Linear Resonance Compressor for Stirling-Type Cryocoolers Activated by Piezoelectric Stack-Type Elements (ID: 180)

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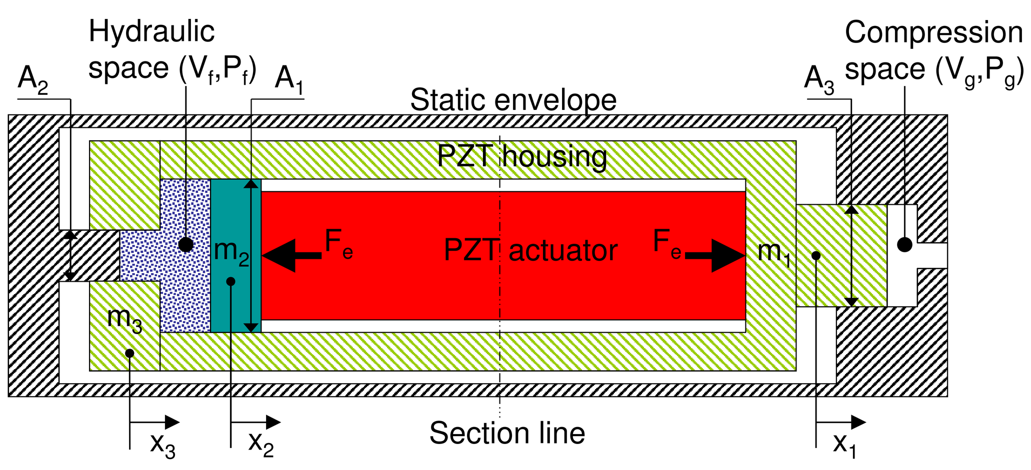
1. INTRODUCTION

Pneumatically activated Stirling-like cryocoolers and all types of the Pulse Tube Refrigerators (PTR) are driven by a pure pressure oscillator, which is sometimes referred to as a valve-less compressor. Concerning a PTR, the pressure oscillator is a critical component, since it is the only mechanically active part subject to wear and failure, and therefore, the lifetime of the cryocooler depends mostly on the compressor reliability. Additionally, occasional contamination of the working gas originates generally from the compressor, which may be a source of wear products, outgassing, lubricant vapour, etc.

In an effort to improve the efficiency and reliability, as well as eliminate the electromagnetic emissions, it was proposed to replace the driving mechanism of the conventional linear compressor by a device activated by a piezoelectric stack-type element (PZT actuator), which is frictionless, has a high volumetric power density and extremely long lifetime. To make the piezo compressor competitive with the conventional one in terms of size and weight, it was critical to force the driving system to operate at mechanical resonance, thus minimizing the electrical power required to elongate exclusively the PZT stack, which is very stiff due to its ceramic nature.

Optimization of the actual compressor concentrated on its ability to drive a miniature Pulse Tube cryocooler, particularly our MTSa model operating at 103 Hz, as well as our future models, which should have similar dimensions, but operate at higher frequencies. The currently available MTSa Pulse Tube requires 11 W average PV power, 40 Bar filling pressure and a pressure ratio of 1.3 to produce a net cooling of 0.4 W at 110 K.

