Ultralight Pressure Regulator for Application in Pneumatic Prostheses

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Abstract—**Available pressure regulators are considered too heavy for application in pneumatic prostheses. Goal of this paper is developing a lighter pressure regulator adapted for this specific application. We present an ultralight mechanical pressure regulator with several possible configurations as well as an optimization strategy for easy adaptation of the designs to slightly differing requirements. Our system functions using a conventional piston-valve configuration and can be used with either mechanical or gas spring. Dependent upon the configuration the pressure regulator can either use disposable CO_2-cartridges or an integrated gas-storage container for gas storage. The pressure regulator performance of both the spring and pressure controlled configurations are evaluated for airflow stability and sufficient pressure recovery. The results have shown stable outlet pressure for both the pressure and spring controlled configurations.**

Index Terms—**hand upper extremity prosthesis prosthethics gas small prostheses pneumatic regulator CO2 energy material -efficient ultralight light low weight compact.**

I. Introduction

THE exact number of upper-extremity amputatees is unknown since a significant part of the group lives in developing countries and rural areas. It is estimated to be circa 25 million by some experts [1]. In order to improve the life quality of amputees a part of this group uses prostheses as aid for performing activities of daily living (ADLs) and for aesthetic reasons. Prostheses are used for ages by different civilizations including the ancient Egyptians [2]. Past research indicates abandonment of upperlimb prostheses between 35% and 45% of the pediatric population and between 23 % and 26 % of the adult population [16]. A significant share of the prostheses is not used for activities of daily living, vocational activities and leisure activities and they clearly do not fulfill all the user requirements for every individual. This brings up the question what people actually expect from a prosthesis. Research suggests [3] that the desires of the patient can be summarized as Cosmetics, Comfort and Control as elaborated below.

Cosmesis: The prosthesis should look appropriate and give the user a sense of wholeness.
Comfort: The prosthesis should be comfortable to wear in all phases of usage.
Control: The system should be easily controllable and give the patient more abilities than without the prosthesis.

Unfortunately no prosthesis exists which fulfills all these demands at the same time. Most patients consider the comfort of the prosthesis too low due to weight issues, perspiration problems and the hassle of putting on a shoulder harness. Control is often difficult with prostheses as externally powered prostheses tend to have no feedback at all and body powered prostheses have reduced feedback due to friction losses. Finally hook prostheses can be considered the most functional prostheses given their ability to pick up small objects. These prostheses are considered cosmetically unacceptable. For these reasons a large number of prostheses is not used.

In order to reduce the weight problems associated with actuated prostheses pneumatic alternatives have been researched. The combined system of actuators, transmission and energy storage for pneumatic designs has the potential to be lighter than for electrically actuated prostheses. A complete gas actuated arm prostheses has been envisioned with a total weight of 2kg [17] while a complete state of the art electrical arm prosthesis has an estimated weight of 3.6kg [18]. The high resistance pneumatic parts have against contamination gives such prostheses a high reliability. Past complications with oversized components and difficult refill procedures have been overcome through customized parts. Given new developments pneumatic prostheses might prove to be a valuable option in future applications. This paper focusses on the improvement of the pressure regulator of a pneumatic hand prosthesis by minimizing the associated weight. This paper is aimed at the WILMER ® family pneumatic prostheses, like the bi-phasic child version (Fig. 1). These prostheses use custom pneumatic actuators and disposable CO_2-cartridges as energy source.

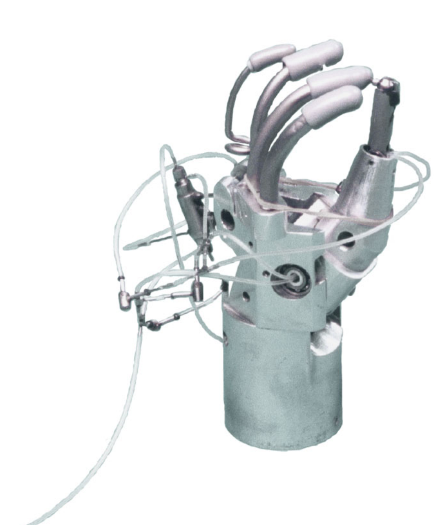

Fig. 1. WILMER ® bi-phasic pneumatically powered hand prosthesis for children [3]

From a pneumatically point of view the system consists of a CO_2 storage container, a regulator, a control switch and an actuator (Fig. 2) The current pressure regulator (Fig. 3) has a

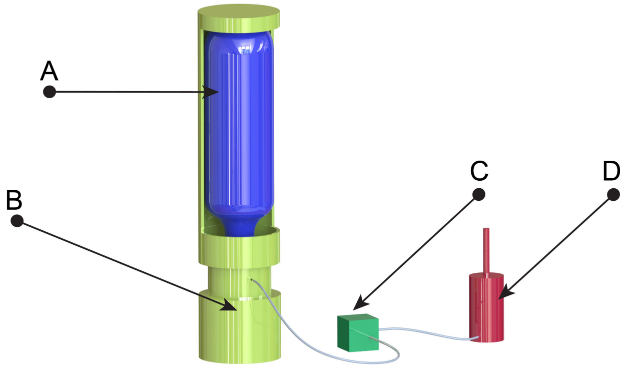

Fig. 2. Schematic view of original system with CO_2 cartridge (A), pressure regulator (B), control valve (C) and actuator (D)

weight of ca. 24 gram?, accepts 7.5 Grams unthreaded CO_2-cartridges with an empty weight of ca. 22 gram? and delivers a controlled pressure of $1.2[MPa]$. The flow it needs to deliver to the actuators is in the order of 1.5 [g/min]

Fig. 3. WILMER ® pressure regulator

The goal of the thesis becomes: *minimize the weight of a CO_2 source with the capability of supplying a gas flow of 1.5 [g/min] for 0.5[s] [3] with a pressure deviation of maximum 10% [9]* and a capacity of 8 grams.
This chapter gave an introduction in the current system and showed the problem with the weight of the current regulator. Chapter II outlines the general working principles behind pressure regulators and based upon these working principles concepts are proposed in Chapter III, relevant safety risks of these principles are shown in Chapter IV. A weight optimization strategy is elaborated in Chapter V. Chapter VI explains how using a prototype the most critical aspects of the designs are validated as well as parameters required for the downscaling strategy are estimated. Results of this testing are shown in Chapter VII followed by a discussion in Chapter VIII and a conclusion in Chapter IX.

II. Pressure Regulators

Pressure regulators perform an elementary feedbackloop (Fig. 4) with one controlled variable.

A. Functional decomposition

In case of the pneumatic prosthesis the desired 'Product' is *gas flow with a pressure deviation between predefined margins*; for an 'Integrated Storage Pressure Regulator' (IGSPR) delivering this flow is the primary function (Fig. 5).

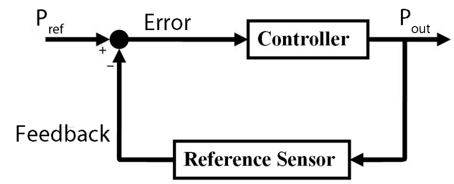

Fig. 4. Feedback loop of pressure regulator

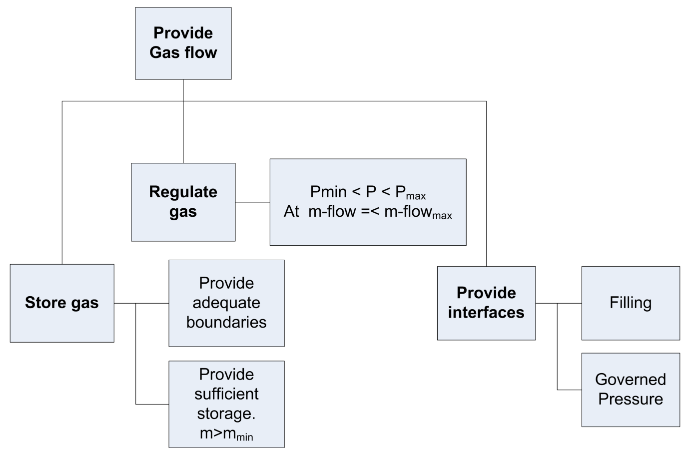

Fig. 5. Functional decomposition of pressure regulator including storage

In case of a pressure regulator which accepts disposable cartridges the storage function is no longer applicable and an interface for the cartridge is required (Fig. 6).

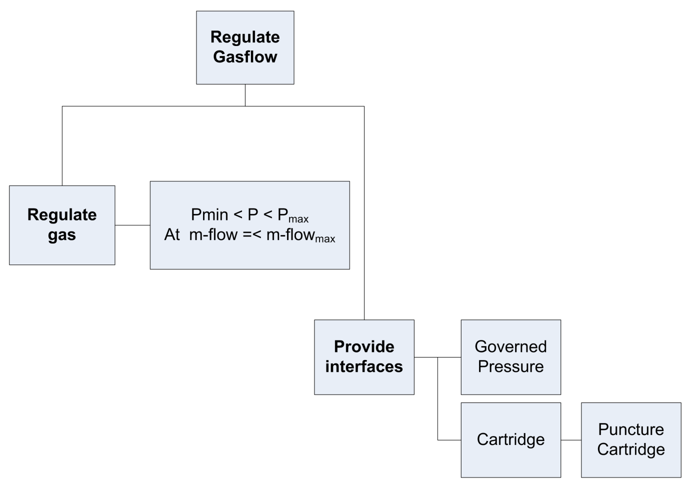

Fig. 6. Functional decomposition of pressure regulator

B. Electronic pressure regulators

Electronic pressure regulators [4] have a sensor, a controller, and a digital on/off valve in combination with an accumulator. When pressure drops below a predefined value the controller opens the valve until the desired pressure is reached.

