# Design of a 2 stages compressor for mobility applications, using compact and efficient Moving Iron Controllable Actuators

Patrick Meneroud
dept *Electro-mécanisms*
*Cedrat Technologies*
Meylan, France
patrick.meneroud@cedrat-tec.com

Steven Rowe
*dept. Electro-Mecanisms*
*Cedrat Technologies*
Meylan, France
steven.rowe@cedrat-tec.com

Gérald Aigouy
*dept. Customer relation*
*Cedrat Technologies*
Meylan, France
gérald.aigouy@cedrat-tec.com

Celen Nil
*Ariane Group*
Bremen, Germany
celen.nil@ariane.group

Armin Isselhorst
*Ariane Group*
Bremen, Germany
armin.isselhorst@ariane.group

Moritz Branco
*ESTEC European Space Agency*
Noordwijk, The Netherlands
Moritz.Branco@esa.int

Jan Persson
*ESTEC European Space Agency*
Noordwijk, The Netherlands
Jan.Persson@esa.int

***Abstract*— An actuator is rescaled for integration into a compressor used for the liquefaction of hydrogen vapor boil off, into a propellant storage system. The goal is to evaluate the feasibility of liquid hydrogen zero boil off, for long duration storage at 20 Kelvin cryogenic liquid condition. This article presents the actuator trade off, selection and special features imposed by the application. The actuator design is presented, its characteristics are measured, and resulting performances are presented and discussed. :**

***Keywords***— *compressor, MICA$^{TM}$, magnetic, linear, actuator, compactness, efficiency, reed valve*

## I. INTRODUCTION

This article presents the design, realisation and test of a double stage reed valve compressor, based on the Cedrat Technologies **M**oving **I**ron **C**ontrollable **A**ctuators concept (**MICA$^{TM}$**). The compressor specifications are defined for the evaluation of Hydrogen micro liquefaction for long-duration liquid cryogenic storage with zero boil off. MICA$^{TM}$ actuators have been used in the compressor because of their special design insures a very long life time and compact design. This article presents the context, explains the main design issues, and details the performances obtained at both actuator and compressor levels.

## II. COMPRESSOR FOR SPACE APPLICATIONS

### A. *Frame of the presented work*

The work presented has been made within the frame of a **T**echnology **R**esearch **P**rogram (TRP), which is financed by the **E**uropean **S**pace **A**gency (ESA). In prevision of future manned missions there is a need [1] to manage the liquid hydrogen propellant stock with **R**educe-**B**oil-**O**ff (RBO) or even **Z**ero **B**oil **O**ff (ZBO) conditions at a reasonable mass cost. Vented hydrogen due to external heating is to be re-circulated thanks to a liquefaction system. Hence there is a need of efficient, compact and long life compressors as part of a **J**oule **T**homson (JT) cooling device. This research program aims at proving the feasibility of the cooling function for such missions by building and testing a reduced size breadboard model.

### B. *Space application requirement for compressors*

The goal is to optimise the use of hydrogen propellant despite external heat sources. Therefore, compressors are needed with a very long life time in the range of many years of continuous working, low energy consumption and low mass. Their use for space applications requires no friction issues despite a mobile piston which is necessary to compress the gas. The consequence is the plunger of the actuator should be suspended with no contact with the fixed parts. This can be achieved due to the use of flexible blades on which the plunger is attached. Flexible blades should show high flexibility in the direction of the stroke and high stiffness in all other directions.

An air gap is required for the pistons including a machining tolerance although this leads to some gas leakage. Great attention must be paid to reduce this piston air gap as small as possible for compression efficiency.

### C. *The compressor function*

Compressors are used to compress the cooling gas which enables the generation of cold liquid through the use of a Joule Thomson nozzle. A description of the gas and fluid circuits is presented in Fig. 1.