2. THEORY

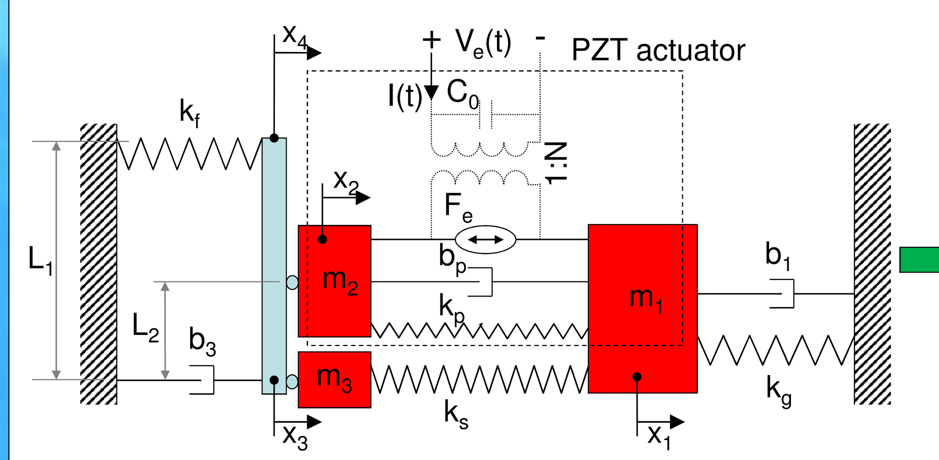


Natural frequency of the idealized one-degree-of-freedom model:

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{\frac{k_P}{a^2}+k_g}{m_1+m_3+\left(1-\frac{1}{a}\right)^2 m_2}}$$

where: a – amplification ratio = A_1/A_2
k_P – PZT stiffness
k_g – cryocooler gas stiffness

Detailed three-degrees-of-freedom spring-mass-damper model:



$$\begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2+m_3 & m_3 \\ 0 & m_3 & m_3 \end{bmatrix}\begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \\ L_2\ddot{\theta} \end{bmatrix} + \begin{bmatrix} b_1+b_P & -b_P & 0 \\ -b_P & b_P+b_3 & b_3 \\ 0 & b_3 & b_3 \end{bmatrix}\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ L_2\dot{\theta} \end{bmatrix} +$$

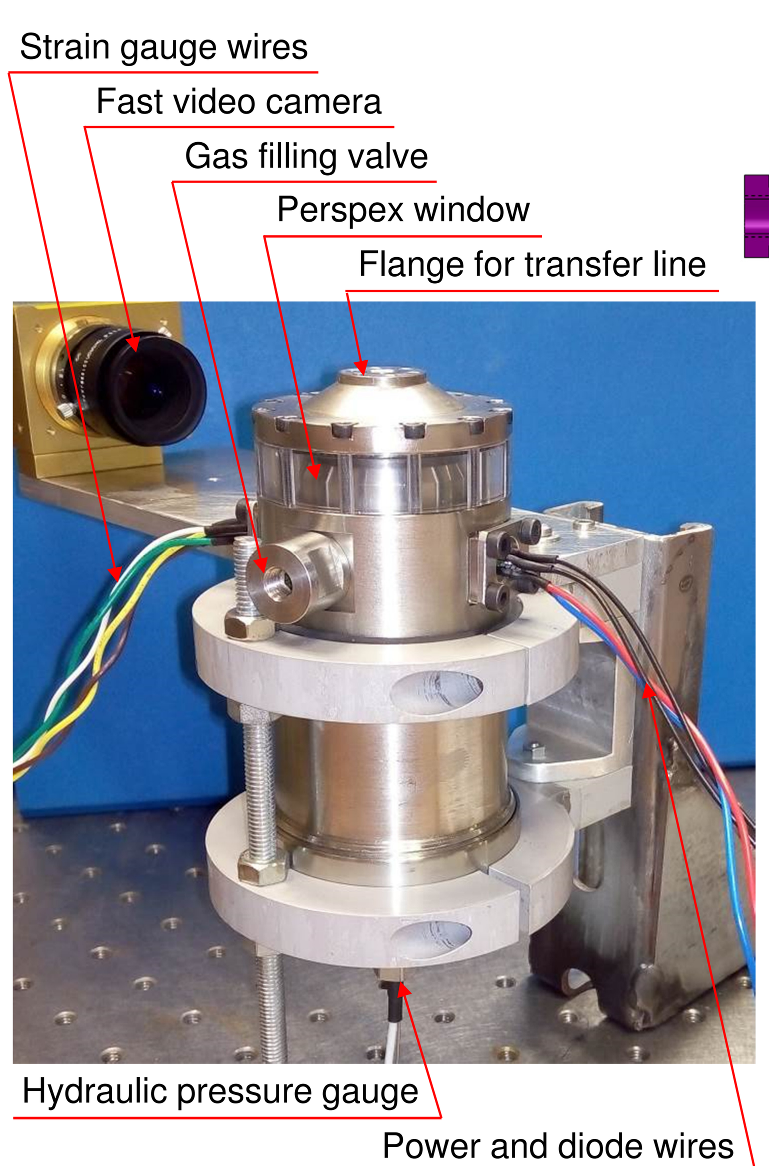
$$+\begin{bmatrix} k_P+k_s+k_g & -(k_P+k_s) & -k_s \\ -(k_P+k_s) & k_P+k_s+k_f & k_s-(a-1)k_f \\ -k_s & k_s-(a-1)k_f & k_s+(a-1)^2 k_f \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \\ L_2\theta \end{bmatrix} = \begin{bmatrix} NV_e \\ -NV_e \\ 0 \end{bmatrix}$$

