Identification and Testing of a Highly Dynamic Linear Actuation System for Active Compressor Stabilization

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Abstract
In the recent years, a novel tip injection device for active aerodynamic compressor stabilization has been developed. The so-called Ejector Injection System (EIS) utilizes the ejector effect to increase the mass flow, injected onto the Larzac 04 LPC blade tip region. As the system will be applied in combination with a stall detection algorithm, low response times are vital in order to activate the injection in time, if a compressor instability is imminent. In order to activate the injection mass flow on the EIS, a ring has to be traversed in axial direction by three actuators around the annulus. Due to the short response time between the detection of a stall precursor and the fully developed rotating stall or surge event, an opening time of the nozzle of 30 ms is targeted, which requires accelerations of more than 200 m/s\(^2\). In addition to the requirement of high performance, the demand for synchronicity accrues, as a tilting and jamming of the traversable ring has to be avoided. First, an appropriate actuation system has to be identified, wherefore a market research on different actuation concepts was performed. The selected linear motors were tested on a motor testbed, first stand-alone and then in different coupled modes. Finally, the actuation system was implemented in the EIS and tested under the real installation conditions.

Keywords
Active Compressor Stabilization — Linear Motor — Ejector Tip Injection

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INTRODUCTION

Nowadays, the demand for higher fuel efficiency and lower emissions is one of the main drivers of research in the aerospace industry. Especially, new jet engines are facing ambitious requirements regarding efficiency aspects, whereas they also need to provide maximum operational safety throughout their lifetime. In some cases, these demands are in conflict and a reduced efficiency is accepted for the benefit of safety. Considering the compression system of the engine, a higher aerodynamic blade loading is often advantageous for an increased pressure ratio and efficiency, however, it usually increases the risk of aerodynamic compressor instability.

These aerodynamic instabilities can be provoked by multiple reasons, such as transient operation of the engine as well as inlet distortion due to high angles of attack or cross wind situations. In order to maintain stable operation, tip air injection in the compressor can be an appropriate countermeasure [1].

The Institute of Jet Propulsion at the University of the German Federal Armed Forces is investigating the application of tip injection in full jet engine applications on a Larzac 04 test vehicle. Therefore, the development objective is an autarkic application of tip injection, in which the injected air mass flow is extracted from a compressor stage downstream of the injection position. Subsequently, this leads to a decrease in efficiency, if the injection system is activated. Hence, the research is focused on active compressor stabilization, which utilizes a stall detection algorithm to detect stall precursors in the compression system and triggers the activation of the tip injection only, if a compressor instability event is imminent [2].

One of the major challenges for such an active compressor stabilization system is the response time between the detection of the stall precursor and the instability event, which requires a fast response valve and a short distance between the valve and the compressor blade tip. These considerations were included into the design of the latest generation of injection systems, developed at the Institute of jet propulsion. The novel Ejector Injection System (EIS) is utilizing the ejector effect in order to further increase the surge margin enhancement of the tip injection, without increasing the extracted mass flow [3, 4].

1. STALL DETECTION

The concept of active compressor stabilization is crucially dependent on the detection of the blade stall prior to the actual compressor instability event. This enables the countermeasure, in this case the tip injection, to intervene and suppress the stall without any surge or rotating stall event.

Developing an algorithm like this requires a comprehensive knowledge of stall inception, the compression system, and the engine. Therefore, the Larzac 04 has been subject to these investigations for many years at the Institute of Jet Propulsion. Compared to rig-tests, these investigations do not only consider the compressor itself, but also the interaction between the two compression systems and all other
engine components. In recent years, the main focus of the investigations has been the low pressure compressor of the engine. The two-stage transonic compressor is tip critical and stalls at the first rotor over the entire operating range. Furthermore, the first stage of the LPC is easy accessible, which makes it very suitable for investigations on stall and active compressor stabilization.

Based on the gathered experience, a stall detection algorithm was developed by Bindl et al. [5], which has been refined continuously. The distinctive feature of this algorithm is its ability to detect stall in real-time during the monitoring of the engine. It utilizes Kulite fast response pressure transducers to calculate the so-called stall-trigger, which indicates that stall is directly imminent. The algorithm is based on the detection of stall precursors, which is possible throughout the entire operating range of the engine.

