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Design, Develop And Analysis Of Effortless Pressure Regulator Considering Pressure Vessel Aspect

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Abstract
This research paper presents on reducing higher pressure forces towards adjusting knob device, new design with effortless operating safe condition by considering Pressure Vessel calculations towards Pressure Equipment Directive. This modular type design reduces the potential effort during pressure regulating at higher pressure ranges and increases the safety level which offers the most reliable product performance. If the operating load increases, then the regulator flow must increase in order to keep the controlled pressure. Once inlet pressure is introduced, the open poppet allows flow. By adjusting the top knob, the downward pressure on the control devise can be increased, requiring more pressure in the upper chamber to maintain equilibrium. In this way, the outlet pressure of the regulator is controlled. So with every rotation Knob has overcome the friction due to higher pressurized fluid forces on Knob. This experienced that increase in the pressure, effort required to turn the knob also increases. Pressure regulators are meant for frequent pressure adjustments, and then the problem for operator to adjust the pressure as manual efforts are required is really high which goes to not availability of higher pressure range manually operated pressure regulator.

Here project focus on reducing higher pressure forces towards adjusting knob device, new design with effortless operating on safety conditions by considering Pressure Vessel calculations towards Pressure Equipment Directive.

Keywords: *Pneumatic Regulator, effortless operating, pressure vessel, PED*

1. Introduction

Pressure regulators are used in a host of fluid dynamic systems to maintain certain pressures or pressure drops constant in the face of variations in system parameters and/or external disturbances.
Tsai and Cassidy [1] considered the dynamics of a single pneumatic pressure reducer. Stability criteria in terms of reducer design and operating parameters were formulated for a linearized model. John Darlaston [2] describes A general perception is that safety factors are there to provide confidence in the safe use of an engineering component, assembly or system. Pressure equipment by its nature is potentially hazardous and needs factors of safety to provide a margin against failure from uncertainties in design, materials, manufacture, inspection, and subsequently in operation.

Different factors of safety may guard against the same uncertainty. In relation to uncertainty in manufacture, design standards provide for a reduction in the design safety factor with increasing inspection requirements. An example of this is the European Unfired Pressure Vessel standard, EN 13445:2002 [3].

The American, Henry Petrowski made many statements on safety and reliability, two of which are worthy of note in the context of this paper [4]. The first one explains that failure is central to the design process in that more is learnt from failures than from success.
The UK and European legislation requirements on conformity assessment of 'new' equipment provide an example of factors of safety within the control and monitoring of the pressure equipment. In the Pressure Equipment Directive [5] hazard categories are identified. The risk is defined in terms of the stored energy and the process fluid. These terms are used to determine the measures that have to be taken to demonstrate conformity with the essential safety requirements. There is increasing stringency on conformity assessment depending on hazard category. It is worthwhile examining this process as it regularises the approach for non-nuclear components and is not far removed from the approach for nuclear components.
The Directive requires that all pressure equipment and assemblies designed to operate above 0.5 bar and within its scope must be safe when placed on the market and put into service.
The controlling legislation for placing pressure equipment on the market or putting into service within Europe comes from European Union Directives. An important instrument is the Pressure Equipment Directive [5] that is transposed into UK legislation as the Pressure Equipment Regulations [6]. Annex I of the Directive (PED) defines Essential Safety requirements for pressure equipment but not the means for achieving them.
Dhananjay Singh Bisht [7] describes the segment of industrial products, hand held products occupy a major section. An important issue in design of these products is to identify the

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factors that lead to human comfort and those leading to discomfort.
'This Paper came with design of Pressure Regulator with reducing higher pressure forces towards adjusting device, new design with effortless operating on safety conditions by considering Pressure Vessel calculations towards Pressure Equipment Directive.'

2. Methodology

Below is the Pressure Regulator assembly, operated by spring force applied on the piston. Below image shows the different components associated with Regulator assembly which are designed for safe stress level and force distributed among each dynamic situation.