C. Mechanical pressure regulators

1) Force balance: Mechanical pressure regulators have a valve, a force balance component (FBC), and an energy buffer

(EB) which applies a reference force (F_{ref}) on the FBC. The FBC is a component with one free DOF and is assymmetrically subjected to the output pressure (p_{out}) with respect to this DOF such that p_{out} exerts a force upon the FBC along this DOF. In order to achieve this the FBC gas to be partly subjected to the regulated gas (Fig. 1). The FBC applies a force upon the valve such that it will open the valve when pressure is too low, making the force balance of the FBC as in Eq. 1. Due to the low mass of the FBC compared to the forces acting upon it inertia effecs are assumed to be zero, leading to Eq 2. Parameters are to be such that logical expressions (Eq. 3) and (Eq. 4) hold. Using the ideal gas law Eq. 5 indicates how the pressure recoveres after a pressure drop as function of the input and outgoing gasflow. Here the input gasflow is assumed to be a function of the valve force, whereas the volume of the system is assumed constant.

$$m_{FBC} \cdot a_{FBC} = F_{ref} - A \cdot p_{out} - F_{valve} \tag{1}$$

$$F_{valve} = F_{ref} - A \cdot p_{out} \tag{2}$$

$$(p_{out} < p_{ref}) \rightarrow (F_{valve} > F_{valve-boundary}) \rightarrow (Valve - opens) \rightarrow (p_{out} = increasing) \tag{3}$$

$$(p_{out} = p_{ref}) \rightarrow (F_{valve} = F_{valve-boundary}) \rightarrow (Valve - just - closed) \rightarrow (p_{out} = p_{ref}) \tag{4}$$

$$\frac{dp_{out}}{dt} = \frac{(\dot{m}_{in}(F_{valve}) - \dot{m}_{out}) \cdot R_{kg} \cdot T}{V_{regulator}} \tag{5}$$

In order to achieve a netto area upon which p_{out} acts the regulated gas has to be negated on the EB side of the FBC. An approach to achieve this is by having an EB which inherently negates gas distribution on the bottom of the FBC (Fig. 7 - I). Possible EBs inherently negating gas on the bottom of the FBC include compressible foam, membrane encapsulated coil springs and balloons. Another approach is negating the regulated gas to reach the bottom of the FBC by impermeable dynamic boundaries between the FBC and the static boundaries (Fig. 7 - II). The dynamic boundaries should allow movement of the FBC relative to the static boundaries in direction of the free DOF of the FBC. Examples of designs using such boundaries are diaphragm based designs [5] and piston/O-ring based designs [6].

Current designs usually use material deformation in the form of coil springs as EB. Other possible methods include magnetic potential energy and compressed air energy storage.

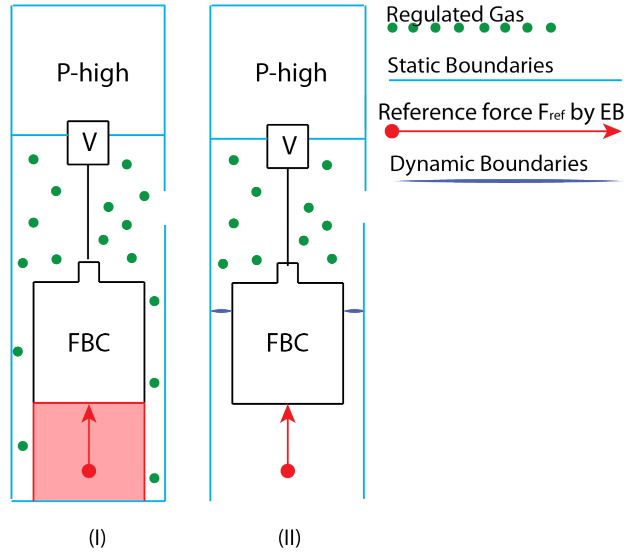

Fig. 7. Force balancing component (FBC) with a reference force applied by an energy buffer(EB) with such shape that it inherently prevents the regulated gas from affecting that side of the (EB) (I). FBC with a reference force applied by an EB with such shape that it does not inherently prevent the regulated gas from affecting that side of the (EB), which has dynamic boundaries between the FBC and the static boundaries (II). With valve(V) actuated by FBC.

III. Concepts

Electronic pressure regulators have the advantage of an electronically adjustable and very precise outlet pressure, however this is not considered of importance in pneumatic prostheses. The combination of actuator, sensor, circuitry and electric powersupply is generally heavier than a light mechanical regulator. Since the main objective of the paper is weight reduction and not precision, electronic regulators are not considered. In order to negate regulated gas below the FBC this paper explores the possibility of piston O-ring assemblies. As F_{ref} the EBs 'mechanical spring' and 'gas spring' are considered because the short comparison in Appendix A indicates these two options are most favorable from weight perspective. Other alternatives such as potential magnetic energy inherently lead to heavier devices due to the lower energy density of such energy buffers.

Also both designs with replaceable cartridges as well as designs with integrated gas storage are considered. Cartridges are considered because the possibility they bring of easy refilling without the need of an external device and have proven to be safe. Solutions with integrated gas storage are considered because the minimum weight is not bound by the weight of a cartridge.

A. Cartridge Based Designs

For a day of average operation the lowest capacity generic CO_2-cartridge size as available by manufacturer [8] is enough. The minimum cartridge capacity for non-threaded cartridges is 7.5 grams and the minimum cartridge capacity for threaded cartridges is 8 grams. In this paper only threaded cartridges are considered since non-threaded cartridges require a bracket for cartridge fixation making the regulator substantially heavier. The connection for the threaded cartridge (Fig. 8) consists of threading, a hardened hollow needle and O-ring. Screwing the cartridge in the connection section forces the needle through the cap allowing gas flow. The O-ring seals the flat end of the cartridge cap and the pressure regulator.

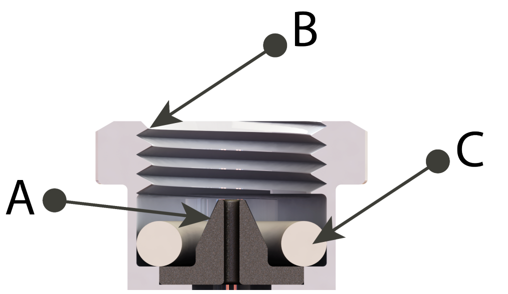

Fig. 8. Cross section view of threaded cartridge connection with hollow needle (A), 3/8-24 UNF threading (B) and 5.5x2.5 70Sh O-ring (C)

1) Pneumatic EB: This concept utilizes a gas reservoir as EB (Fig. 9) and a dynamic boundary between wall and piston in the form of two O-rings. The top O-ring seals the top chamber and the bottom O-ring encloses the pneumatic spring.

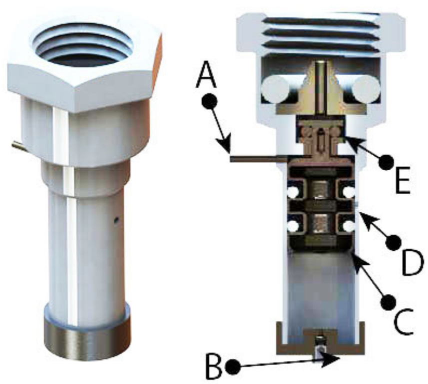

Fig. 9. Isometric/cross section view of cartridge based design using pneumatic force as reference with regulated pressure outlet (A), reference pressure filling (B), piston (C), overload outlet (D) and valve/valve-seat assembly (E)

2) Spring EB: This concept [Fig. 10] utilizes a spring as EB and a dynamic boundary in the form of an O-ring - wall assembly.

B. Integrated Gas Storage

The proposed IGSPR consists of a ball filled with saturated CO_2 and a regulator integrated within the ball.

1) Pneumatic EB: This concept is an IGSPR utilizing a gas reservoir as EB (Fig. 11), it has four connectors grouped on one end of the ball which fulfill the functions of high pressure filling, spring pressure filling, regulated output and overpressure.

2) Spring EB: This concept (not drawn) consists of a IGSPR utilizing a spring as EB.

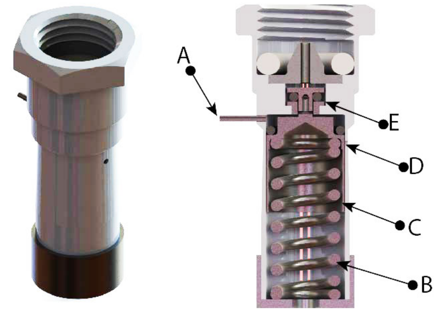

Fig. 10. Isometric/cross section view of cartridge based design using pneumatic force as reference with regulated pressure outlet (A), spring (B), piston (C), overload outlet (D) and valve/valve-seat assembly (E)

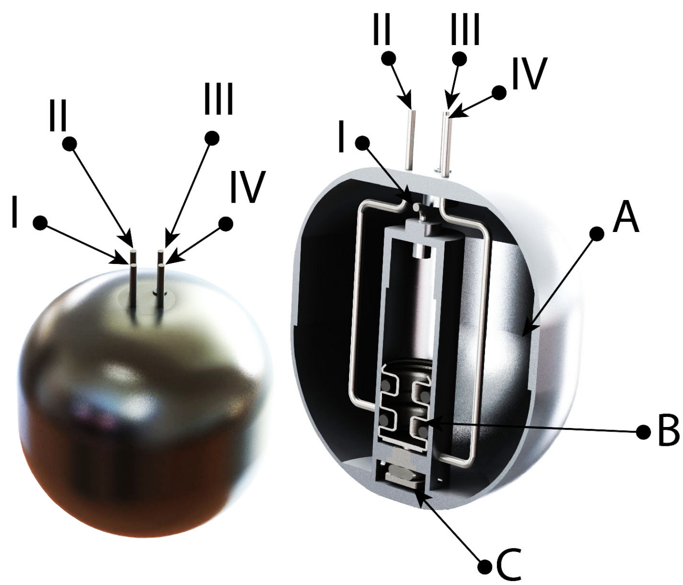

Fig. 11. Isometric view and cross-section of the IGSPR with connections for spring pressure (I), filling (III) output pressure (II) and overload (IV). With valve-seat assembly (C), piston (B) and high pressure CO_2 storage in hull (A)

3) Refilling: The IGSPR requires refilling , which can either be done at the residence of the client using a home-refill system or in an external location. Refilling in an external location requires the client to have multiple IGSPRs and an infrastructure to exchange empty for full devices. Because of costs accompanied with the infrastructure and the manufacturing of multiple regulators this option is deemed financially unfavorable so a home refill system is assumed. The precise configuration of a home refill system is outside the scope of this project, however a possible envisionment is given. The concept home refill system (Fig. 12) has a connection interface for generic 460 grams CO_2 canisters which can be used for multiple weeks. These canisters are available at local retailers for use in soda carbonators.The system has one knob to adjust the spring pressure (p_{FBC}) of the IGSPR and thus the operating pressure of the prosthesis. During every refill of the IGSPR the p_{FBC} should be updated again, such that the effect of a tiny leakage does not accumulate during the usage of the product. Connection with the IGSPR is done by an interface with single possible fit ensureing proper insertion, by insertion the refill procedure will be (mechanically) activated and when the IGSPR is refilled it can be tucked out and inserted in the prosthesis again. This concept works by pure mechanical means and no electricity is needed for operation. This design should be of a weight such that it will not move when connecting a new CO_2 canister single handed. Potentially a binary mechanical pressure indicator might be added which changes color when canister pressure drops below CO_2 saturation pressure, thus indicating the need for replacement.

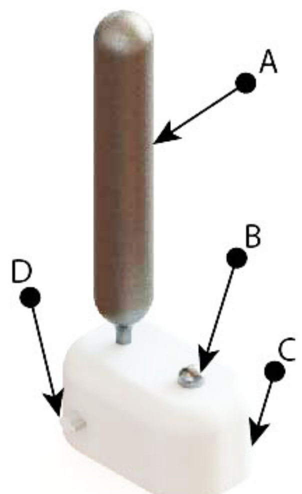

Fig. 12. Impression of home refill system consisting of base station (C) with 460 grams CO_2-canister (A), pressure adjustment knob (D) and a socket for IGSPR insertion (B)

IV. SAFETY

Three safety risks related to CO_2 suppliest are 'explosions', 'frost bites' and 'poisoning'. Poisoning is not considered of relevance on this scale.

