor and canals, the Mach number is sufficiently low for the flow to be considered incompressible (Brøns, 1990). $S_c$ and $L_c$ are the dimensions (length and cross section) of the compression system, and $V_p$ is the volume of the tank. The basic relationship of the dynamic system is defined as follows:

$$\frac{d^2x}{dt^2} = \frac{F}{m},$$  \hspace{1cm} (10)

The compressor is assumed to act as a “plug”, and the interface between the reservoir and the compressor can move. Thus,

$$|F| = \int \Delta P \cdot dS,$$

$$m = \rho S_c L_c,$$

$$P \cdot V' = Cst \Rightarrow \frac{P}{V_p} = -\gamma \frac{\Delta V}{V_p} = -\gamma \frac{S_c x}{V_p},$$

where $S_c$ and $L_c$ are the dimensions (length and cross section) of the compression system, respectively; $V_p$ and $P_p$ are the volume and pressure in the reservoir, respectively; $x$ is the displacement; $P$ and $\rho$ are the pressure and density at the interface of the compressor of tank, respectively; and $\gamma$ is the report coefficients of the specific heat ($\gamma = c_p/c_v$).

Thus, Equation (12) is obtained

$$\frac{d^2x}{dt^2} = -\gamma \frac{S_c x}{V_p L_c}.$$

After defining $w_h^2 = -(\gamma \cdot S_c P_p / \rho V_p L_c)$ and $c^2 = (\gamma \cdot P_p / \rho)$, the previous differential equation for this system is reduced to the following:

$$\frac{d^2x}{dt^2} = -w_h^2 x,$$

where $w_h$ corresponds to the pulsation frequency of the Helmholtz resonance:

$$w_h = c \sqrt{\frac{S_c}{V_p \cdot L_c}}.$$
This model of the surge has the advantage of simply explaining the origin of a complex phenomenon. Thus, the stability limit and the surge frequency can be theoretically estimated. However, the scope of the model is limited because it does not take into account the characteristics of the compression system (rotor interactions/stator aerodynamic inertia and channel geometry). In practice, this simple model cannot accurately determine the maximum rate of pressure and, consequently, the onset of the surge. Experience has also shown that compressors do not necessarily exhibit an instability surge priority category. In fact, the majority of multi-stage machines are subject to the rotating stall phenomenon before the surge. Based on the Helmholtz resonance frequency, Greitzer identified a dimensionless parameter, denoted \( B \), which indicates the type of instability that may develop first. The Greitzer parameter is given by the following equation (Greitzer, 1976; Hafaifa, Laaouad et al., 2011; Moore, 1984; Moore & Greitzer, 1986; Paduano et al., 1994; Shehata, Abdullah, & Areed, 2009):

\[
B = \frac{U}{2w_h \cdot L_c} = \frac{U}{2c} \sqrt{\frac{V_p}{S_c L_c}}. \tag{15}
\]

Large values of \( B \gg 1 \) indicate that the surge is the most unstable phenomenon, while low values of \( B \ll 1 \) favour the rotating stall, as shown in Figure 6.

This work confirms that the pumping appears as a result of greater pressure in the reservoir and downstream in the compressor (positive part of the characteristic). The pumping phenomenon is violent over this large pressure difference (importance of the slope of the characteristic). Furthermore, larger tank volumes are important to favour pump turn over and a pumping frequency for low separation. The surge results in oscillations of the system due to shocks and strong variations in the amplitude of the internal pressure. This violent and dangerous phenomenon must be avoided to maintain the compressor outside the zone of instability. To prevent the surge, compressors feature supervision systems for maintaining the latter in a stable operation field, irrespective of the compression ratio, and a flow rate corresponding to the upper suction pumping speed by referring to the sample gas suction of the discharge.

In this context, the objective of this study was the analysis of experimental data for surge control in order to improve the understanding and prediction of this phenomenon via the integration of new diagnostic techniques that can help to protect the compressor in real time.

4. Application results

In this part of the work, we consider a compression system shown in Figure 7 that consists of a centrifugal compressor, valve, compressor duct, plenum volume and throttle. The throttle can be regarded as a simplified model of a turbine.

The model developed by Gravdahl (Bøhagen & Gravdahl, 2008; Gravdahl et al., 2002) is based on the concepts of thermodynamics and fluid dynamics and is defined as follows:

\[
\dot{P}_p = \frac{a_{10}^2}{V_p} (m - m_t),
\]

\[
\dot{m} = \frac{A_1}{L_c} (P_2 - P_p),
\]

\[
\dot{N} = \frac{1}{2J} (\tau_t - \tau_c),
\tag{16}
\]

where \( p_p \) is the plenum pressure, \( V_p \) is the plenum volume, \( A_1 \) is the area of the impeller eye (used as reference area), \( N \) is the spool moment of inertia, \( m \) is the compressor inlet mass flow, \( L_c \) is the length of compressor and duct, \( m_t \) is the mass flow in the output of the anti-surge valve, \( a_{10} \) is the speed of sound in the inlet of the compressor, \( J \) is the

![Figure 7. Compression system.](image-url)
moment of inertia of the compressor, \(\tau_c\) is the compressor torque and \(\tau_t\) is the turbine torque.