Fig. 1, shows the full cooling circuit, the MICA actuator which compresses boiled hydrogen up to 50 bars via two stages. The compression operation heats the $H_2$ gas up, typically to 200°C and it is then cooled to ambient temperature by a water jacket and chiller exchanger system. Then the cooling of the compressed gas continues in two counter flow heat exchangers within a Nitrogen exchanger which extracts the heat taken by the transformation of Nitrogen from liquid to gas. Finally the cooled compressed hydrogen at about 40°K is injected through a joule Thomson nozzle. The depression liquefies about 45% of the Hydrogen

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The feasibility study of hydrogen recirculation for long term mission is a TRP program financed by ESA

which returns to the storage tank at about 5 bars. Boiled-off hydrogen is feed to the compressor.

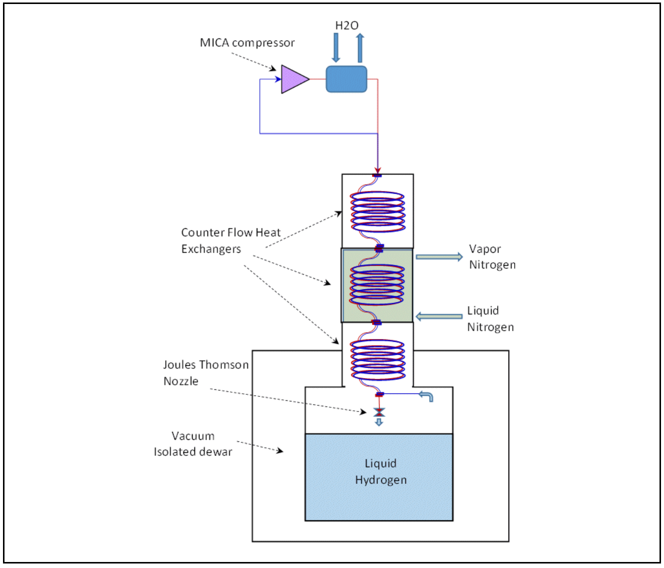


Fig. 1. Breadboard cooling circuit principle

### D. *Space compressor structure*

Gas compressors are to be included in a cryogenic upper stage propulsion system. Therefore attention is paid to make a breadboard compatible with the European Cooperation for Space Standardisation (ECSS) criteria. In particular space application require compactness, no friction, efficiency with low vibrations and low electromagnetic noise. Friction is a source of wear and one impact of this is a change of the behaviour of a system over time. For long duration missions it is desirable that equipment should work the same way during the full period time. Another consequence of wear is the creation and the scattering of fine particles which pollute the space environment and may damage the quality of optic equipment.

The compression function requires the relative movement of at least two parts to reduce the volume of the compression chamber. Possible moves are rotation and translation. A non-limited rotation is incompatible with flexible bearings. It requires either friction rolls, which are forbidden or a no contact suspension which may be performed with magnetic bearings but this option remain complex, heavy and costly. The preferred solution therefore is a linear movement even although this implies the constant need of mass acceleration. The energy impact of acceleration is reduced by taking advantage of the resonance frequency of the system introduced by the stiffness of flexible bearings. In order to balance the use of the actuator, both sides of the plunger movement, both forward and backward are used to compress the gas via a design with two compression chambers, one at each end of the active motor part of the magnetic actuator.

### E. *Description of the compressor*

The compressor is a 2 stage design (Fig. 2) increasing the pressure in a first stage from 5 bar to 18 bar and then from 18 bar to 50 bar. Fig. 3 presents the principle of the compression steps. A compression is performed for any move forward and backward of the actuator. After this, each compression operation gas is cooled in a water jacket with cold water to get Hydrogen at 50 bars and ambient temperature.