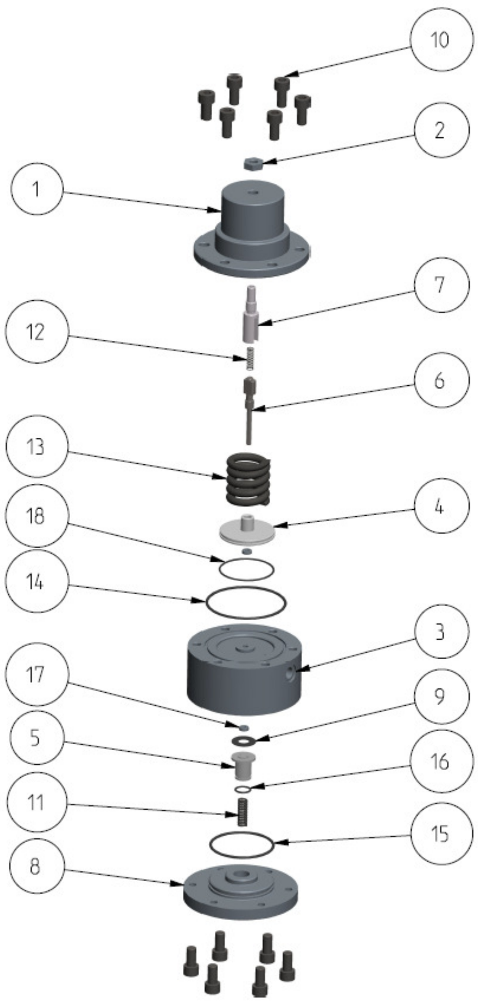


Fig. 1 Exploded view of Pressure Regulator assembly

Pressure regulator is pressure-reducing valves; maintain constant output pressure in compressed-air systems regardless of variations in input pressure or output flow. Fig 1 The Special arrangement of internal Piston, spring, Stem and Bonnet.
Fig. 1 shows the modular design reduces the potential effort during adjustment of pressure regulating at higher pressure ranges and increases the safety level which offers the most reliable product performance.

Here major component which exerts forces on the adjusting devise are Piston spring, Piston with Fluid forces.

TABLE 1
Bill OF Material (BOM)

Balloon No.	*Part Name*	*Qty*
1	Bonnet	1
2	Lock Nut	1
3	Valve Body	1
4	Piston	1
5	Disc Holder	1
6	Stem	1
7	Knob	1
8	End Cap	1
9	Flat Seal	1
10	Screw M10	12
11	Spring –Disc Holder	1
12	Spring-Stem	1
13	Spring-Piston	1
14	O-Ring	1
15	O-Ring	1
16	O-Ring	1
17	U Seal	2

Pressure Regulator responds on dynamic pressure forces, which acts on the above listed components *fig.1* throughout the applicable pressure range 0-20 bar.

Done a systematic pressure balance calculation to responds to pressure variation, designed a functional dimensions of piston, orifice, spring force requirement.

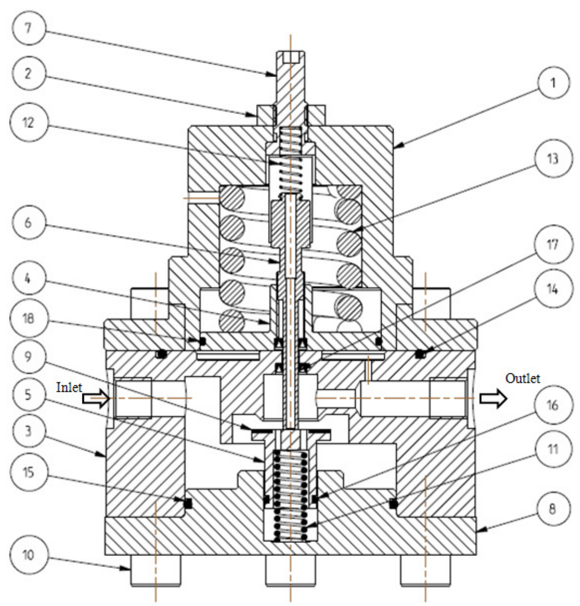
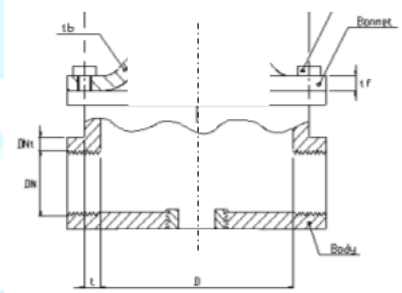