Model (16) is similar to the model developed by Fink et al. given by Gravdahl et al. (2002), while the first two equations are equivalent to the model developed by Greitzer (Greitzer, 1976; Moore, 1984; Moore & Greitzer, 1986; Paduano et al., 1994), where the speed in revolutions per second was not included. The equation for \(P_v\) is derived from the mass balance in the capacity by assuming isentropic process (adiabatic). The third equation is derived from the second law of dynamics:

\[
\sum M = J\ddot{W} \Rightarrow \dot{W} = \frac{1}{J}(\tau_t - \tau_c),
\]

where \(M\) is the moment inertia applied to the system and \(W\) is the angular acceleration.

The compression process is assumed to be isentropic; thus, the following holds true:

\[
\frac{dP_v}{P_v} = -k\frac{dV_v}{V_v} = \frac{k}{\rho_v}d\rho_v, \quad \rho_v = (1/V_v),
\]

which implies the following: 
\[
\frac{dP_v}{P_v} = -(dV/V),
\]

\[
\rho_v = \frac{1}{V_v}(m - m_i) \Rightarrow \dot{P}_v = kRT_v\rho_v = kRT_v\dot{m}_v. \quad (17)
\]

The mass balance in the cavity is given by the following equation:

\[
\rho_v = \frac{1}{V_v}(m - m_i) \Rightarrow \dot{P}_v = kRT_v\rho_v = (m - m_i), \quad (19)
\]

where \(a_s = \sqrt{kRT_v}\) is the speed of sound in the cavity.

Greitzer provided an approximation (Greitzer, 1976; Paduano et al., 1994). If the temperature ratio of the compression is assumed to be close to unity, the amount \((P_v/\rho_v) = R T_v\) is known to also approximate \(P_{01}/\rho_{01}\). Thus, we can conclude the following:

\[
\dot{P}_v = k\frac{P_{01}}{\rho_{01}}V_p(m - m_i). \quad (20)
\]

The isentropic efficiency (adiabatic) is given by the following equation for the pressure calculation:

\[
\eta_i(m,N) = \frac{h_2 - h_0}{h_0 - h_01}, \quad (21)
\]

where \(h_01\) is the input heat stagnation, \(h_02\) is the output stagnation enthalpy and \(h_2\) is the static enthalpy, because \(h = T C_p, h_2 = T_2 C_p\) and \(h_01 = T_01 C_p\).

Using Equation (21), we find \(\eta_i(m,N) = (T_2/T_01)(k-1/k)\) for the ideal gas and determine the following formula:

\[
\eta_i(m,N) = \frac{(P_2/P_01)\left((k-1/k)-1\right)}{\left(T_{01}/T_{02} - 1\right)} = \frac{T_{01}C_p\left[(P_2/P_01)^{(k-1)/k} - 1\right]}{C_p(T_{02} - T_{01})}, \quad (22)
\]

where \(\eta_i(m,N) = T_01C_p\left[(P_2/P_01)^{(k-1)/k} - 1\right]/\Delta h_{\text{ideal}}\). Thus,

\[
P_2 = P_{01}\left[1 + \eta_i(m,N)\frac{\Delta h_{\text{ideal}}}{C_p T_{01}}\right]^{(k-1)/k}. \quad (22)
\]

The following holds true during energy transfer:

\[
\tau_c = m(r_2 C_{01} - r_1 C_{02}), \quad \tau_t = w m(r_2 C_{02} - r_1 C_{01}), \quad \Rightarrow \dot{w} = w \tau_c = w m(r_2 C_{02} - r_1 C_{01}), \quad \Rightarrow \dot{w} = m (U_2 C_{02} - U_1 C_{01}) = m\Delta h_{\text{ideal}}. \quad (23)
\]

If \(\alpha_1 = 90^\circ\), the flow equation is defined as follows:

\[
\dot{w} = m(U_2 C_{02} - m\Delta h_{\text{ideal}}). \quad (24)
\]

We can define the coefficient load, \(\sigma\) (slip factor), using the formula given by Stodola (Nakahira, Ozawa, & Mizusawa, 1992) as follows:

\[
\sigma = C_{\theta2}/U_2 = 1 - \frac{\pi}{Z \sin(\beta_{2b})} - \psi_2 \cot g(\beta_{2b}). \quad (25)
\]

4.1. Control system of surge phenomenon

Surge regulating aims to maintain the compressor within the stable operating ranges. Any regulation will be based on the measurement of flow through the machine such that the adopted regulation can satisfy the following basic requirements:

- The adjustment surge line must be as close as possible to the surge limit line parallel shift by an amount determined in relation to the axis of flow.
- The conditions of the study ensure that the surge control line should not approach the surge line more than necessary for the functioning of the anti-surge system when the conditions of the aspirated fluid vary.
- The system must protect the machine at all possible operating conditions, including starting and stopping.
- The system must have a solution, but the flow measurements are very noisy. Thus, the time constant must be taken into account to calculate the drive.