Fig. 2. A 2 stages compressor

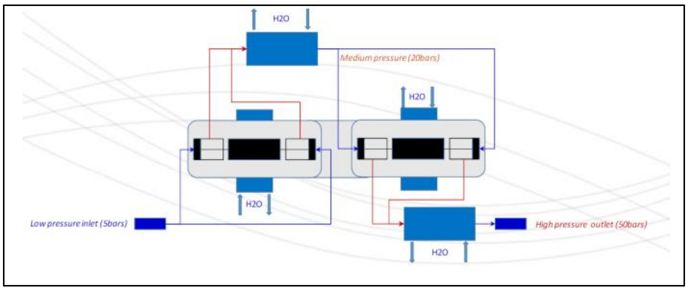


Fig. 3. Principle of 2 stage compressor

## III. DESIGN OF HYDROGENE COMPRESSOR

### A. *Compressor & Actuator trade off*

Existing **C**ustomer **O**ff-**T**he-**S**helf (COTS) compressors for $H_2$ gas are industrial devices with significant larger size and power than required for the feasibility of an embedded space system. Consequently a special compressor has been designed with attention to the mass, materials used and the capability to produce it with space clean constraints. In the same way, the compressor might have used a COTS actuator, like the MICA$^{TM}$ 300CM. However in this case the actuator has to be a $H_2$ tight interface with zero chance of the Hydrogen leaking. Therefore it is preferable to integrate the active part of the MICA$^{TM}$ actuator inside a specially designed compressor housing.

### B. *Special actuator features required for compresor application*

Long life duration is insured by the absence of wear due to friction between the moving parts. All bearings between moving and static parts are managed either by frictionless flexure bearings or by an air bearings type design. The flexible bearing design is made for a theoretical infinite life time. Cedrat Technologies has developed a special skill for the design bearing blades with extreme life time, and a typical stroke capabilities of between ±5 to ±6 mm, low stiffness in the direction of actuation, high stiffness in radial direction and with no buckling. For the application of the compressor $H_2$, 10 flexible bearings presented in Fig. 4 suspend the plunger on each side and insure precise positioning.

Fig. 4. MICA flexible bearing

The fixing of the moving part with spring blades also allows for optimisation of the system stiffness. This bearing design is adapted together with the mass to target a 50Hz resonance, this takes into account the stiffness property of the compressed gas.

The magnetic circuit material has been chosen as a compromise between pure magnetic performances and a reduction of eddy current losses. Several solutions for eddy current reduction have been considered, lamination, cuts in the conductive parts or using high resistive composite materials. Lamination is the classical solution. However this solution goes together with magnetic circuit shape constraints. Typical shapes common with lamination design are circuits which are built with extruded-shape form. Then lamination are positioned in the direction of the extrusion. Attempts to bypass theses shape constraints result in a significant increase of the component cost. The second classical solution is to use a material with high intrinsic resistivity. Materials obtained by compression and sintering of power in moulds are commonly used and allow greater freedom of 3D design. The obtained resistivity is very high in the range of 70 µOhm.m up to 20000µOhm.m. The drawback of these is the fragility of the final piece and also the cost of the mould. For a few prototypes, it remains an expensive solution. The chosen solution for this project is the use of stainless steel whose resistivity is close to 0.8µOhm.m, which furthermore has the advantage of being stainless. However for the range of power of the compressor application, the size of part is too large versus the skin depth, which would result in important eddy current losses. Therefore, part thickness has been reduced by adding cuts in the bulk material. This solution allows the flux density flowing in the whole volume to be controlled and hence reducing sufficiently the eddy current losses.

### C. *MICA$^{TM}$ Actuators characteristics*

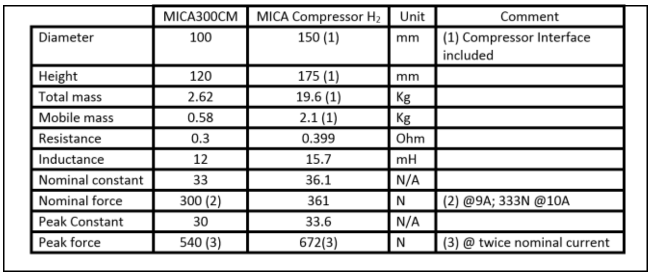
MICA$^{TM}$ actuators are particularly suitable for compact embedded applications. In the first step the theoretical advantage of moving iron actuator concerning compactness versus other type of actuator like **M**oving **M**agnet (MMA) or **M**oving **C**oils (MCA) actuators is explained. However, moving iron are sometime mistaken with electromagnets which are a kind of actuator difficult to control. MICA$^{TM}$ actuator magnetic polarised devices show force to stroke characteristics superior to MCA concerning the continuous force density and are equivalent to MCA regarding the linearity. Thus MICA$^{TM}$ are highly controllable [2] and can be used in closed loop applications.