A. Explosions

In case of IGSPRs a custom CO_2 storage with minimal weight is designed. Improper design may lead to explosive failure with the energy estimated in Eq. 6. Kinetic energy of shrapnell will be below the explosive energy and when shrapnell energy stays under 80J [15] it is considered non-lethal with a conservative safety margin. In case of the 8 grams IGSPR potential explosive energy is within this margin.

$$E_{explosive} = p_{CO_2-sat} \cdot V_{CO_2-storage} \tag{6}$$

B. Frost bite

During leakage the liquid CO_2 transforms in gas and solid flakes. This has a temperature of circa -78 degrees celsius and when it comes into contact with the skin it may lead to frost bite.

V. DOWNSCALING

This section supplies a general strategy upon downscaling of a pressure regulator based upon the specific demands of the application. The most relevant design parameters are stated below.

A. General Configuration

The general working principle and configuration of components defines most weight boundaries. The usage of an internal gas storage gives the opportunity of designing a lighter CO_2-storage than the original 22 grams (empty weight) cartridges. Designs utilizing a pneumatic FBC force can potentially be lighter due to the lack of a spring. In this paper configurations using the valve-piston working principle are used, however it is most likely that other (non-discovered) more favorable configurations exist. For now this paper assumes the IGSPR with gas spring to be the lightest configuration possible.

B. Material choice

When choosing a material for the housing of a pressure regulator or IGSPR several aspects have to be taken into account [10]; a global material comparison of relevant materials is given in Table I.

1) Specific strength: increase leads to a lower weight design and should be maximized

2) Workability: of the material should be such that the desired shapes can be achieved with the required tolerances within available budget.

3) Failure modes: In case of IGSPRs the pressure vessel component can be assumed a safety critical device and explosive failure modes may lead to injury of the user of the device. The material should be such that 'Leak Before Cracking' [11] occurs instead of 'explosive' failure modes.

4) Thermal conductivity: The thermal energy required for the phase change will mainly be extracted from the expansion chamber above the piston since phase change is expected to take place at here. By utilizing a material with a higher heat transfer coefficient, thermal energy will spread more throughout the regulator and the total surface of the pressure regulator can be utilized more effectively for heat transfer with environment.

5) Consideration: Table I suggests that titanium is the most favorable material when taking the regulator functionality as guideline (specific strength, failure modes and temperature conduction). Aluminum 7075 T6 is the most favorable material from these three out of the perspective of producability.

	CFRP	Ti	Al
Specific strength	++	+	O
Workability	-	O	++
Failure modes	+*	++	+
Temperature conductivity	O	+	++

TABLE I
GLOBAL MATERIAL COMPARISON FOR A PRESSURE REGULATOR.
*ASSUMING PROPER FIBER ALIGNMENT [12]

C. Size reduction

1) Thermal effects of general size reduction: Reduction in volume and mass of the pressure regulator influences its thermal properties. When delivering short pulses with (very) high flow velocities it is assumed that heat transfer from the pressure regulator to the environment will be negligible compared to the enthalpy change of the CO_2-flow and resulting heat transfer from the CO_2-flow to the pressure regulator. In this case it is assumed that thermal energy required for phase change comes from the thermal buffer of the pressure regulator and that thermal behavior can globally be estimated using Eq. 7. Designs using an integrated gas storage are assumed to always have a minimum reserve of CO_2, which can also be seen as a thermal energy buffer. When delivering long periods of lower flow velocities it is assumed that the heat buffer effect becomes negligible and the thermal behavior can be assumed using the heat conduction equation: Eq. 8.

$$\overbrace{\dot{m}\cdot(h_{sat}-h_{out})}^{Enthalpy\ change\ CO_2} = \overbrace{\frac{dT}{dt}\cdot(m_{reg}\cdot c_{p-reg}+m_{CO_2}\cdot c_{p-CO_2})}^{Heat\ Absorption} \tag{7}$$

$$\overbrace{\dot{m}\cdot(h_{sat}-h_{out})}^{Enthalpy\ change\ CO_2} = \overbrace{h_c\cdot A_{reg-surface}\cdot(T_s-T_e)}^{Conduction} \tag{8}$$

When thermal effects as result of short pulses become restrictive for further minimization an IGSPR design can lead to a lighter solution due to the heat buffer the CO_2 provides.

2) Wall thickness reduction: The minimum wall thickness is restricted by strength requirements as well as producibility.

Producibility depends strongly on the desired process which strongly depends upon the desired batch size and unit price. The hypothetical minimum dimensions out of strength perspective are estimated by hand calculations of pressure vessel weights. Minimum weight of the pressure regulator is estimated using Eq. 9 by assuming the pressure regulator body to be a pressure vessel consisting of a cylinder and two hemispherical ends, which should just be able to hold the maximum internal pressure P_{int}. This pressure can be either p_{FBC} for the design with pneumatic reference force of p_{out} for a design with a spring reference force. The valve is considered a fixed weight component, the piston is considered a cylinder with a hemispherical end just strong enough for the maximum internal pressure, the CO_2 connection is considered a cylinder with a hemispherical bottom with such dimensions that it can fit the end of a CO_2 cartridge and is just strong enough to hold the saturation pressure contained within the cartridge (Fig 13 - I). Finally the optional spring is considered a fixed weight component. The length of the straight section of the piston is assumed to be $2\cdot R$ as by heuristics pistons may jam if the diameter is bigger than the length of the sliding area.

The IGSPR (Fig. 13 - II) is calculated as the regular cartridge based pressure regulator with the differences that the CO_2 connector is replaced by a CO_2-vessel and the pressure the housing of the regulator should be able to withstand is now increased to the saturation pressure of CO_2 (Eq. 10).

$$\begin{aligned} m_{min} = & \\ & \overbrace{2\cdot\pi\cdot R^2\cdot(R+W)\cdot P_{int}\cdot\frac{\rho}{\sigma}\cdot f_{p-int}}^{body} + \overbrace{m_{valve}}^{Valve} + \\ & \overbrace{\pi\cdot R_{cart}^2\cdot(R_{cart}+2\cdot l_{cart})\cdot P_{CO_2-sat}\cdot\frac{\rho}{\sigma}\cdot f_{CO_2-sat}}^{CO_2-connection} + \\ & \overbrace{\pi\cdot R^2\cdot(R+4\cdot R)\cdot P_{int}\cdot\frac{\rho}{\sigma}\cdot f_{p-int}}^{Piston} + \overbrace{m_{spring}}^{optional:spring} \end{aligned} \tag{9}$$

$$\begin{aligned} m_{min} = & \\ & \overbrace{2\cdot\pi\cdot R^2\cdot(R+W)\cdot P_{CO_2-sat}\cdot\frac{\rho}{\sigma}\cdot f_{CO_2-sat}}^{body} \\ & + \overbrace{m_{valve}}^{Valve} + \overbrace{\pi\cdot R^2\cdot(R+4\cdot R)\cdot P_{int}\cdot\frac{\rho}{\sigma}\cdot f_{p-int}}^{Piston} + \\ & \overbrace{\frac{3}{2}\cdot P_{CO_2-sat}\cdot(\frac{m_{CO_2-max}}{\rho_{CO_2-s+at}}\cdot f_{CO_2}+\frac{4}{3}\pi\cdot R^3+R^2\cdot\pi\cdot W)\frac{\rho}{\sigma}}^{Gasstorage} \\ & \overbrace{\cdot f_{CO_2-sat}}^{Safety\ factor\ gasstorage} + \overbrace{m_{spring}}^{optional:spring} \end{aligned} \tag{10}$$

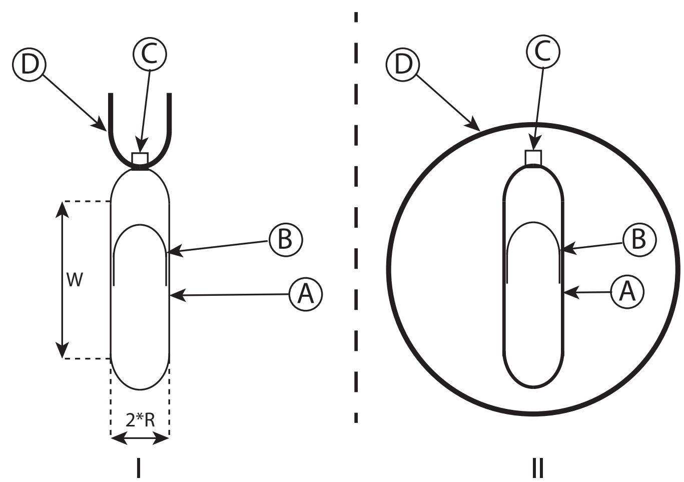

Fig. 13. Sketch of pressure regulator (I) with body (A), piston(B), valve (C) and CO_2 connection (D); Sketch of IGSPR with body (A), piston (B), valve (C) and CO_2-container (D)

The hypothetical formulation assumes an ideal ball. Effects like stress concentrations at interfaces of the capillaries with the ball of the IGSPR as in the practical application can lead to failure at significantly lower pressures as globally indicated by the FEM-sketch (Fig. 14) [COSMOS ®using Tetrahedical elements and surface pressure].