According to the similarity law in the compressor, the surge curve at different speeds is a parabola (in terms of the
polytropic height vs. the aspirated flow rate). For each rate and for each point of the surge curve, the flow is given by the following:

$$Q_1 = K \cdot N.$$  \hspace{1cm} (26)

The polytropic height is given by the following:

$$H_p = K \cdot N^2 = Z_1 \cdot R \cdot T_1 \frac{n}{n-1} \left(\rho^{(n-1)/n} - 1 \right).$$  \hspace{1cm} (27)

Equations (26) and (27) yield the following:

$$Q_p = K \cdot Q_1^2.$$  \hspace{1cm} (28)

For a suction flow measured with a calibrated orifice,

$$Q_1 = \beta \cdot \sqrt{\frac{h_1}{\gamma_1}}.$$  \hspace{1cm} (29)

$h_1$ is the pressure loss of the orifice, with $\gamma_1 = (1/R)(P_1/Z_1)(P_1/Z_1)(10^4/T_1)$ [kg/m$^3$]. Substituting the value of $\gamma_1$ into Equation (29) yields the following:

$$Q_1 = \beta \cdot \sqrt{\frac{h_1 \cdot R \cdot T_1 \cdot Z_1}{P_1}}.$$  \hspace{1cm} (30)

Substituting Equations (28) and (30) into Equation (29) results in the following:

$$Z_1 \cdot R \cdot T_1 \cdot \frac{n}{n-1} \left(\rho^{(n-1)/n} - 1 \right) = K \cdot \beta^2 \cdot \frac{h_1 \cdot R \cdot Z_1 \cdot T_1}{P_1}.$$  \hspace{1cm} (31)

Simplifying this equation results in the following:

$$\left(\frac{n}{n-1}\right) \cdot \left(\rho^{(n-1)/n} - 1 \right) = K \cdot \beta^2 \cdot \frac{h_1}{P_1},$$  \hspace{1cm} (32)

where

$$\frac{1}{(K \cdot \beta^2)} \cdot P_1 \cdot \frac{n}{n-1} \cdot \left(\rho^{(n-1)/n} - 1 \right).$$

According to this formula, the following is sufficient to avoid surge for all suction conditions:

$$\frac{h_1}{P_1 \cdot (n/n - 1)(\rho^{(n-1)/n} - 1)} \frac{1}{K \cdot \beta^2}.$$  \hspace{1cm} (33)

The inequality given by Equation (33) is important for the following reasons:

- It is independent of the speed of the compressor.
- It only requires the measurement of the loss of the orifice, $h_1$, without compensation for the pressure, temperature, gas composition and linearisation.
- It is influenced by the gas composition only via $n$, whose variation only slightly influences the quantity $n/n - 1 \left(\rho^{(n-1)/n} - 1 \right)$.

In conclusion, the inequality given by Equation (33) uses the compression ratio, which is sensitive to the suction and speed conditions, to compensate for the flow. Although greatly simplified, inequality (33) is too complicated to be verified with simple instrumentation.

If the compression ratio is sufficiently small, as is the case of single- and two-stage compressors, the amount can be replaced with $(\rho - 1)$ in Equation (34) without measuring the large error for safety reasons, as follows:

$$(\rho - 1) \frac{n}{n-1} \cdot \left(\rho^{(n-1)/n} - 1 \right).$$  \hspace{1cm} (34)

Thus, $n/n - 1 \cdot \left(\rho^{(n-1)/n} - 1 \right)$ can be replaced with $(\rho - 1)$ without introducing large errors.