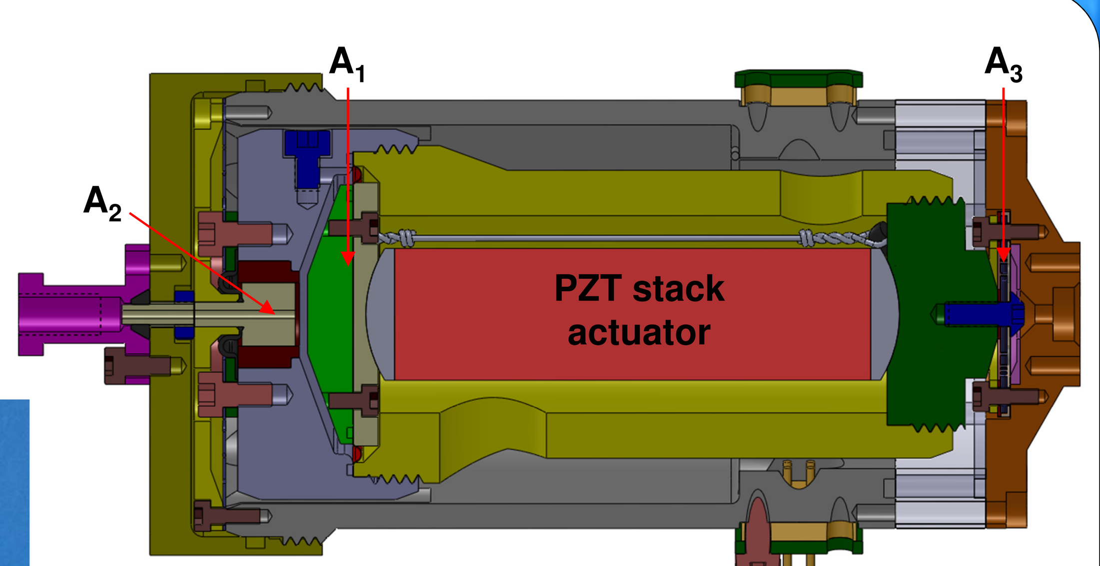
$$I = N\left(\dot{x}_1 - \dot{x}_2\right) + C_0\frac{dV_e}{dt}$$

$$a = \frac{L_1}{L_2} = \frac{A_1}{A_2}$$

$$x_3 = x_2 + L_2\theta \qquad x_4 \equiv \frac{\Delta V_f}{A_2} = x_2 - (a-1)L_2\theta \qquad \Delta P_g = \frac{k_g x_1}{A_1 \sin\phi_{\dot{V}_g P_g}} \qquad \Delta P_f = -\frac{k_f x_4}{A_2}$$

3. DESIGN





PZT: high voltage hard PZT multilayer actuator P-016.40P, 58 x Ø16 mm, of PI company. k_P = 94 N/μm, C_0 = 510 nF, N = 5.6 N/V, F_e up to 5600 N and quasi-static elongation up to 60 μm.