As spike type stall precursors are dominant in the Larzac 04 LPC at all engine operating points, the reaction time between the detection of the stall precursor by the algorithm and the fully developed compressor instability is short. This reaction time is even shorter for higher engine power settings. Figure 1 depicts the processed pressure signal and the respective state of the stall trigger at maximum power of the engine. Respectively, the reaction time of 24 ms is the shortest to occur in the Larzac 04 LPC. This reaction time is also one of the most important aspects for the selection of the actuation system.

The mixed primary and secondary air is then injected onto the blade tips, allowing for an increased injection mass flow rate. The demonstration of the working principle was performed on a slightly modified first-generation injection system. Even though, the demonstration was successful, a much higher potential is expected from the new concept. Subsequently, a novel injection system was developed, designed specifically for the application of the ejector effect, which is called Ejector Injection system (EIS).

2.1 Overview

The design focus of the EIS has been on the application of the ejector effect. As it is desired to achieve a maximum injection mass flow rate with a minimum amount of primary air (HPC air), the design goal for the ejector is maximum entrainment. This required as drastic change in design compared to previous injection systems, which featured multiple injection ducts, distributed around the circumference. In contrast to that, the EIS is designed axisymmetric with one single air injection duct around the entire annulus.

Figure 2 shows the composition of the EIS schematically. The EIS is mounted directly upstream of the Larzac 04 LPC, as the inlet adaptor can be removed from the engine casing. In order to provide the suitable interface for the engine’s bellmouth inlet, it is installed upstream of the EIS. Downstream of the EIS, a flexible sealing provides vibration decoupling from the engine, which is important for two reasons. First, it protects the EIS from potential excitation by the engine. Second, the addition of a considerable amount of weight at the front of a jet engine’s casing could influence the vibration characteristics and lead to undesired behavior of the engine itself.

Inside the EIS, the high pressure primary air is injected through the ejector nozzle into the mixing duct. Here, energy and momentum is transferred through the shear layer to the secondary mass flow that is entrained from the ambiance. After the mixing process, the injection mass flow is released onto the blade tip region further downstream.

As the ejector is axisymmetric, the flow-path cross section of both primary and secondary duct is a ring. The air however is provided by means of tubes and hoses, which all have a circular cross section. The supply system upstream of the ejector is providing the transition from twelve circular hose connectors to a single ring-shaped duct, both for the primary and secondary air supply.

Further details on the design of the ejector and the inner aerodynamics of the EIS are presented by Brehm et al. [4] and Kern et al. [3].

2.2 Injection Nozzle

Besides the application of the ejector effect, maximum flexibility and minimum response time have been major design goals of the EIS. In order to enable large parametric studies, a variable area injection nozzle is required. As the entire design is axisymmetric, this functionality is realized by a traversable ring at the inner casing of the EIS, as shown in figure 3.
The traversable ring cannot only alter the nozzle exit area, but is also able to close the nozzle completely. This design feature has two main benefits. First, the closed nozzle generates minimum flow disturbance, if the EIS is inactive. Second, the nozzle can be utilized as a valve with minimum distance to the fan blades. This is of particular importance, as the response time of the EIS has to be very short. In the first-generation injection systems, the limited propagation speed of the flow from the valve through the injection duct had a negative influence on the response time of the system.

The mounting of the traversable ring is located at three positions around the annulus. One of the so-called actuation mechanisms is depicted in figure 4. The mechanism has been designed to be both compact and stiff, in order to create minimum flow disturbance and to avoid vibrations of the traversable ring. The actuation mechanisms are driven by the actuators, which are the subject of this publication.

2.3 Overall System Aspects
A digital mock-up of the Ejector Injection System, comprised of all modules is shown in figure 5. As the design and concept of the EIS is very different from the previous injection systems utilized at the Institute of Jet Propulsion, many system aspects have to be considered, before the first jet engine test campaign.

Initially, the mechanical integrity has to be ensured. Therefore, a structural analysis was performed for the critical parts. Especially, the traversable ring is subject to high stresses due to pressure loads, acceleration forces, and the thin, lightweight design. This lightweight design results in a relatively low stiffness in circumferential direction, which also makes the traversable ring prone to vibrations. Therefore, a special aluminum alloy was chosen for this part.