### D. *MICA$^{TM}$ & Compressor realisation*

The MICA$^{TM}$300CM [3] actuator has been customized to fit with the application and for feasibility study purpose (Fig. 5). The force capability has been improved by 7.7%.

Fig. 5. MICA$^{TM}$ compressor $H_2$ actuator

The table presented in Fig. 6 shows the evolution of the MICA 300 CM after integration into a compressor device.



| | MICA300CM | MICA Compressor $H_2$ | Unit | Comment |
|---|---|---|---|---|
| Diameter | 100 | 150 (1) | mm | (1) Compressor Interface included |
| Height | 120 | 175 (1) | mm | |
| Total mass | 2.62 | 19.6 (1) | Kg | |
| Mobile mass | 0.58 | 2.1 (1) | Kg | |
| Resistance | 0.3 | 0.399 | Ohm | |
| Inductance | 12 | 15.7 | mH | |
| Nominal constant | 33 | 36.1 | N/A | |
| Nominal force | 300 (2) | 361 | N | (2) @9A; 333N @10A |
| Peak Constant | 30 | 33.6 | N/A | |
| Peak force | 540 (3) | 672(3) | N | (3) @ twice nominal current |

Fig. 6. Comparison between MICA 300CM and MICA Compresor $H_2$

Even though both actuators MICA$^{TM}$ 300CM and MICA$^{TM}$ Compressor $H_2$ are comparable with quite similar magnetic components, the table shows clearly that the integration of the MICA$^{TM}$ into a compressor structure modifies significantly the actuator volume and mass due to the volume required for the double compressor system interface and the hydrogen leak tightness. These are very similar actuators, but are not designed for the same application. This will have an impact on the MICA$^{TM}$ compressor $H_2$ performance criteria presented hereafter.

### E. *Projet status*

The project passed in early 2019 the Test Readiness Review (TRR) and the actuator as well the 2 stages compressor is now in a phase of evaluation and testing.

### F. *MICA$^{TM}$ & compresor tests*

The static measurement of MICA$^{TM}$ actuator forces versus the current and mobile part position are presented in Fig.7 & Fig 8. The nominal functioning point is determined by the stabilised 150°C temperature reached in the coil, which corresponds to a 10A DC current. Nominal force is 361N. Curves of Fig.4 show a maximal loss of force of 46% at nominal current for the extreme position of the mobile part.

Fig 8 presents dual representation of force measurement. The linearity of the actuator is excellent until twice the nominal current, with a loss of the force constant of only 7% at 20 A. This current value is defined in the present article as the peak force of 672N and is limited by the available current supplier used for these measurements. The graph shows that actually the actuator is far from being saturated at this 20A current value, and it would be interesting to use the actuator at even higher currents when a higher force is required. The duration of the supply at peak current is limited by the coil temperature that should not exceed the maximum continuous temperature of 150°C.