A_1: stroke to diameter ratio about 1:1000, pressure amplitude up to 40 bar. Implemented as a deformed spring steel diaphragm with a rigid interior area.

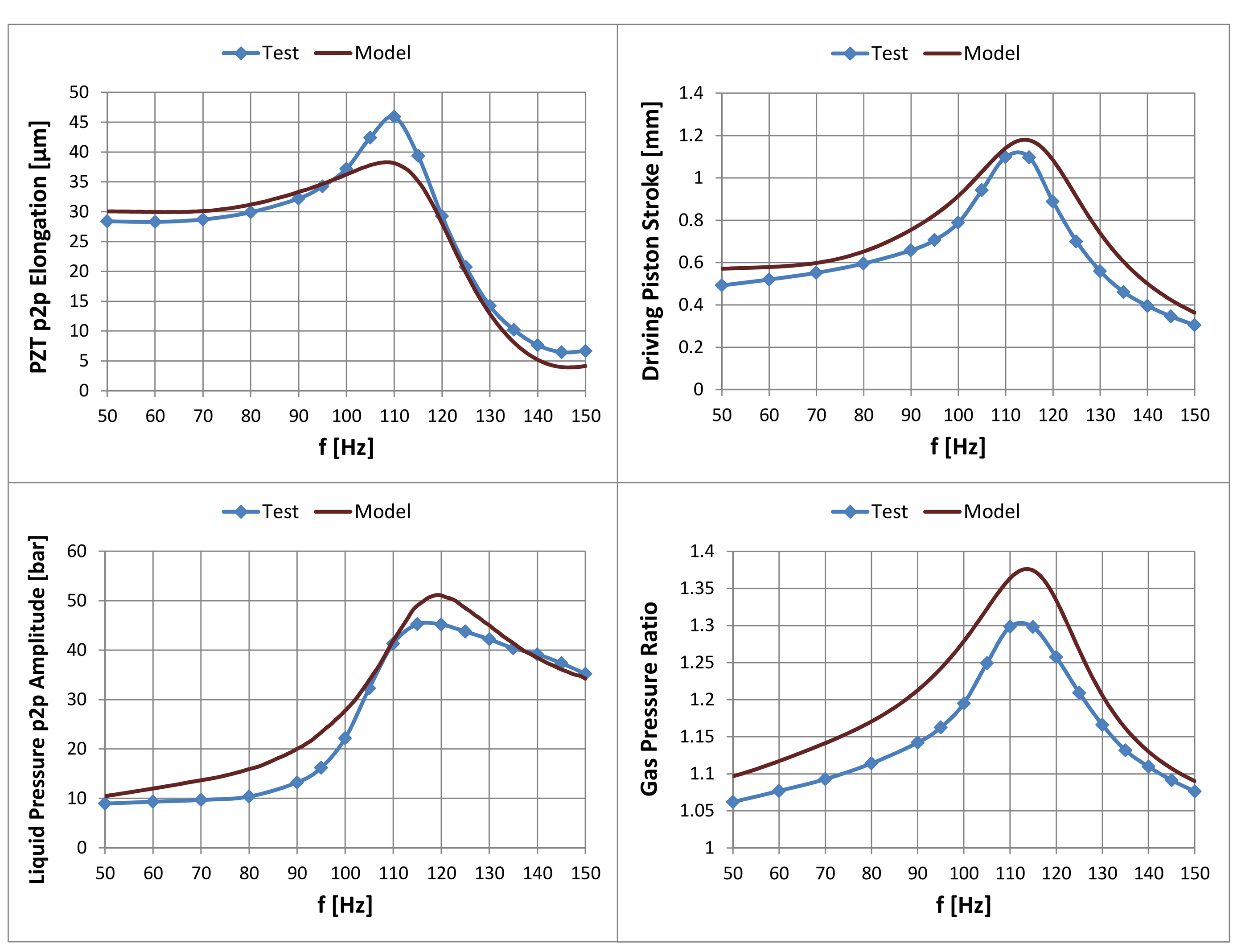
A_2: stroke to diameter ratio about 1:5, pressure amplitude up to 40 bar. Implemented as a gap-sealed piston-cylinder assembly. The fluid separation occurs on the opposite side of the piston, outside the PZT housing, by using a compliant diaphragm.