Furthermore, aspects of aerodynamics have to be considered for two reasons. First, the performance of the ejector has to be verified. In a stand-alone test setup, it was shown experimentally that the EIS ejector increases the injected air mass flow rate by a factor of around \(2.5\). Accompanying CFD simulations confirmed the achieved design goals concerning the inner aerodynamics. Second, the interaction between jet engine and EIS needs to be considered. Even though, the EIS and the engine are decoupled mechanically, the injected air can cause undesired oscillations of the compressor blades. In order to avoid the excitation of the rotor’s resonance frequencies, a vibration analysis has been performed. This way, critical rotational speeds can be identified and avoided during testing.

More information on the design of the EIS, its testing, and the accompanying CFD investigations were published by Brehm et al. and Kern et al. [3].
3. ACTUATION SYSTEM

The injection nozzle of the EIS features a traversable ring, which is mounted on three actuation mechanisms. These mechanisms are driven by the actuation system, comprised of three actuators and their respective accessories.

3.1 Requirements

With its injection nozzle design, the EIS offers the unique possibility of a valve directly upstream of the compressor blades. This is the first major prerequisite to achieve minimum response time of the tip injection, in order to meet the requirements set by the stall detection algorithm. The second precondition for the utilization of the EIS for active compressor stabilization is the valve speed itself. This requirement sets a high benchmark for the performance of the actuation system, as the minimum response time is 24 ms. Therefore, the nozzle is required to open to nominal position in a similar time, which results in a maximum acceleration and deceleration of around 200 m/s². Besides the actuation speed itself, also the response time of the actuators control or accessory systems is of major importance.

As a tilting of the traversable ring could result in a mechanical damage, the synchronous operation of the actuators is a requirement of major importance. This requires a online monitoring and control of the actuators, as the friction coefficient in the plain bearing is not identical in all three actuation mechanisms. Closely connected to the issue of synchronicity, is the requirement for a high repeat accuracy, as this can also affect the angular position of the traversable ring.

Another aspect to consider is the process of acceleration and deceleration. In order to ensure the synchronicity and mechanical durability of all parts, a defined acceleration and deceleration is required. For example, a deceleration by a buffer is not an option. Furthermore, the actuation system is supposed to open the nozzle to defined positions, for example at different operating points of the engine.

Finally, also the aspect of mechanical integration has to be considered in the selection process. Each actuator generates a downstream wake, which interacts with the compressor blades. Thus, the flow distortion at the outlet of the EIS is strongly influenced by the width of the actuators.

In summary, the following requirements were defined:

- Actuation performance (200 m/s²)
- Control response time
- Synchronous operation
- Repeat accuracy
- Defined acceleration and deceleration
- Targeted positioning
- Mechanical integration

3.2 Selection

Beside the defined requirements, the selection process of the actuation system was driven by the aspects of availability and the cost-value ratio. Therefore, the presented results are focused on solutions, which are available in small numbers from companies with a present sales and service structure, whereas the theoretical capabilities of the respective technologies might be higher.

Step 1 The selection was carried out according to the top-down method. In this respect, the first step was the selection of the force generation, with the option of pneumatic, hydraulic, and electrical actuators.

Pneumatic Pneumatic actuators are lightweight and compact, however, the requested acceleration can only be achieved by an additional preloading using a spring. The deceleration would in this case be realized by a buffer. As this violates several requirements, a pneumatic actuation system is not considered.

Hydraulic Hydraulics can generate enormous forces with compact actuators and are able to reach the required dynamics. However, the synchronicity at those accelerations cannot be ensured. Furthermore, the control response time is very limited. Therefore, also hydraulic actuators are not considered any further.

Electric Electric actuators offer an extraordinary performance with the possibility of minimum control response times. However, considering high speed actuators, the available force is usually relatively low compared to equally sized hydraulics. Nevertheless, the electric actuation system is the only option to fulfill the requirements.

Step 2 In the next step, the working principal of the electric actuation system is evaluated. Amongst the variety of electrical actuators, only those with a linear actuation direction were considered.

Spindle drive Spindle drives offer high forces, but usually cannot reach the required dynamics and are therefore not considered.
Moving coil actuator Moving coil actuators can reach high accelerations, however, the motor force is too low for the intended application. Furthermore, synchronization seems to be a major challenge.

Linear motor The linear motor can reach the required performance and can provide the appropriate force to accelerate the weight of the traversable ring. Moreover, its repeat accuracy is usually very high.