3) Cylinder length reduction: Shortening of the cylinder requires a shorter EB. This generally has as consequence a higher force deviation of the trajectory to open the valve (Δx). The pressure deviation is calculated as in Eq. 11. In this estimation the wall thickness of the piston if not taken into account nor the volume in the hemispherical section of the pressure regulator. For the pneumatic cylinder with

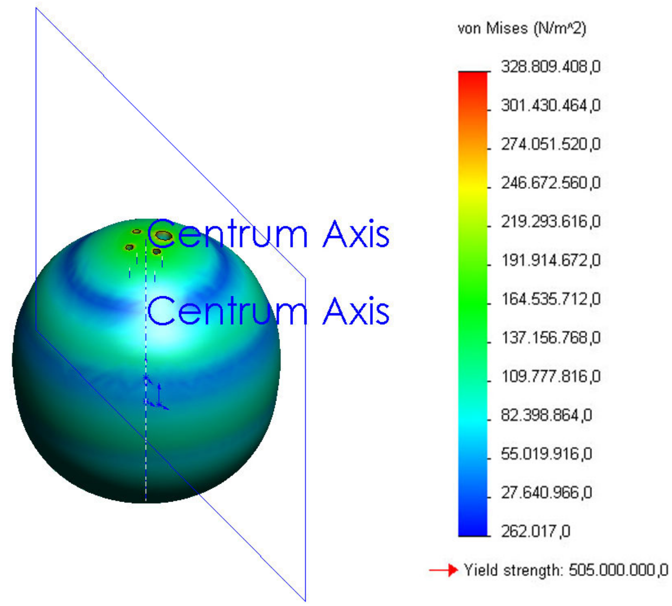

Fig. 14. Sketch of IGSPR using COSMOS®with Tetrahedical elements

configuration (Fig. 15) the pressure deviation can (roughly) be estimated using the ideal gas law, leading to Eq. 12

$$\Delta P_{length-spring} = P_{ref} \cdot (1 - \frac{l_{spring} - ls0}{\Delta x + l_{spring} - ls0}) \quad (11)$$

$$\Delta P_{length-pneumatic} = P_{ref} \cdot (1 - \frac{ls0}{ls0 + \Delta x}) \quad (12)$$

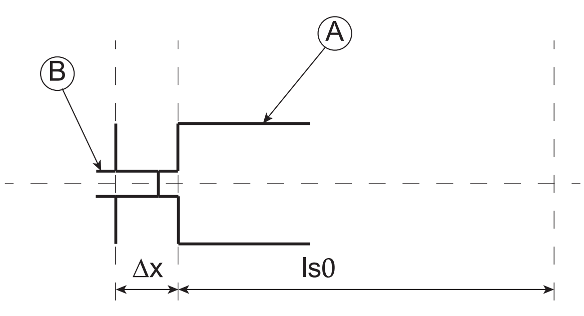

Fig. 15. Schematic sketch of piston (a) and valve(b) at which a (pneumatic) spring is used with length ls0 and trajectory Δx

4) Diameter reduction: Reduction in diameter of the piston leads to a quadratic reduction of the reference force, while the friction forces reduce linearly and the valve force remains constant as shown in Eq. 13. In this equation it is assumed that the pressure deviation (ΔP) is a function of the force needed to get the valve from 'just closed position' to 'open' and the friction force of the piston. The friction force of the piston is assumed to be factor multiplied with the circumference and the required force for valve actuation is assumed a constant. For weight reduction the diameter has to be chosen such that it is just big enough not to deliver a pressure difference above the allowed value.

$$\Delta P_{diameter} = \frac{2 \cdot R \cdot \pi \cdot C_f + F_{valve-actuation}}{R^2 \cdot \pi} \quad (13)$$

A possible limitation in diameter reduction is the availability of springs for designs with a spring F_{ref}. The following prerequisites should be met: maximum spring force $> F_{ref}$ and spring radius $<$ R.

5) Valve size reduction: The force required for opening the valve correlates with the diameter as estimated in Eq. 14 which estimates the force as function of the pressure difference and dimensions. This equation expects the O-ring to perform like a binary closing device. In practice however the required force will be less due to microscopic imperfections in the O-ring. Estimation of the valve actuation force when taking this into account can be done via experimental data and extrapoliation as in Eq. 15 which scales experimental results.

$$F_{valve} = \pi \cdot (Rw_{O-ring} + Ri_{O-ring})^2 \cdot (P_{CO_2-sat} - P_{out}) \quad (14)$$

$$F_{valve} = F_r \cdot \frac{(Rw_{o-ring} + Ri_{o-ring})^2}{(Rrw_{o-ring} + Rri_{o-ring})^2} \quad (15)$$

In order for the valve not being the restricting factor in flow velocity a possible guideline is taking the cross sectional flow area of the valve to be slightly larger than the output capillary flow area. For weight reduction the diameter of the valve has to be chosen such that it allows just enough flow.

D. Force balance

In order to have proper control the force balances in static position have to be met (Eq. 16) with either the spring force (Eq. 17) or the pneumatic force (Eq. 18)

$$F_{res} = F_{valve-boundary} + \pi \cdot R^2 \cdot p_{ref} \quad (16)$$

$$F_{res-spring} = (l_{spring} - ls0) \cdot k \quad (17)$$

$$F_{res-pneumatic} = p_{FBC} \cdot \pi \cdot R^2 \quad (18)$$

E. Optimization Strategy

This subsection supplies a simple design/optimization strategy which can be used as guide for adapting the designs in this paper to the specific needs of the application.

Choose design criteria:

1) Flow duration (t_{flow})
2) Output pressure P_{out}
3) Flow velocity ($\dot{m}$)
4) Desired configuration upon demands. Spring/Pneumatic and Integrated Storage/Cartridge. Further implications of these choices are elaborated in Appendix C
5) Desired materials, based upon price, production and weight requirements. Long term operation is dependent

upon the wear characteristics between the chosen materials. Note that calculations are only valid for homogenous metals. Density ρ and maximum work stress σ

6) Acceptable total pressure deviation $\Delta P_{total} = \Delta P_{length} + \Delta P_{diameter}$
7) Safety margin for components subjected to P_{int} and P_{CO_2-sat}; f_{p-int} and f_{CO_2-sat}
8) Cartridge threading for cartridge type regulators with radius R_{cart} and length l_{cart}
9) Capacity m_{CO_2} and volumetric safety factor f_{CO_2} for IGSPRs
10) Desired O-ring configuration with resulting friction coefficient C_f and thermal operating range
11) Working temperature range and resulting P_{CO_2-sat}
12) Spring type (usually iterative as function of other steps) with resulting mass m_{spring}, stiffness K, neutral length l_{spring} and pretensioned length $ls0$

Design:

Design an optimal valve for the intended flow requirements given available O-ring sizes. This determines the static valve force $F_{valve-static}$ at which the valve just does not release gas, the valve actuation force $F_{valve-actuation}$ as well as the valve mass m_{valve}.

Optimizing:

1) In case of high flow velocities and significant pulse times the heat capacity or the heat transfer between environment and regulator may become restrictive, in which case further downscaling becomes a heat transfer problem which is considered outside the scope of this paper.
2) (Spring as EB): Verify available springs for the prerequisites: 'maximum spring force' $> F_{ref}$ and Radius of Spring $<$ R. Experience dictates that for higher pressures the available springs might become restrictive and further optimization will be limited to the available components.
3) Utilizing the design parameters and the estimation of ΔP_{total} (Eq. 11), (Eq. 12), (Eq. 13)
(with assumptions $ls0 = W - \Delta x$ an expression of R in W can be obtained
4) Utilizing the design parameters and mass equations (Eq. 9) or (Eq. 10) an expression of R in W can be obtained.
5) by calculating the optimum of these Equations global design parameters can be obtained.
6) Using the design goals of this specific pneumatic prosthesis and the parameters acquired from prototype testing a two surface plots are generated with Matlab ® (Appendix D). Detailed parameters are explained in the code. One plot estimated pressure deviation as function of R and W (Fig. 16) and the other plot estimates mass as function of R and W (Fig. 17). In the mass plot the red circles indicate combinations of R and W at which a pressure deviation of 60kPa is estimated. This can be used as guideline for minimum weight design; for different designs parameters can be adjusted in the m-file.

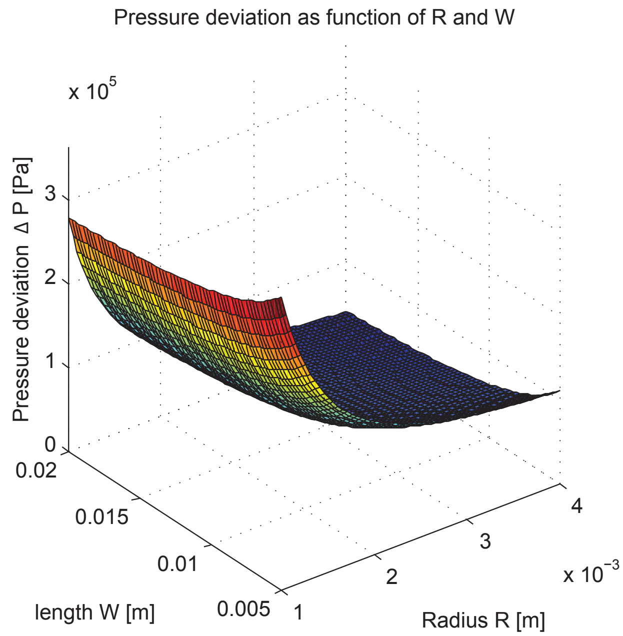

Fig. 16. Estimated pressure deviation ΔP as function of R and W

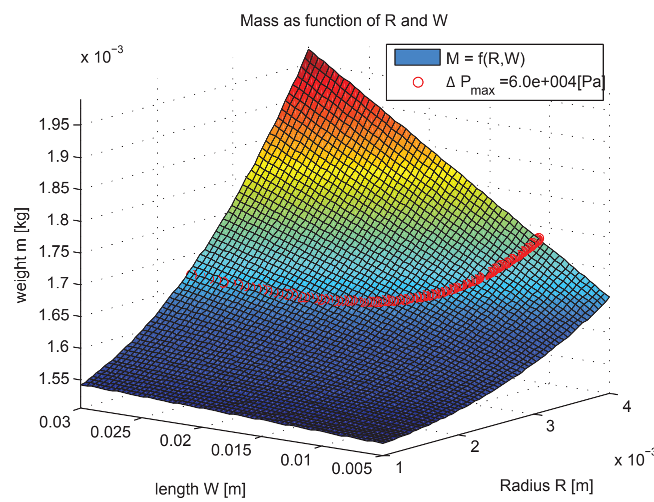

Fig. 17. Minimum Mass as function of R and W

VI. Validation & parameters acquisition

From the proposed concepts the assumed critical design features are piston sealing when using a pneumatic EB, sealing of the threaded CO_2 cartridge using this puncture system and finally operation in general of both the pneumatic as well as coil spring configuration. In order to validate these principles a prototype is constructed with a connnection for threaded cartridges. It has the possibility to use either a pneumatic or coil-spring through replacable pistons and bottom plate. Also design parameters for further downscaling are acquired from the prototype; the valve actuation forces and the piston friction forces. The different set-ups of the prototype are elaborated in Appendix B and construction drawings of the prototype are to be found in Appendix F.

VII. Results

A. Prototype

Results are obtained from prototype as shown in [Fig. 18], detailed production drawings are to be found in Appendix F

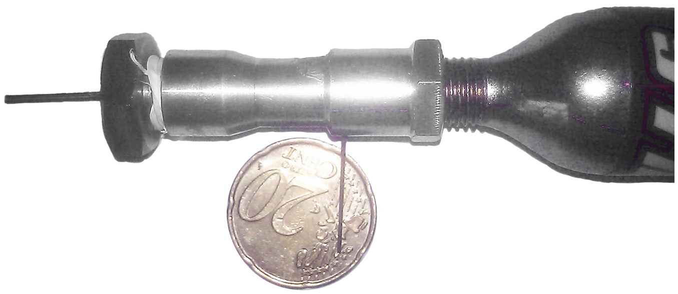

Fig. 18. Prototype in pneumatic configuration connected to a 16 grams CO_2 cartridge and a 20 Euro Cent coin for comparison.