Hence, Equation (33) takes the following form:

$$\frac{h_1}{P_1 \cdot (\rho - 1)} \frac{1}{K \cdot \beta^2}.$$  \hspace{1cm} (35)

The error percentage is given by the following expression:

$$\epsilon = 100 \left[ \frac{(\rho - 1)}{(n/n - 1)(\rho^{(n-1)/n} - 1)} - 1 \right].$$  \hspace{1cm} (36)

Expression (36) results in an error percentage of the intake flow given by the following equation.

$$Q_1 \cdot \epsilon = 100 \left(1 + \frac{\epsilon}{100} - 1 \right).$$  \hspace{1cm} (37)

In general, the error of the output is approximately half of the error of substituting $(\rho - 1)$ with $n/n - 1 \cdot \left(\rho^{(n-1)/n} - 1 \right)$, that is, $\epsilon \cdot Q_1 \approx (\epsilon/2)$. Inequality (33)

![Figure 8. Anti-surge control operation of the loop](image)
defines the following:

\[ \frac{h_1}{P_1(\rho - 1)} = K', \]  

with \( K' = h_1/P_1 \cdot \beta^2 \). Thus, the expression \( K = h_1/P_2 - P_1 \) can be redefined in the following form:

\[ h_1 \cdot K = \Delta P. \]  

Equation (39) represents a solution for anti-surge control that is a parabolic equation in the plane which has its origin at the point \((\rho, Q_1)\) of coordinates \((1,0)\), which is represented in Figure 8.

This figure highlights the extreme simplicity of the anti-surge system, which collects all of the necessary information from the process to protect the compressor with only two instruments on premise (two differential transmitters).

The preceding equations do not consider the variation of the temperature at the inlet. However, this parameter affects the control process. We can define the following equation:

\[ Q = K' \cdot \sqrt{\frac{h \cdot T_1}{P_1}} \Rightarrow h = K' \cdot \left( \frac{Q^2 \cdot P_1}{T_1} \right). \]  

For specific values of \( Q \) and \( p_1 \), Figure 6 shows how \( h \) varies inversely with the temperature, and two members, \( \Delta P = K \cdot h \), exchange in a proportional manner. In most cases, moving the control line to the right of the surge curve often solves this problem. If the temperature change is important, compensation via the control system given in Figure 9 is necessary. In this case, the following holds:

\[ K = \frac{h}{\Delta P \cdot T_1}. \]  

In practice, a series of validation tests were carried out in parallel on the surge model developed in this work when each potential state of the compression system was uniformly distributed over time. This approach generated the impact validation of this model at each operating point for each system status. This study was limited to the sensors corresponding to the outputs of the previously developed surge model. Four types of sensors are mainly used in this system: pressure, flow, speed and temperature sensors. Some sensors directly provide feedback signals to be used in control loops, such as the discharge rate sensor and the suction pressure sensor. Others are used to develop models of the surge, such as the discharge temperature sensor, the suction flow rate sensor and discharge pressure sensor.

Figures 10 and 11 present the experimental results obtained by using the proposed surge model developed in this work for the different parameters involved in our real-time compression system. To highlight and analyse
the pumping phenomenon, we conducted specific tests using actual data of our compression system. These tests intended to create sensor faults in real time by changing the settings of their conversion to electrical signals. The obtained experimental results are satisfactory and furthermore justify clearly the applicability of the proposed surge model in industrial environments, especially for the control and the monitoring of complex industrial processes’ problems. Therefore, before performing control, several tests have to be performed to generate the residuals correctly. Indeed, these tests are based on the developed surge phenomenon model of exploited variables. The advantage of the developed model in the present lies in its flexibility which allows the study and the analysis of the aerodynamic instabilities. Hence, it is possible to analyse separately surge phenomenon and rotating stall, or determine the behaviour of the system separately.

5. Conclusion
The petroleum industry uses facilities of gas compression stations, with the advent of some machines such as centrifugal compressors that are capable of processing huge amounts of gas. On the other hand, they not only allow recovering these gases, but also increase the carrying capacity of the gas. Indeed, the use of the reliable control systems for centrifugal compressors allows increasing the operating coefficients of gas compression stations. However, these machines are subjected to very serious phenomenon, such as the surge phenomenon, which is a state of unstable operation in centrifugal compressors. In the present work, a gas compression system is examined, in order to study the unsteady phenomena (i.e. the surge phenomenon). The main purpose is the study of the surge model used in the development of control systems in centrifugal compressors and to prove that the characteristics of a robust control of the presented system are mainly related to the characteristics of the surge model.

In this work, the surge behaviour model was developed by taking into account the fluid mechanics and thermodynamics. The dimensions and parameters include the aerodynamic performance of the compressor, the characteristics of the upstream and downstream network and the nature of the gas. To address all issues related to the surge phenomenon, the systems supervision and the control are integrated in order to help protect the compressor in real time. From the point of view of the industrial processes’ complexity which is due to the intrinsic physical structure, a robust surge model is needed to be implemented. This robustness requirement may be satisfied if a direct modelling strategy based mainly on the parameters of the process is adapted adequately to replicate the main physical characteristics of the process in real time.

The obtained results show clearly that the proposed model can predict the dynamic characteristics of the examined gas compression system. It is obvious that the solutions presented by the proposed model can reduce the waste of gases, due to the permanent opening valves that recycle gas in gas compression installations. This will recover the lost production to a certain extent while ensuring the operation of the machine and protect the compressor from all dangers.

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