Step 3 As there are numerous variants of linear motors, the respective motor type has to be selected. Many linear motor types are built for the application in large industrial machines with long traverse paths. These motors are typically not suitable for the application on the EIS, as they are usually large and require an external mounting and position measurement system. The servo-tube linear motor, however, is very well suited for the presented application. Many motors of this type offer high performance at compact size and usually feature an integrated position measurement system, which makes them ideal for the application in the EIS.

Step 4 The last step of the selection process is the definition of the manufacturer, which is closely connected to the dimensioning. For all manufacturers under consideration, the most suitable motor was selected, then the respective performance data was compared. The motors are operated by a drive, which carries the electronics and the communication system. The available drives were also evaluated, considering the response time and the communication capabilities. After taking all these aspects into account, the NTI LinMot® PS01-23 high-performance linear motor, shown in figure 6, with an E1250 drive was selected.

3.3 Dimensioning Usually, manufacturers offer a broad variety of different motors for different applications. A servo-tube linear motor consists of the primary part with the coils, and the secondary part, called slider, which carries the permanent magnets. Several variants of primary and secondary part are available, and have to be combined for the desired application. The process of finding the most suitable combination is called dimensioning and was performed by means of the motor dimensioning software, provided by the manufacturer.

In order to ensure comparability, a simplified but representative movement was utilized, which is displayed in the first graph of figure 7. The task was a movement of 40 mm symmetric to the mid-position of the slider in 30 ms, following a sine curve. This test-case was chosen for the following reasons:

- 40 mm: Approximately the distance for the nominal nozzle opening.
- 30 ms: Equals 90 % nominal nozzle opening after 24 ms (for sine curve).
- Sine curve: Soft acceleration and deceleration
- Symmetry to mid-position: Comparability of different slider lengths.

The second and third graph of figure 7 show the resulting speed and acceleration of the test case. The fourth diagram displays the resulting actuator force and the upper and lower force limit for the most favorable slider-stator combination. Additionally, the operation of the actuator in extended stroke position is visible. In this case, the slider’s permanent magnets are no longer covered by the coils of the stator. Therefore, the maximum force is decreased in the outside positions. However, as the permanent magnets are comparably heavy, it can be beneficial to accept a decrease maximum force in favor of the reduced actuator force, due to the reduced accelerated weight.

During the movement, the velocity-dependent self induction effect reduces the maximum positive force and increases the maximum negative force. This effect has to be accounted for, if the motor is operated at its maximum force.

![NTI LinMot® servo-tube linear motor](image)

**Figure 6.** NTI LinMot® servo-tube linear motor

![Motor dimensioning calculation](image)

**Figure 7.** Motor dimensioning calculation
4. EXPERIMENTS

Servo-tube linear motors were identified as an appropriate actuation system for the injection nozzle of the EIS. In order to gain experience in the operation of the motors, a comprehensive pretest campaign and initiation procedure was performed, which consists of three major steps. First, the motors were investigated on a small motor test rig in stand-alone operation. Second, all three motors were tested on the motor test rig. Third, the motors were tested on the EIS.

4.1 Stand-alone motor tests

For a first evaluation of the performance and the behavior of the motors, the stand-alone test rig, depicted in figure 8, was set up. As the slider bearings of the linear motor should not be loaded, the test rig is designed to apply only an axial load on the motor, just as the actuation mechanism of the EIS. The load setup is also intended to reproduce the real operating condition as good as possible. Therefore, the applied load is composed of an acceleration load, caused by the moved mass, and a friction load from the bearings. The moving mass was designed to match one third of the accumulated mass of the traversable ring and the three actuation mechanisms on the EIS. The friction load is applied via three bearings. The normal force, on the middle bearing can be adjusted by a strain mechanism, which consists of a high-precision spring that is pressed on the bearing carrier by a screw. By measuring the deformation of the spring, the normal force $F$ on the bearing can be determined.

![Figure 8. Motor test rig](image)

In the first tests, the maximum stroke that is possible on the EIS was tested, which is about 50 mm. For this distance, a target time of 35 ms was set and different load settings were tested. Figure 9 shows the position, speed, and acceleration of the motors. Even though several normal force settings were tested, only the curves for the lowest (0 N) and highest (90 N) loads are displayed.