B. Valve actuation force

The force required to actuate the valve of the pressure regulator is measured for both 'just starting to open' ($F_{valve-boundary}$) as well as 'fully opened' ($F_{valve-open}$) condition are shown in Table II. The setup used is described in [Appendix B-B5]. During full flow the gas consumption was significant and ice formed in seconds.

Flow type	$F_{actuation}$	No measurements	σ
$F_{valve-boundary}$	15 [N]	3	0.3[N]
$F_{valve-open}$	18[N]	1	[-]

TABLE II
Measurement results of required force for valve actuation for both 'just beginning flow' as for full flow.

C. Piston friction

The friction force of the piston is estimated for semistatic movement. A force will be applied to the piston such that it just starts to move; the test setup is described in Appendix B-B6. Both the piston for the spring configuration with 0% compression as well as for the pneumatic configuration with 8% compression are tested; results are shown in Table III.

Piston configuration	$F_{friction}$	No measurements	σ
Top O-ring 8 % comp.	0.9 [N]	3	0.1[N]
Bottom O-ring 8% comp.	0.7[N]	3	0.1 [N]
Double O-ring 8% comp.	1.7[N]	3	0.3 [N]
Top O-ring 0 % comp.	0.13[N]	3	0.03 [N]

TABLE III
Measurement results for the semi-static friction forces between the cylinder and piston using different configurations with 6.5x1.0x 70 Shore O-rings and both with 8 % compression as without compression

D. Static Leakage along piston

Static leakage is measured using the experimental set-up as described in Appendix B-B4. A 12 hour test with a pressure gradient of 1.2[MPa] did no deliver any significant volume for the configuration with single O-ring with 8% compression.

E. dynamic leakage

Leakage from the gas spring compartment did occur some times. After lubrication of the O-rings with silicon grease this leakage was strongly reduced and unnoticable in measurements.

F. Free flow velocity

Using a test setup with a high capacity regulator and comparable tubing as the light pressure regulator as described in Appendix B-B1 the flow velocity is estimated for a reference pressure of 1.2[MPa]. The measured maximum flow is circa 6 [gram/min]. The free flow velocity can be used to compensate for the pressure drop as result of the flow resistance of the tubing. The flow resistance can be estimated with Eq. 19 using zero flow p_{out} and $\dot{m}$ at free flow. The compensation factor P_{comp} can be estimated using Eq. 20 with the flow velocity of the measured pulse $\dot{m}$ It has to be noted that the test setup is not designed for high flow velocities and this measurement may contain a significant deviation.

$$R = \frac{P_{out}}{\dot{m}} \tag{19}$$

$$P_{comp} = R \cdot \dot{m} \tag{20}$$

G. leakage of CO_2-cartridge connection

The connection between the CO_2-cartridge and the prototype is tested by keeping the system under water in order to see if bubbles form, which was not the case. Also the contents of the CO_2 cartridge did not deplete or reduce notably after one week of storage while being connected to the pressure regulator. From both experiences it is concluded that the CO_2 connection has no (significant) leakage.

H. Operation using pneumatic spring

Testing of the pressure regulator in pneumatic configuration is described in Appendix B-A5. Tests are done for both a low flow velocity (Fig. 19), as well as two pulses with measured flow velocity (Fig. 20),(Fig. 21). Finally a test is conducted with five very short maximum flow pulses for which no flow velocity measurement is conducted (Fig. 22).

I. Operation using mechanical spring

Operation of the pressure regulator in spring configuration as described in Appendix B-A7 both with a low flow velocity (Fig. 24) and with three pulses (Fig. 25).

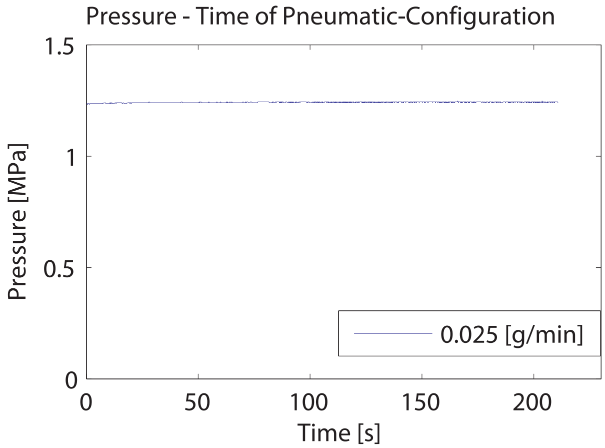

Fig. 19. Pressure - Time diagram of the prototype pneumatic configuration during a constant airflow.

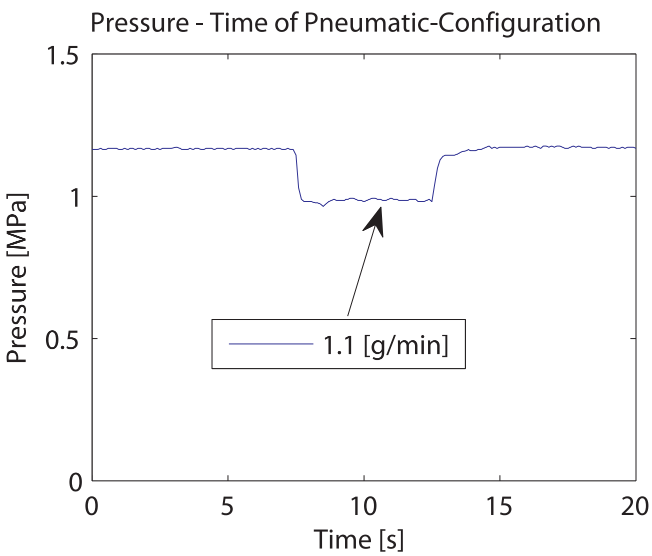

Fig. 20. Pressure - Time diagram of the prototype in pneumatic configuration during one pulse.

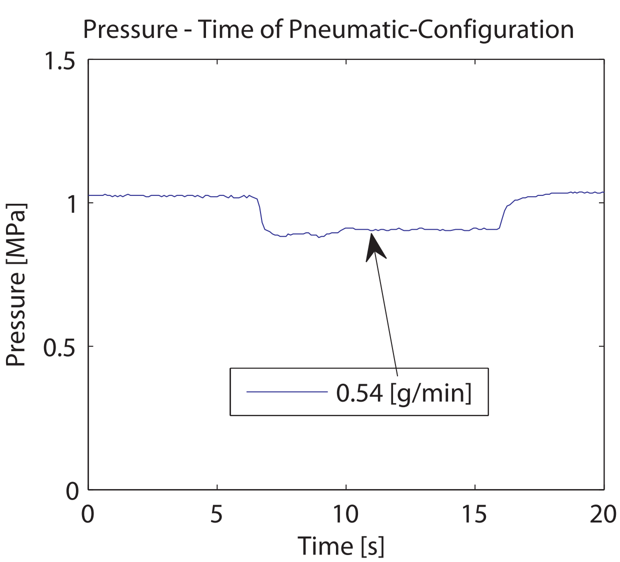

Fig. 21. Pressure - Time diagram of the prototype in pneumatic configuration during one pulse.

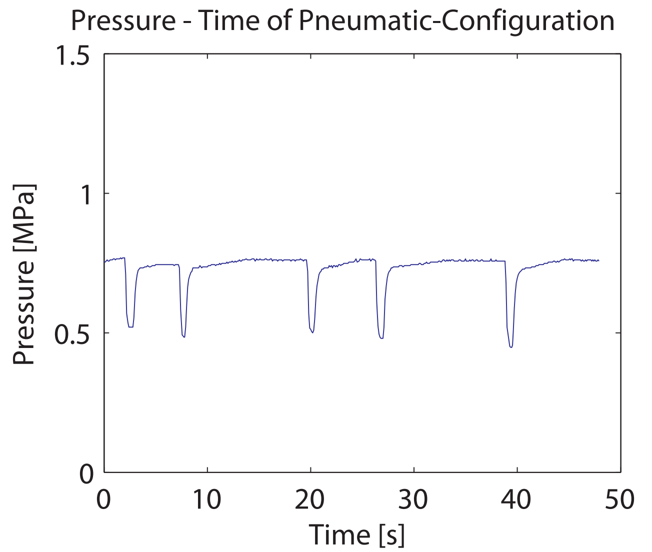

Fig. 22. Pressure - Time diagram of the prototype in pneumatic configuration during 5 pulses.

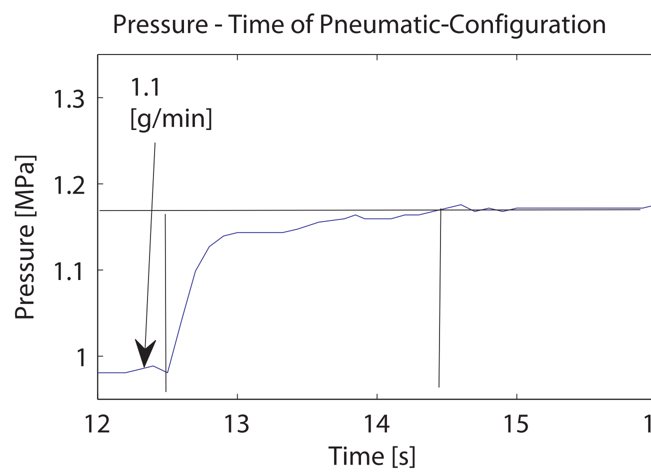

Fig. 23. Cut-out of Pressure - Time diagram in [Fig 20]

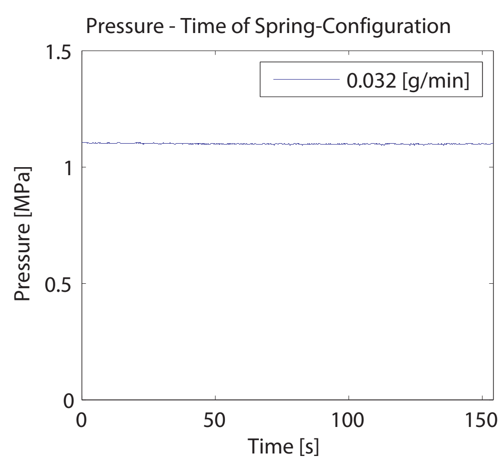

Fig. 24. Pressure - Time diagram of the prototype in spring configuration during a constant airflow

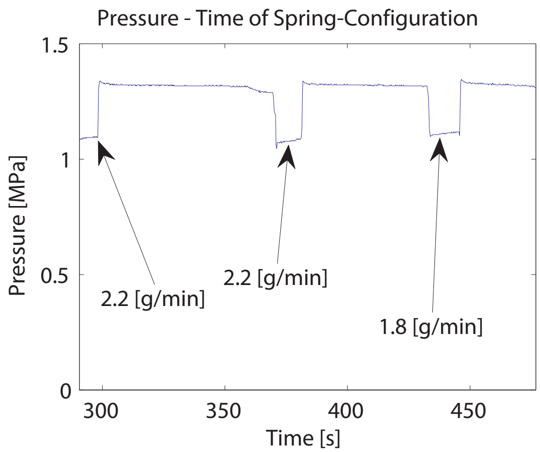

Fig. 25. Pressure - Time diagram of the prototype in spring configuration during three pulses with zero flow in between