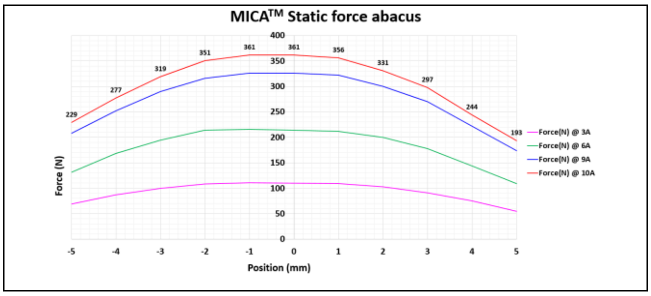


Fig. 7. MICA$^{TM}$ actuator measured forces versus position

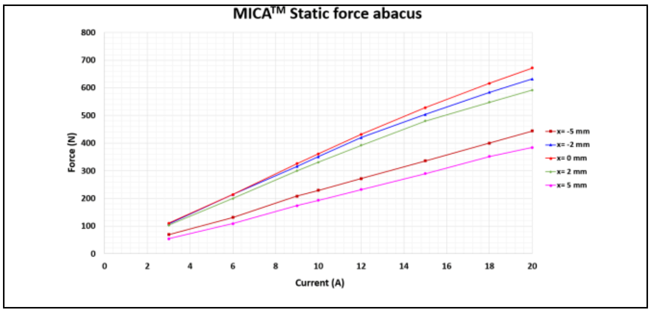


Fig. 8. MICA$^{TM}$ actuator measures forces versus current

Fig. 9 presents the actuator impedance measurement while the plunger is free to move without any gas load to compress. The resonance frequency in this case is at 26.8Hz and takes into account the stiffness of mobile part half the suspension, the mass of the mobile part of the magnetic actuator, together with the two moving pistons of the compressor. Measurement used an impedance analyser whose operating frequencies are between 20 Hz up to 20MHz. Fig. 6 presents the measurement from 20Hz up to 200Hz. At higher frequencies, one other resonance due to the coil capacitance is present, this has no interest here. The resonance Q factor is evaluated from the thickness of the resonance peak with the formulae

$$\boldsymbol{Q} = \frac{\boldsymbol{f_{res}}}{\boldsymbol{\Delta f}} \tag{1}$$

Due to the minimum frequency of the impedance analyser (20Hz) and the resonance frequency (22.7Hz) it is not easy to extract static resistance and static inductance from the measurement results. One should proceed with another method to determine the static resistance and inductance.

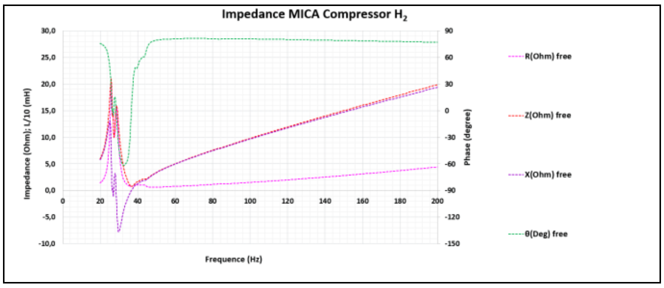


Fig. 9. MICA$^{TM}$ Impedance measurement in plunger free conditions

Another impedance measurement of the MICA compressor $H_2$ has been performed in the blocked condition. Unfortunately as the application purpose is compression of a gas, and the mobile part interface is a double piston fully enclosed. There is no strong mechanical interface which will enable the blocking of the mobile plunger/pistons. Consequently, the fixation of the plunger has been performed with a small screw, which does not show a sufficient stiffness. Therefore the plunger/pistons are not totally blocked and all that happens is the stiffness is increased and hence the resonance, just shifted to a higher frequency as shown in Fig. 10.

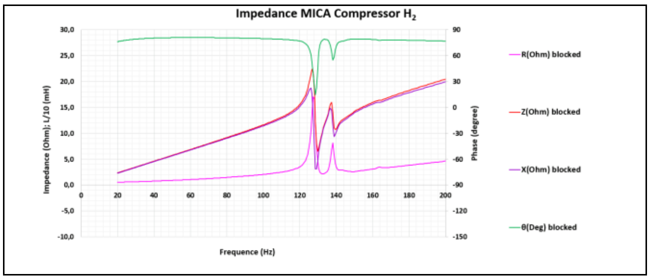


Fig. 10. MICA$^{TM}$ Impedance measurement in plunger weakly blocked conditions

The variation of the resistance in the blocked condition between 20Hz and 100Hz is far enough for the resonance to be exploitable and shows a parabolic variation versus frequency expected for resistance which includes eddy currents dissipation. A fit is performed on this variation and an extrapolation (see Fig. 11) up to static conditions allows to compute the static resistance. It is evaluated to 0.458 Ohm and is decomposed in the actuator resistance measured at winding R=0.399 Ohm and the 4,4m bidirectional connecting cable evaluated at 0.059Ohm.