Initially, it can be noted that in both load configurations, the motors reach the targeted time. However, in the high load case, the motion is noticeably delayed during the acceleration period. During deceleration, the delay is compensated by an increased deceleration. The delay is caused by the rather soft P-control setup and the increased static friction. The increased deceleration is enabled by the increased dynamic friction.

Nevertheless, two major points could be proven. First, the motors reach accelerations of more than $200 \text{ m/s}^2$, even with high friction loads. Second, by utilizing the entire stroke on the EIS, it is possible to open the injection nozzle to nominal position (40 mm) in about 24 ms. This could be a backup option if the targeted nozzle opening time of 30 ms should not be sufficient.

![Figure 9. Stand-alone measurements](image)

The stand-alone tests delivered an insight into the general performance of the motors and drew the attention to the control setup. As a displacement of the motors due to friction is undesirable, a stiffer PD-control setup was utilized for the following tests, in order to minimize the delay.

4.2 Coupled motor tests

Subsequent to the stand-alone tests, the coupled motor tests were performed in order to find the best coupling technique and an appropriate control setup for the application of the motors on the EIS.

The coupled motor tests were performed in a test setup similar to the stand-alone tests. As depicted in figure 10, an additional friction load was applied to one of the motors whereas both other motors were running freely. The main evaluation criterion for the synchronicity of the motors is the tilt angle of the traversable ring, derived from the actual position of the motors during the tests. The evaluation benchmark is a maximum tilt angle of $1^\circ$, which equals a displacement of about 6 mm of one motor, while the other two are exactly at the demand position.

These high demands of synchronicity require an appropriate coupling method. There are three different ways to couple the selected linear motors:

- Master-slave mode
- Collective trigger
- EtherCAT® communication
For the master-slave mode, one of the motor drives is configured as the master, which transfers position and other parameters to the slave drives via CAN-bus.

In the collective trigger mode, the motors are all configured the same. The trigger is an electrical signal (rising edge) applied to a digital input of the drive. The motors follow a pre-defined curve, which enables a continuous position feedback to the controller.

Alternatively, the drives also offer the possibility for EtherCAT® real-time communication. Thereby, all motors are configured as EtherCAT® slaves, controlled by an external EtherCAT® master, in this case a NI CompactRIO® real-time controller. However in this publication, only the first two variants are considered, as the setup of the EtherCAT® communication with the available hardware and software is an enormous effort, compared to the other methods.

In a first step, the master-slave mode was investigated. In approach to the real application on the EIS, a movement task was chosen, which is both simple and realistic. In the chosen scenario, the motors were pulling a distance of 40 mm, which is roughly the nominal opening distance of the EIS injection nozzle. The motors are following a sine curve with a length of 30 ms.

Figure 11 depicts the movement of the three motors with 0 N normal force and the resulting tilt angle. The lower graph shows, that the tilt angle exceeds the limit by a factor of 3. The reason for this large deflection is visible in the upper graph. The motors follow the demand curve with only little deviation, however, the demand curves show an offset about 10 ms. This delay is caused by the CAN-bus communication between the drives and is independent from the motor speed. Thus, for the required dynamics, the master-slave mode is not an option.

Subsequently, the coupling of the drives by a collective trigger was investigated. Thereby, a maximum delay of 3.5 µs is expected, as this is the reaction time of the drives. The test method and the software setup of the drive is exactly as for the master-slave mode tests.

Figure 12 shows, that the issue of the delay could be resolved by utilizing the collective trigger. Even with a relatively high load setting of 81 N applied to motor 1 the calculated tilt angle is lower than 0.15°.

Nevertheless, the graphs point out further potential for improvement. The position of the motors during standstill is supposed to be 0 mm or 40 mm within the tolerances of the motor measurement system. However, a slight deviation of less than 1 mm is noticeable, which results in a slight pre-tilt of the traversable ring. Even though the tilt during standstill is relatively small in the presented case, it is desirable that the movement begins with the ring aligned in the best possible way.

The drives offer two different default control setups. One soft P-control setup and one stiff PD-control setup, which are both not stationary accurate. Thus, the control setup is responsible for the slight tilt of the ring during standstill. By adding an integral term, the control setup can be made stationary accurate. With the resulting PID-control, a positioning accuracy of less than 3 µm within the measurement accuracy of the positioning system can be achieved at standstill.