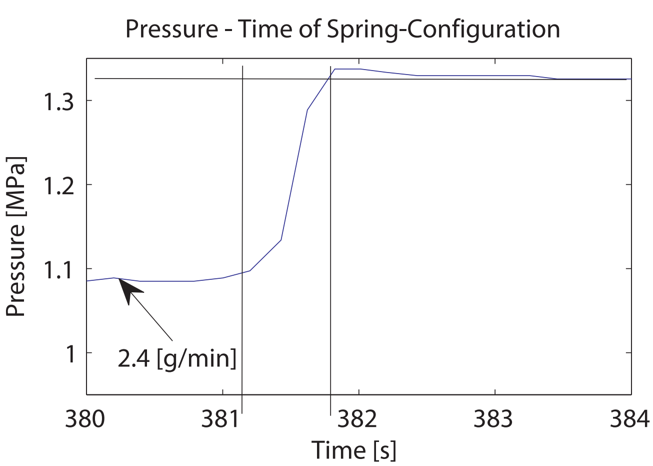

Fig. 26. Cut-out of Pressure - Time diagram in [Fig 25]

VIII. Discussion

Both the pneumatic design and spring design were able to control gas flow. From weight perspective the IGSPR in pneumatic spring configuration is most favorable. This paper showed that the critical design criteria with pneumatic spring regulation are met, making this design feasible. It has to be noted that dynamic leakage occured at some measurements while at other measurements (after applying grease) it was absent. More research has to be conducted on verifying this effect in order to assure reliable operation. The steady state error could not reliably be estimated due to deviations in the measurement setup. Data seems to indicate though that pressure deviation is within the design criteria.

IX. Conclusion

When observing the data it can be concluded that both pressure regulators using a pneumatic spring as well as a mechanical spring deliver a constant pressure given a low flow demand. In the case of the pneumatic spring [Fig 19] as well as the mechanical spring [Fig 24] no significant pressure deviation could be observed during the flow. Also both the configuration with pneumatic spring as well as the configuration with mechanical spring restore governed pressure to 90% of the pre-pulse pressure within 0.5[s]. The free flow velocity was *not* an order of magnitude higher than the measured flows, so the internal flow resistance of the capillaries leads to a significant pressure drop and this data can not be used for estimation of steady state error at higher flow velocities. When compensating for the flow resistance using equations Eq. 19 and Eq. 20 the pressure deviation falls within the 10% margin and the design obective could be seen as achieved. From the data it can be concluded that both the spring based as pneumatic based design are able to control a CO_2 flow to a stable output pressure. From the lack of leakage of the CO_2 cartridge it can be concluded that this cartridge connection method is effective.

X. Recommendations

Recommendations are based upon available research time and budget. An overview of estimated weight of different solutions is given in Fig. 27

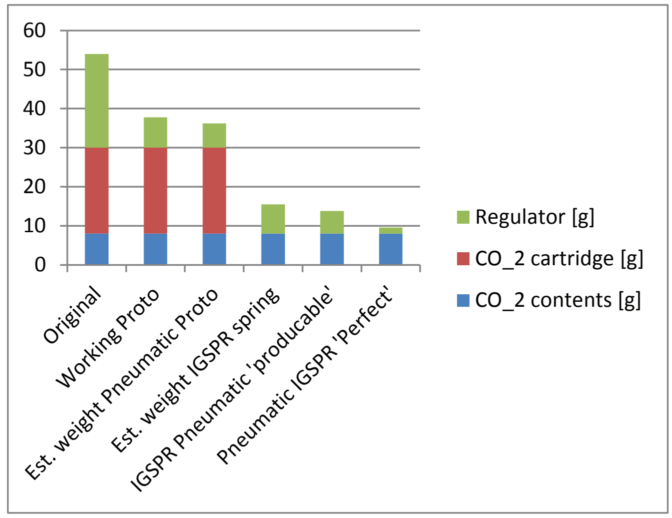

Fig. 27. Comparison of estimated weight of different concepts

A. Low budget

Given a (very) low budget, circa 1 week full time equivalent salary of an engineer, it is recommended to use the spring based pressure regulator utilizing external cartridges as shown in the construction drawings in Appendix F. Performance of the device was acceptable with and with a weight of circa 7.8 grams for the device a relatively light option compared to the original regulator. Further optimization of cartridge based pressure regulators is not recommended, since the cartridge weight of circa 22 grams makes the potential for relative weight savings very small. Also the connection interface with needle requires the regulator to be 'large', thus limiting further downscaling. It has to be noted that pressure deviation at higher flow velocities became high due to the flow resistance of the capillary tubes. This can be taken care of with a larger tube. Further it is recommended to do durability tests before implementing the device in critical applications.

B. Medium budget

Given a significant budget, circa 1 month full time equivalent salary of an engineer, it is recommended to construct an IGSPR with a spring and optimize for the desired application using the hand calculations from section V and/or the Matlab ®script. For this also a refill device has to be envisioned and proper testing has to be done in order to guarantee safety of the pressure vessel. Further research and testing is required to guarantee long time performance of check valves and tube sealings. It has to be noted that with this concept the combination of storage and regulator probably has a lower mass than the CO_2 contents to be stored. Further minimization of the gas supply will be bound by the CO_2 weight.

C. High budget

Given a very high budget, circa 1 year full time equivalent salary of an engineer, it is recommended to develop an IGSPR with pneumatic F_{ref} and reconsider the configuration of the pressure regulator by analysis of other options including membrame based FBC's. A proper refill station can be conceived which provides ergonomic and safe refilling of the device. Optimization can be done using a multiparameter optimization and multi-physics model utilizing desired design criteria.

D. General recommendations

Finally it is recommended to wear protective glasses and long sleeves during experimentation with a prototype using CO_2 in saturation pressure. If adaptors disconnect from the hoses they may fly very fast and cause injury.

XI. Constants and variables

This section provides an overview of constants and variables used in the equations.

Variable/Constant	Description
a_{FBC}	Acceleration of Force Balance Component in its free Degree of Freedom
$A_{reg-surface}$	Surface of pressure regulator considered for heat transfer with environment
c_{p-reg}	Specific heat capacity of pressure regulator
C_{p-CO_2}	Specific heat capacity of saturated CO_2
C_f	Radius dependend friction coefficient between piston and housing given a certain O-ring configuration
$E_{explosive}$	Energy released when IGSPR explodes
f_{CO_2-sat}	safety factor of components subject to saturation pressure of CO_2
f_{p-int}	Safety factor for components subject to internal pressure
F_{valve}	Force applied upon valve
$F_{valve-boundary}$	Force required to just get valve to 'just opening/closing' position
$F_{valve-actuation}$	Force required for valve actuation; assumed to be $F_{valve-open} - F_{valve-boundary}$
$F_{valve-open}$	Force required to fully open the valve
$F_{FBC-spring}$	Force a spring energy buffer applies on force balancing component
$F_{FBC-pneumatic}$	Force a pneumatic energy buffer applies on force balancing component
F_{res}	Resultant force on the 'force balance component'
F_{ref}	Reference Force on 'force balance component' by EB
h_{sat}	Enthalpy of saturated CO_2
h_{out}	Enthalpy of outgoing CO_2 flow
h_c	Heat transfery coëfficient between pressure regulator and environment
l_{cart}	Length of the connection part of the cartridge
l_{spring}	Untensioned spring length
$ls0$	Assumed length of the pressure regulator, excluding the 'Δx'; a zero wall thickness of the piston is assumed
m_{valve}	Mass of valve
m_{spring}	Mass of spring
m_{CO_2-max}	Mass of maximum amount of CO_2 to be stored within an IGSPR
m_{reg}	Mass of pressure regulator
m_{CO_2}	Mass of CO_2 assumed to be everpresent
m_{FBC}	Mass of Force Balance Component
$\dot{m}$	Mass flow of regulated CO_2
$\dot{m}_{in}$	massflow from saturated CO_2 storage to pressure regulator
$\dot{m}_{out}$	massflow from pressure regulator to pneumatic system
p_{CO_2-sat}	Assumed saturation pressure of CO_2
P_{out}	Output pressure of the pressure regulator
P_{ref}	Reference pressure of outgoing gasflow
p_{FBC}	pressure of the 'gas spring compartment'
P_{int}	Maximum internal pressure of pressure regulator, either p_{out} or p_{FBC}
$\Delta P_{length-spring}$	Pressure deviation as result of the not-infinite spring length
$\Delta P_{length-pneumatic}$	Pressure deviation as result of the non-infinite gas reservoir length
R	Radius of pressure regulator housing and piston. A zero wall thickness is assumed
R_{kg}	Gas constant, [kg] based
Rw_{O-ring}	Cord width of O-ring
Ri_{O-ring}	Internal diameter of O-ring
Rrw_{O-ring}	Cord width of reference O-ring
Rri_{O-ring}	Internal diameter of reference O-ring
ρ	Densitity of material of pressure regulator
ρ_{CO_2-sat}	Density of saturated CO_2
σ	Maximum work stress of material of pressure regulator
T_s	Surface temperature of pressure regulator
T_e	Environment temperature around pressure regulator
T	Gas temperature in pneumatic system
$v_{CO_2-storage}$	Volume of CO_2 storage
V_reg	volume of regulator
$\frac{dT}{dt}$	Temperature change rate of pressure regulator
W	Length of pressure regulator housing. A zero wall thickness is assumed
Δx	Movement of piston and valve when moving from 'just closed' to 'open' position

TABLE IV
OVERVIEW OF CONSTANTS AND VARIABLES

ACKNOWLEDGMENT

The author would like to thank:
Dick Plettenburg for being my supervisor
Jan van Frankenhuyzen for the great advice upon the prototype
Gerwin Smit for his useful remarks
John Dukker for the skillful construction of high precision prototype parts
Jos van Driel for his advice upon the measurement setup
Diones Supriana & Anouk de Goede for keeping an eye upon the administration
Friends & crew from the compliant laboratory for providing an intellectually stimulating environment

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APPENDIX A
WEIGHT ESTIMATION DIFFERENT SOURCES OF F_{ref}

Assumption 1: For the regulation a F_{ref} of 50N is required. Assumption 2: The weight difference between the housing/piston for the spring, magnetic and pneumatic based concepts is negligible. The only weight difference lies withing the spring and magnets. Assumption 3: The weight for the gas of the spring can be neglected

Magnetic
The weight needed for a magnetic configuration with a constanf reference force over a preset trajectory is unknown since this configuration has not been envisioned yet. However a magnet with a holding force of 45N already has a weight of $6.7 \cdot 10^{-}3[kg]$ and for a constant force over a certain directory a configuration is needed which is heavier than this.