A_3: stroke to diameter ratio about 1:10, pressure amplitude up to 6 bar. Implemented as a composite structure, consisting of a stack of the spring steel semi-flexural disks with spiral-like cuts filled and coated by soft silicone rubber.

4. RESULTS

$V_e(t)$ = 300(1-cos ωt) V
d_1 = 34 mm
d_2 = 7.7 mm
d_3 = 12.5 mm
a = 19.5
m_1 = 0.2537 kg
m_2 = 0.0888 kg
m_3 = 0.2918 kg
b_P = 1000 Ns/m
b_3 = 10 Ns/m
$k_{P,\,add}$ = 5.4 N/μm
$k_{g,\,add}$ = 38 N/mm
k_f = 4.81 N/μm
k_s = 1.5k_P (estimated)

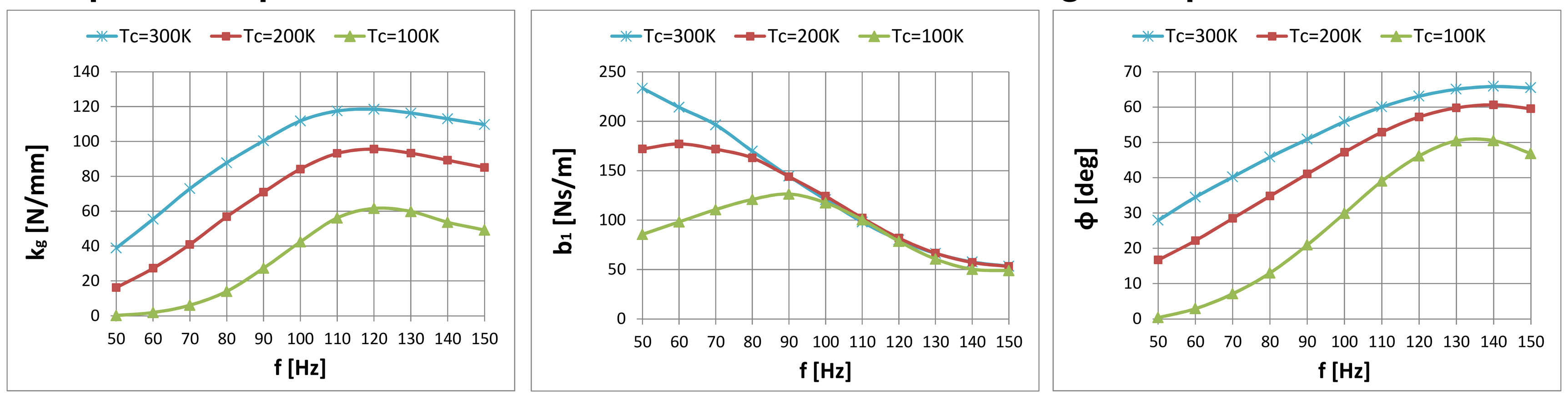
$P_{g,\,fill}$ = 40 bar
$P_{f,\,fill}$ = 40 bar (self levelled)