Figure 10. Motor test rig for coupled motor tests

Figure 11. Master-slave mode with 0 N load

Figure 12. Collective trigger with 81 N load on motor 1
4.3 EIS motor tests
In the last step, the linear motors were integrated into the EIS. For this purpose, the collective trigger coupling method was used in combination with the previously determined PID control setup. The EIS motor tests were conducted with increasing velocity in the three following steps:

1. Determination of dry friction force
2. Determination of viscous friction force
3. Increase of velocity to maximum load

In addition to the PID-controller, the motor drives also feature a pre-controller, which offers the possibility to take friction forces, acceleration forces, and other external loads into account. Whereas the moved weight is known, the friction forces are unknown. Determining these can further improve the positioning accuracy during the movement and thus, can reduce the tilt angle of the traversable ring. The described procedure was performed for each motor individually, as each actuation mechanism behaves differently.

In the first step, the dry friction force was determined. Thereby, the ring was slowly moved, and the motor force was monitored. Due to the low velocities, the acceleration forces and viscous friction forces are negligible.

The determination of the viscous forces was performed at moderate speeds. The viscous forces occur due to the design of the actuation mechanism, depicted in figure 13. During the opening of the nozzle, the expansion volume in the actuation mechanism is filled through a ventilation hole.

Finally, the actuation time was decreased step by step, while monitoring the tilt angle. For each nozzle opening time several experiments have been performed, whereby good reproducibility was observed. The obtained results are presented in table 1. The tilt angle increases noticeably for the lowest nozzle opening time and meets the previously defined limit of 1°. This is caused by the maximum current limit of the motors, which prevents an overload of the system. If the maximum current is reached, the controller cannot further correct the position of the slider. The movement an opening time of 30 ms and the resulting tilt angle is depicted in figure 14. The mentioned deviation between actual and demand curve due to the current limiter can be seen in the upper graph. It is shown that motor 3 is following the demand curve very well, whereas motor 1 and 2 have already reached their limit. This is caused by the different friction behavior of the three actuation mechanisms.

Nevertheless, it can be noted that none of the presented experiments caused a tilt high enough to jam the mechanisms of the traversable ring, which fulfills the safety requirement. The experiments at maximum performance showed that the targeted opening time of 30 ms can be reached, though there is no power reserve left.

<table>
<thead>
<tr>
<th>opening time</th>
<th>max. acceleration</th>
<th>max. tilt angle</th>
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<tbody>
<tr>
<td>2000 ms</td>
<td>0.05 m/s²</td>
<td>0.03°</td>
</tr>
<tr>
<td>500 ms</td>
<td>0.79 m/s²</td>
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<td>40 ms</td>
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<td>30 ms</td>
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<td>1.09°</td>
</tr>
</tbody>
</table>

Table 1. Maximum tilt angle

5. CONCLUSIONS
The implementation of the linear actuators is a major milestone for the commissioning of the Ejector Injection System. A system of three coupled linear actuators was chosen for the challenging application, which has high demands concerning dynamics and synchronicity. The following conclusions can be drawn from the presented work:

1. There are various concepts with each numerous variants for linear actuation systems commercially available. The conducted market research was focused on off-the-shelf solutions with an available sales and service infrastructure. On that basis, linear motors were identified as a clear favorite for the desired application.

2. The stand-alone test campaign offered a first insight into the performance of the motors. In this setup, different friction loads were tested while a load similar to the targeted
application had to be accelerated. The tests showed that the targeted accelerations can be achieved.

3. Subsequently, the coupling of the three motors was investigated on the motor test rig, where two different coupling methods were examined. The tests showed that the collective trigger method provides the best results for high dynamics. Furthermore, a PID control setup was chosen.

4. Finally, the motors were installed on the EIS and tested in realistic installation conditions. Even though the friction forces are higher than initially estimated, it could be demonstrated that the targeted nozzle opening time of 30 ms can be achieved.

The linear actuation system will make a decisive contribution to the application of the EIS in active compressor stabilization. It enables the EIS to respond to compressor instabilities throughout the entire operating range of the Larzac 04 LPC.

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NOMENCLATURE

CFD Computational Fluid Dynamics
EIS Ejector Injection System
HPC High Pressure Compressor
LPC Low Pressure Compressor
max. maximum
min. minimum
stat. static

REFERENCES