Pneumatic
Pneumatic concepts have no additional weight

Spring
Tevema 22120 Spring m = $1.67 \cdot 10^{-}3[kg]$

APPENDIX B
TEST SETUPS

For the experimental validation different setups are used. For the different setups some sets identical (sets) of components are used. In order to be able to facilitate drawing these subsets are replaced by blocks as shown in section B-A and the complete setups are represented using these blocks.

A. Blocks

1) Pressure Measurement: Pressure is measured digitaly by using a Feteris Components DMP331 pressure sensor, connected to a laptop running LabView®2009 through a National Instruments NI USB-6008 AD Converter [Fig. 28]. The Reference voltage over the sensor is supplied by a Sontronics AD-2400400R 24V AC/DC Adaptor. Connection of the tubes to the sensor is done through a custom part with 1/4" G threading for connecting the pressure sensor and two $\varnothing 0.65mmx0.25mm$ cannulae for connecting the (overstretched) $\varnothing 1.0mmx0.50mm$ tubes.

2) Volume/flow measurement: Volume is measured using a Simax 50ml measurement cylinder top side down in a container filled with tapwater (Fig 29). The flow which is to be measured is lead through a $\varnothing 1.0mmx0.50mm$ tube under the measurement cylinder, such that the gas will form a volume under the cylinder and can be measured as such. Flow will be measured by dividing the volume over the time measured using an online stop watch e.g. "http://www.online-stopwatch.com/".

3) FlowControl: Flow is controlled using a finely adjustable tube clamp (Fig. 30) around the $\varnothing 1.0mmx0.50mm$ tube.

4) Reference Flow: The reference flow/pressure is generated using a 425 gram $CO-2$-cartridge with a custom pressure regulator (Fig. 31)

5) Prototype configuration 1: Prototype Configuration 1 (Fig. 32) is the pneumatic configuration which consists of the prototype itself attached to a CO_2 cartridge with saturated CO_2. Both the regulated output flow as well as the reference flow utilizes a $\varnothing 1.0mmx0.65mm$ tube; the reference flow is assumed to have a constant pressure p_{ref} and a negligigble flow velocity.

6) Prototype Configuration 2: Prototype Configuration 2 (Fig. 33) consists of the pneumatic prototype with the CO_2-connection blocked. Both the input flow as well as the ouput flow are connected through $\varnothing 1.0mmx0.50mm$ tube.

and a tube connected to the flow output as well as the bottom input.

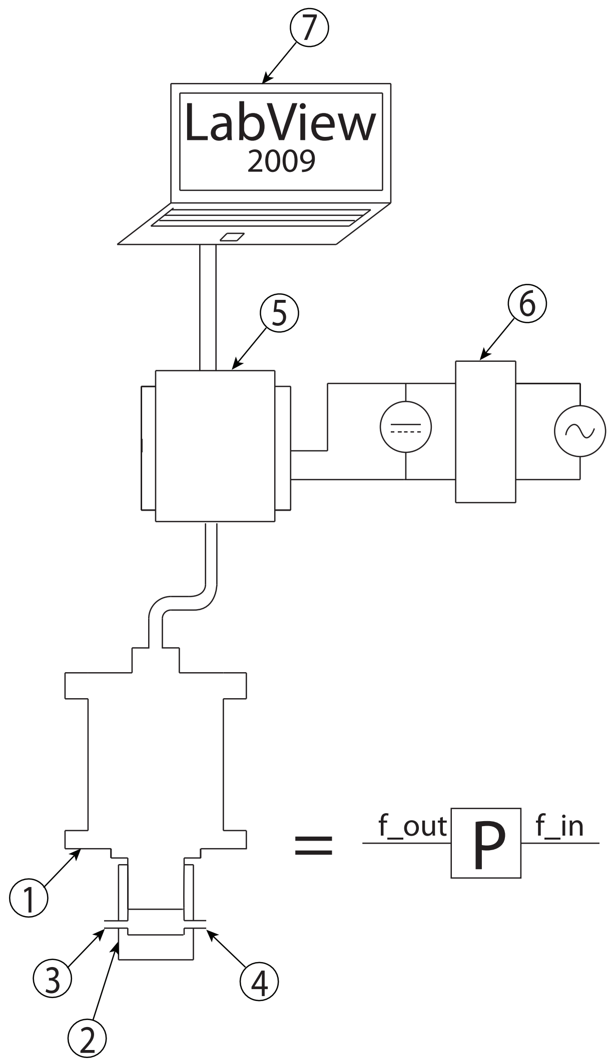

Fig. 28. Description of testequipment used for pressure measurement. Test setup consists of pressure sensor(1) with custom part(2) connecting the output flow,f_{out} (3) and the input flow,f_{in} (4). The sensor is connected to a National Instruments NI USB-6008 AD-Converter (5) and a reference voltage supplied by a Sontronics AD-2400400R 24V AC/DC Adaptor (6). Data acquisition is done using a laptop with LabView 2009 ®(7)

7) Prototype Configuration 3: Prototype Configuration 3, 'Pr3' 34, consists of the spring configuration of the prototype itself with a CO_2 cartridge with saturated CO_2 as high pressure input.The regulated output flow goes through a $\varnothing 1.0mmx0.65mm$ tube.

B. Experimental set-up configurations

1) free flow: Free flow is measured using a high capacity pressure regulator with a 0.5mm hose with 75cm long connected directly to the pressure regulator output and a 5cm 0.25mm capillary. The setup is shown in Fig. 35

2) Operation in spring configuration: The experimental set-up in spring configuration is constructed to test the real life performance of the prototype 36 . The prototype is configured with an input and average output pressure such as in the intended application. Pressure is monitored, flow velocity is

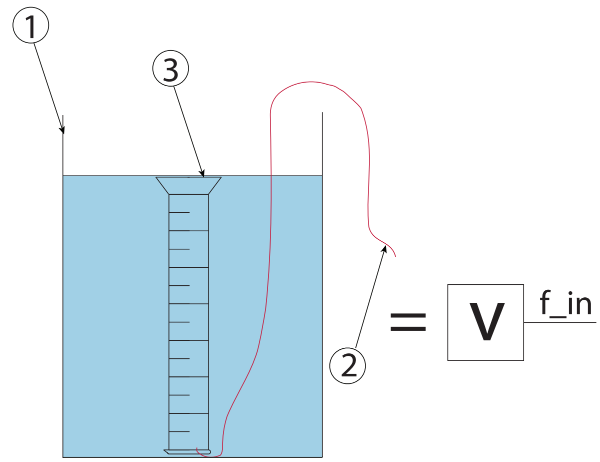

Fig. 29. Description of testequipment used for volume measurement. Test setup consists of container (1) filled with tapwater with a $\varnothing 1.0mmx0.50mm$ tube(2) leading from flow to be measured under Simax 50ml measurement cylinder(3)

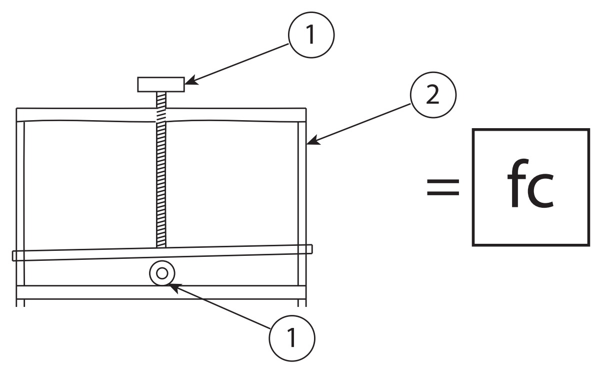

Fig. 30. Flowcontrol through adjustable tube clamp with rotation knob(1), linear guide(2) and tube(3)

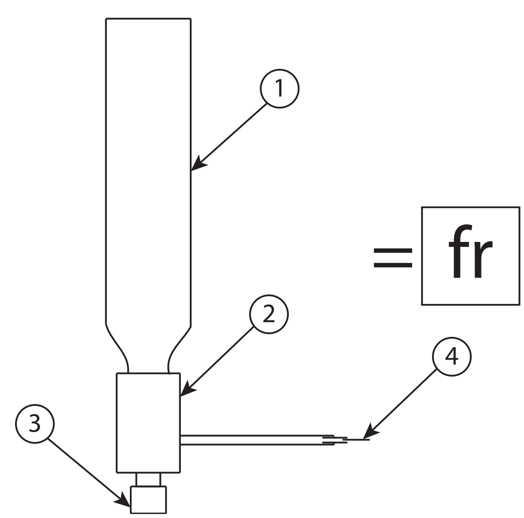

Fig. 31. Reference pressure (fr) supplied by 425 gram CO_2-cartridge(1), regulated with custom pressure regulator(2) which is adjustable through knob(3). Output is through $\varnothing$4.0mm x 2.5 mm tube with an adaptor to $\varnothing$1.0mm x 0.50mm tube (4)

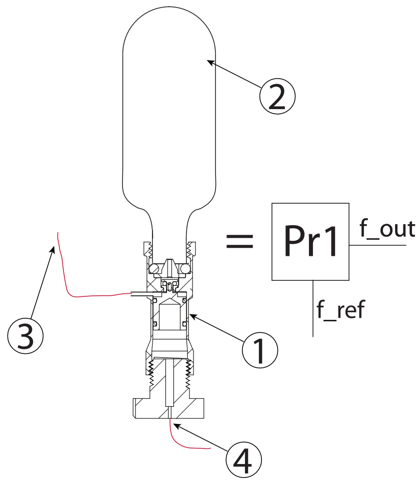

Fig. 32. Prototype Configuration 1, 'Pr1', with prototype body(1), CO_2 bottle(2), output flow, 'f_{out}(3) and reference pressure/flow,'f_{ref}(4). Forf_{ref} it is assumed that the pressure is constant at p_{ref} and flow velocity is negligible

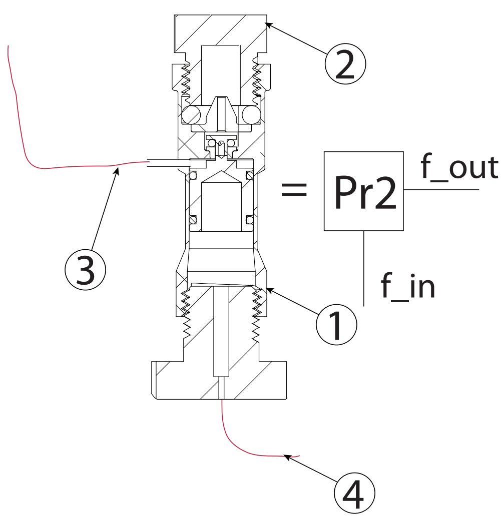

Fig. 33. Prototype Configuration 2, 'Pr2', with prototype body(1), flow block(2), 'f_{out}(3) and f_{in}(4).

varied and measured.