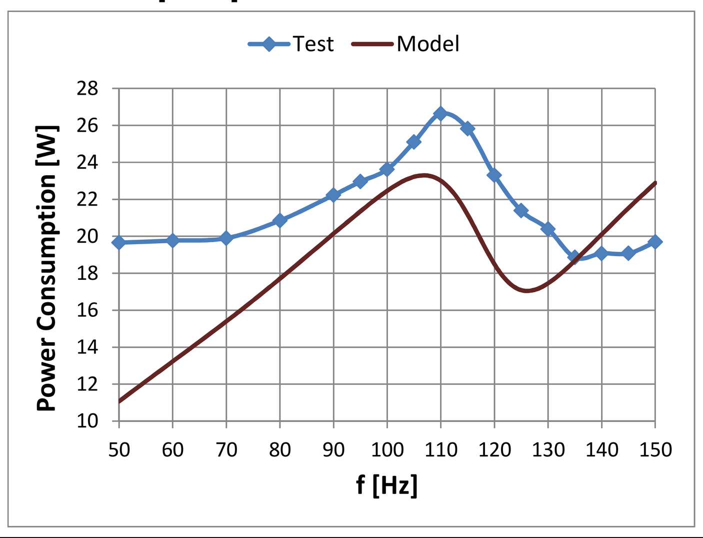
Note: The measurements were taken in the first few seconds from the start of the run; thus, the test system was retained at room temperature, except for the cold heat exchanger of the cryocooler, which was at about 280 K.

5. MISCELLANEOUS

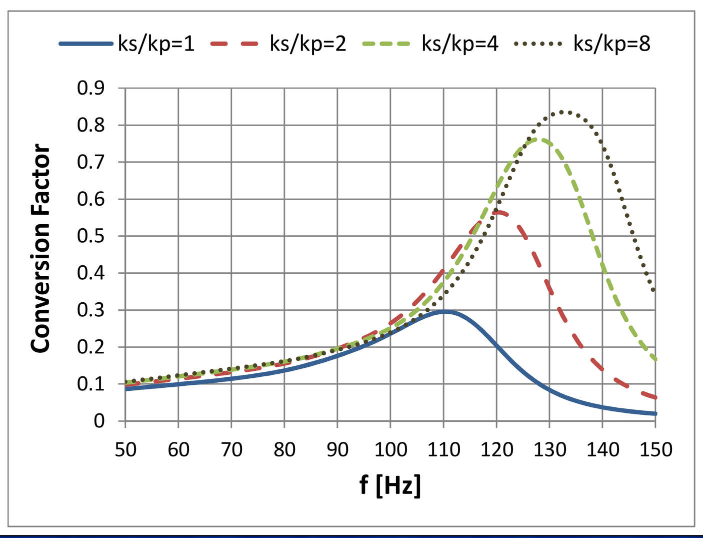
MTSa Pulse Tube characteristics estimated at Ø12 mm piston boundary using SAGE™ with piston amplitude retained at 0.5 mm and hot heat exchangers kept at 300 K:



Measured and calculated mean input powers:



Calculated electromechanical conversion factor vs. structural stiffness:



6. SUMMARY AND CONCLUSIONS

1. A PZT driven compressor for Stirling-type pneumatic cryocoolers operating in mechanical resonance was modelled, optimized, designed, constructed and tested in this research work.
2. An experiment showed that under an actual load applied to the compressor the PZT stack amplitude increased at 110 Hz by 62% relative to the quasi-static operation, and thus proved the concept of the low frequency PZT resonance, which is the main point of the proposed compressor.
3. Despite some mismatching between the impedances, the compressor demonstrated the feasibility to drive our miniature Pulse Tube cryocooler MTSa, operating at 103 Hz. A cryocooler similar to MTSa optimized for operation at a frequency about 115 Hz should fit perfectly to our produced compressor.
4. A satisfactory agreement was obtained between the model simulations and experiments on the actual compressor, with the experimental results somewhat lower relative to the simulations. This can be explained by the secondary losses and non-linear effects that were not considered in the model.
5. Insufficient structural stiffness obtained within the actual assembly relative to the ideal CAD model (1.5k_P relative to 5k_P) decreased drastically the initially expected electromechanical conversion factor. According to the model, the compressor efficiency can be increased, on account of the reactive power, by 55% when the present structural stiffness is doubled, and by 81% when increasing the stiffness four times.

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