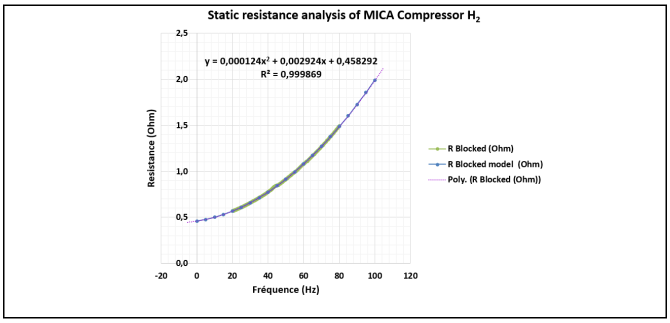


Fig. 11. MICA$^{TM}$ Resistance evaluation

For both measurement cases, free and blocked, the inductance measurement includes the electrical inductance modified by the electromechanical coupling with the plunger displacement. With this in mind the dependence of the measured inductance becomes obvious.

At a lower frequency than the resonance, in mode plunger free, the movement of the mobile plunger due to electromagnetic forces modifies significantly the impedance, hence also the inductance seen from the actuator coil terminals. Above the resonant frequency, the stroke of the plunger is mainly limited by the plunger mass and its inertial forces. The plunger can be then considered as blocked and the measured inductance is $L_0$.

When the plunger is weakly blocked, the fixation stiffness is modified and the plunger will still move under magnetic forces. The measured inductance is closer to the real one but the value is incorrect. At a higher frequency than both resonances, and far enough away from the resonances, both inductances become identical and represent the actual resonance of the coil as plunger move is negligible. Fig. 12

shows inductance and reactance of both cases and the inductance estimated value is 15.7mH.

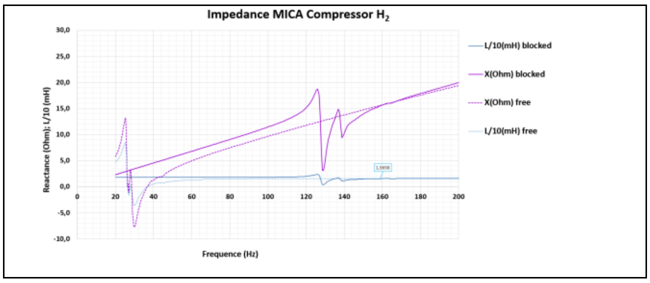


Fig. 12. MICA$^{TM}$ Inductance evaluation

The simplified thermal resistance of the MICA compressor $H_2$ has been measured in nominal conditions. This is defined as the difference between the hot coil point and the MICA external cylinder. The joule power dissipated in experiments is 69.2W. In these conditions the coil temperature is 144°C and the external cylinder temperature is 60°C. Coil temperature is measured by a PT1000 sensor and the external MICA cylinder is measured with a thermal imaging camera. The obtained equivalent thermal resistivity is 1.3 °K/W.

The temperature time constant has also been evaluated by measurement and is presented in Fig. 13. The fitting done with the first part of both heating and cooling curves has been consolidated with other measurement and shows a time constant of about 7 min.

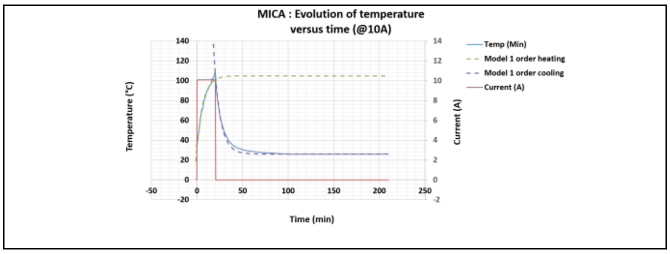


Fig. 13. MICA$^{TM}$ Thermal constant evaluation

The performance criteria for electromagnetic actuators depends on the required application. When the application is the moving of an object, the maximal acceleration at peak force is interesting. The MICA actuator for the compressor $H_2$ maximal acceleration is 32.5G. When the application consists in applying a force within a size/mass constraint system, the criteria peak force versus the actuator mass is interesting and gives a value for the compactness of the actuator system. The MICA compressor $H_2$ peak force versus mass is 34.3 N/kg due to the additional mass of the compressor system. When the application consists in performing a given job for a long time within an embedded device, the criteria Nominal force versus the square root of dissipated power may be the right one. The MICA compressor $H_2$ nominal force versus power is 45.5N/W$^{0.5}$. These three criteria depend only on the actuator itself and not the way it is used. The criteria of maximum efficiency may not be a criteria for qualifying the actuator as it depends also on the application.