3) Operation in pneumatic configuration: The experimental set-up in pneumatic configuration is constructed to test the real life performance of the prototype (Fig. 37) . The prototype is configured with a CO_2 input at saturation pressure and with a reference pressure supplied by another pressure regulator, which can be blocked off. The flow can be varied and both the output flow and pressure can be measured.

4) Leakage along piston in pneumatic configuration: The experimental set-up in leakage test configuration 38 is constructed to be able to quantify leakage along the piston given a certain pressure differential. The prototype has the CO_2

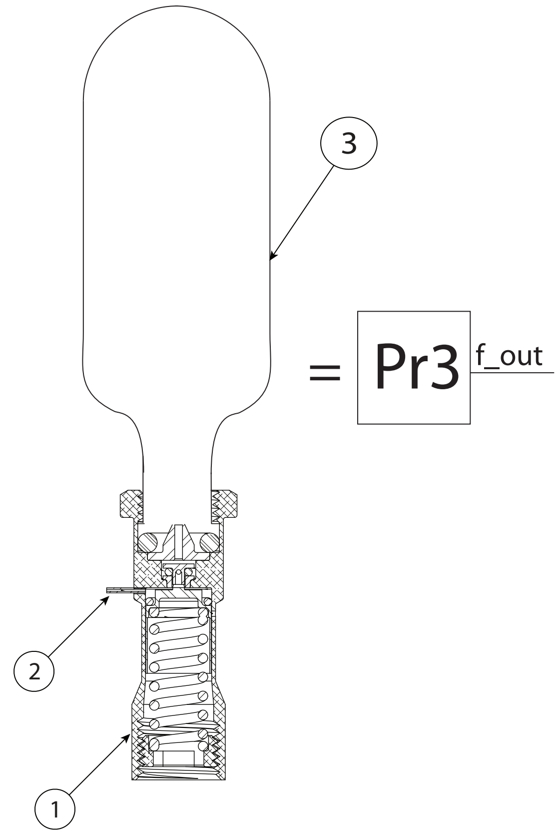

Fig. 34. Prototype Configuration 3, 'Pr3', with prototype body(1), f_{out} (2) and CO_2-cartridge (3)

Fig. 35. Test for free flow velocity

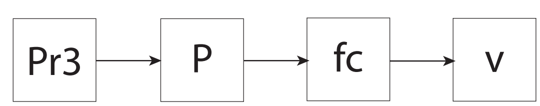

Fig. 36. Experiment with the prototype in spring configuration including attached CO_2 supply (Pr3), pressure measurement (P), flow control (fc) and volume/flow measurement (v).

connection as well as the overpressure exhaust blocked and a pressure is supplied at the bottom of the prototype. The gas flowing past the piston through the output is measured.

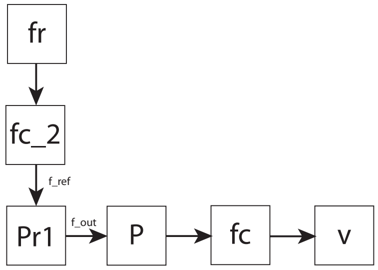

Fig. 37. Experiment with the prototype in pneumatic configuration (Pr1) with reference pressure (fr) which can be blocked through a seond flow control (fc_2). Output flow is controlled (fc) and both pressure (P) and flow velocity (v) are measured.

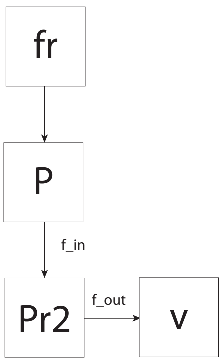

Fig. 38. Experiment with the prototype for testing leakage along the piston with prototype in leakage test configuration (Pr2), with reference pressire (fr), pressure measurement (P) and volume measurement (v)

5) Required force for valve opening: The force required for opening the valve is measured by a stick between the valve and the weighing scale (type Sartorius 1213MP) . An external force will be applied upon the regulator which will increase until the gas starts to flow.

6) Piston friction: The friction force of the piston is estimated using a stick between the piston and the weighing scale 40 . An external force (F1) will be applied upon the regulator which will increase until the piston starts just to move.

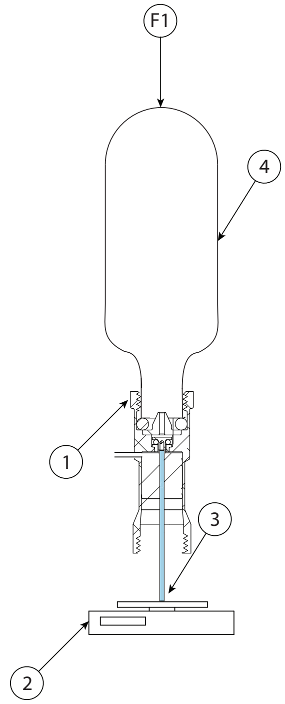

Fig. 39. Experiment determination of the required force for valve actuation. The prototype body(1) with CO_2-cylinder using a stick(3) connected to weighting scale (2); an external force

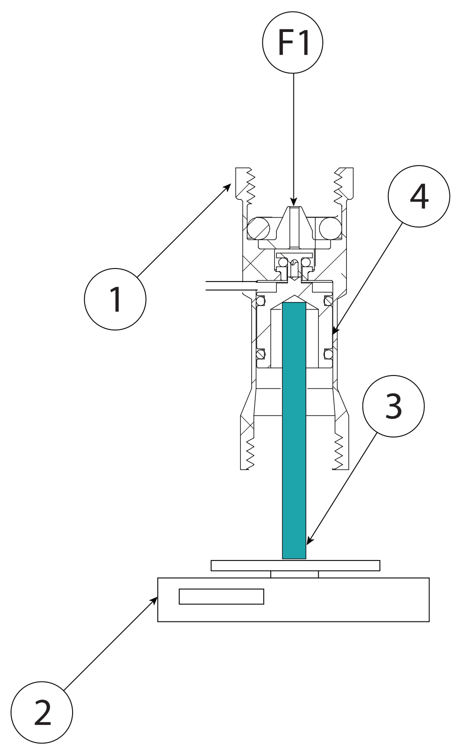

Fig. 40. Experiment determination of the piston friction. The prototype body(1) with piston(4) which is connected using stick (3) to weighing scale (2). An external force (F1) is applied such that the piston just starts to move.

APPENDIX C
CONFIGURATION CHOICE

A. *Spring vs Pneumatic*

A Pneumatic reference force can potentially lead to lighter designs, since dimensions are not restricted by spring sizes and weight, no force concentrations are present as with spring designs. This design is not yet applied with these dimensions on a large scale so implications in long term operation are unknown. Significant research is required upon this working principle before one should apply this in sensitive applications. Another factor which has to be taken into account is the heat sensitivity of a pneumatic reference force. The pressure within the closed compartment will deviate as function of temperature and for very accurate control in situations with stong temperature deviations this is not recommended.

B. *Cartridge vs Integrated*

Integrated gas storage has the potential of delivering a much lighter design. However testing has to be done in order to make sure the constructed pressure vessel is safe. Further a good refill system has to be constructed which does not overfill the regulator. Cartridges are an off the shelf, suboptimal, gas storage solution.

APPENDIX D
MATLAB CODE

An m-file is created in order to estimate the optimal length (W) and radius (R) of a pressure regulator in terms of weight, given a certain allowable pressure deviation (Fig. 41)

10-2-12 14:20 H:\Final Paper\Final Pa...\PerfectSystem.m 1 of 1

```
close all hidden
syms W R Mass real;  %Defines Width and Radius as symbolic
p_ref = 1.2E6   % Reference Pressure [Pa]
p_dev = 0.05    % Maximum allowed pressure Deviation factor [-]
Dx = 0.4E-3     % Movement of piston [m]
Cf = 0.4E-3     % Friction coefficient of O-ring configuration [N/m]
pCO2sat = 6E6   % Saturation pressure CO2 [Pa]
rho = 2.81E3    % density of material [kg/m^3]
sigma = 5E8     % maximum work stress material [N/m^2]
fCO2 = 3        % Safety factor for components subject to CO2 pressure [-]
mvalve = 0.0001 % Mass of valve [kg]
fInt = 3        % Safety factor for components subject to internal pressure [-]
mCO2 = 8E-3     % Desired CO2 storage capacity [kg]
rhoCO2 = 800    % Density of saturated CO2 [kg/m^3]
Fact = 0.8      % Actuation force valve[N]
Rmin = 0.001    % Minimum R considered [m]
Rmax = 0.004    % Maximum R considered [m]
Wmin = 0.005    % Minimum W considered [m]
Wmax = 0.03     % Maximum W considered [m]
Step_DP =0.00001% Step size of the DP plot
mBody = 2*pi*R^2*(R+W)*pCO2sat*(rho/sigma)*fCO2
mPiston = 2*pi*R^2*(R/2+2*R)*p_ref*(rho/sigma)*fInt
mStorage = (3/2)*pCO2sat*((mCO2/rhoCO2)*fCO2+(4/3)*pi*R^3+R^2*pi*W)*(rho/sigma)
dPdiameter = (2*R*pi*Cf+Fact)/(R^2*pi)
dPlength = p_ref*(1-((W-Dx)/W))
dPtot = dPdiameter + dPlength
figure(1)
ezsurf(dPtot, [Rmin,Rmax],[Wmin,Wmax])
xlabel('Radius R [m]')
title('Pressure deviation as function of R and W')
ylabel('length W [m]')
zlabel('Pressure deviation  \Delta P [Pa]')
figure(3)
PLimit = dPtot - p_dev*p_ref
PLimitLine = solve(PLimit,W)
ezplot(PLimitLine,[Wmin,Wmax])
mtot = mBody+mPiston+mStorage
RDum = (Rmin:Step_DP:Rmax)';
WDum = subs(PLimitLine,R,RDum);
MDum = subs(mtot,{R,W},{RDum,WDum});
figure(2)
mtot = mBody+mPiston+mStorage, hold on
ezsurf(mtot,[Rmin,Rmax],[Wmin,Wmax])
xlabel('Radius R [m]')
title('Mass as function of R and W')
ylabel('length W [m]')
zlabel('weight m [kg]')
plot3(RDum(WDum>=Wmin&WDum<=Wmax),WDum(WDum>=Wmin&WDum<=Wmax),MDum↙
(WDum>=Wmin&WDum<=Wmax),'o r')
StringDPMax = strcat ('\Delta P_{max} = ',num2str(p_ref*p_dev,'%1.1e'), '[Pa]')
legend('M = f(R,W)', StringDPMax)
```

Fig. 41. Matlab Code creating the surface plots of mass and pressure deviation as function of pressure regulator length (W) and radius (R).

Appendix E – Pneumatic IGSPR –
and
Appendix F – Drawings Prototype –
are not in the public domain

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