In the present application, the recycling of hydrogen through liquefaction during a long term missions the most pertinent criteria is the last one and the actuator has been optimised in this way. The actuator load comprises:

- An integrated mass of 0.5kg for the two pistons which limits the maximum acceleration given previously,
- A quite high stiffness due to the compressed gas at 173N/mm,
- The useful work necessary for compressing the $H_2$ gas which is modelled by a dissipation factor of 200Ns/m.

The MICA compressor $H_2$ behaviour is evaluated with a system simulation tool which shows as expected that under the frequency of 72Hz, the full actuator stroke is achievable (Fig. 14) with a power supply of 200V, and 20A. Over this frequency the actuator is limited first by the supply current due to additional force required to move the actuator at high frequency, then by the supply voltage due to the increasing reactance with increasing frequencies. Considering the efficiency, the actuator shows with the Compressed H2 gas load a maximal efficiency of 92.7% at 53.1Hz (Fig. 15), which is the resonant frequency of the system including the load.

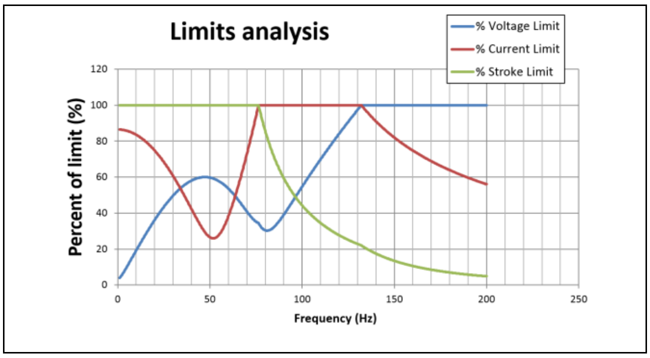


Fig. 14. MICA$^{TM}$ MICA & Supply limits versus excitation frequency

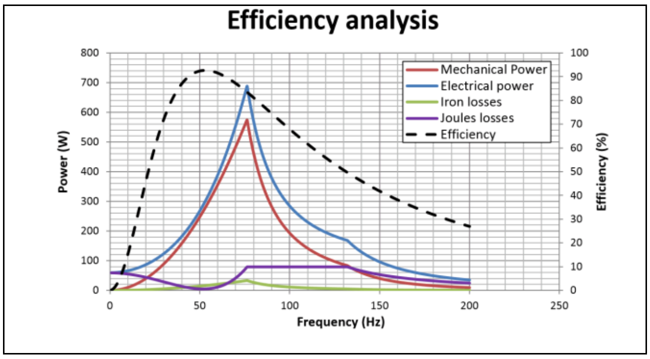


Fig. 15. MICA$^{TM}$ Efficiency curve versus excitation frequency

In this application as a compressor motor for a long term space mission, the critical criteria is the amount of embedded energy, hence the use of the actuator at its resonant frequency which depends both on the compressor and the actuator design is critical.

Finally, in order to be compatible with long term spatial mission, the actuator is designed for infinite life time, insured by correct cooling which reduces fatigue constraints and the absence of wear thanks to the use of both flexible and air bearings design. Flexible bearing fatigue tests are currently being undertaken at Cedrat Technologies, at the moment after two years of testing, close to one billion cycles have been achieved.

## IV. CONCLUSION

The TRP project aims at evaluating the feasibility of hydrogen propellant recycling using a 2 stages compressor system. This includes the design of a custom actuator, its realization and its characterization. The measured performances of the customized MICA$^{TM}$ are presented and show that the actuator is particularly suited for the targeted compressor application. The next activities will consolidate the pure actuator performances into compressor system performances.

## ACKNOWLEDGMENT

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