Fig. 2 Cross-section view of Functional flow path

Table 2
Calculation for valve performance against
Pressure equipment directive

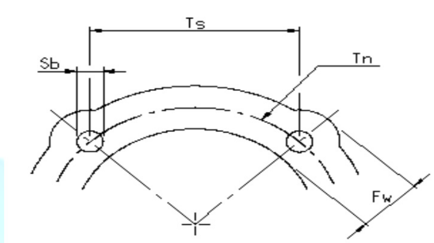
Input			
Parameters	Symbols	Values	Units - IPS
Material Properties			
Aluminum 6026-T9 : Valve Body Components			
Modulus Of Elasticity	E_{al}	70.0	GPa
Tensile Strength	σBb	240	MPa
0.2% Yield Limit	Rpb	160	MPa
Carbon Steel H.R.3000: Fastner			
Tensile Strength	σBb	827.37	MPa
Yield Limit =σBb*0.8	Rpb	661.897	MPa
Stainless Steel AISI 305 : Spring			
Modulus Of Elasticity	Ebr	200	GPa
Tensile Strength	σBb	586	MPa
0.2% Yield Limit	Rpb	207	MPa

Calculations
1. STRESS IN BODY BY INTERNAL PRESSURE
Pipe Size: 1/2" DN15 (20.95 mm Outside Dia.)

Maximum operating pressure differential (MOPD)	PS	1	Bar
Safe working pressure (SWP)	PW	20	Bar
Minimum burst pressure (according to calculation)	P	40	Bar
Safety factor on MOPD	$v = \frac{P}{PS}$	40	Bar
Pressure Regulator Body Specification			
Pipe Size DN15	DN	20.95	mm
Wall thickness body at pipe connection	DNt	5	mm
Inside diameter of body	D	58	mm
Wall thickness of body	t	4	mm
Flange thickness of body or thread depth of screws (smallest value)	tfb	8	mm
Body material Aluminum 6026-T9	Matbody		
			
Inside radius of body	$Ri = \frac{D}{2}$	29	mm
Outside radius of body	$Ru = \frac{D}{2} + t$	34	mm
Values To Be Calculated			
Longitudinal Stress	$\sigma 1 = \frac{P \times Ri^2}{Ru^2 - Ri^2}$	10.68	MPa
Maximum circumferential stress at wall	$\sigma 2 = P[\frac{Ru^2 + Ri^2}{Ru^2 - Ri^2}]$	25.36	MPa
Maximum radial stress at	σ3 = (-P)	-4.00	MPa
Maximum equivalent stress at wall	$\sigma_e = \sqrt{\sigma 1^2 + \sigma 2^2 + \sigma 3^2}$	34.191	MPa
Result: Here σe <Mat_{body} : Maximum equivalent stress at wall are Safe; value is below the maximum allowable stress			

1.1) Calculation of stress in body, caused by force applied on pipe. Forces acting on pipe and body according to EN 161			
Bending moment on 1/2" pipe	M15	105	N.m
Section modulus of minimum cross section of pipe connection	$W_p := \frac{\pi}{32} \cdot \frac{\left[(DN + 2 \cdot DN_t)^4 - DN^4\right]}{(DN + 2 \cdot DN_t)}$	2300	mm^3
Maximum bending stress at pipe connection	$\sigma_b = \frac{M}{Wp}$	45.661	MPa
Result: Maximum bending stress at pipe connection is σ_b < Mat_{body}; value is below the maximum allowable stress			
1.2) Calculation of stress in body, caused by torque applied on pipe. Torque acting on pipe and body pipe according to EN 161			
Torque on 1/2" pipe (DN15	T15	50	N.m
Maximum torque to be applied on pipe connection	T	50	N.m
Section modulus of minimum cross section of pipe connection	$W_t := \frac{\pi}{16} \cdot \frac{\left[(DN + 2 \cdot DN_t)^4 - DN^4\right]}{(DN + 2 \cdot DN_t)}$	4599	mm^3
Maximum shear stress at cross-section at pipe connection	$\tau t = \frac{T}{W_t}$	10.872	MPa
Result: Maximum shear stress at pipe connection is τt <0.67. Mat_{body} ; Value is below the maximum allowable stress.			

2. Calculation of stress in bonnet , caused by internal pressure			
Bonnet Specification			
Bonnet Material	Mat_{bonnet}	AL 6026-T9	
Wall thickness of bonnet at inside diameter of body	t_b	5	mm
Values To Be Calculated			
Bending moment in bonnet at inside diameter of body	$M_{bon} = \frac{P \times D^2}{32}$	420.50	N.mm
Maximum bending stress at bonnet wall	$\sigma_{bon} = \frac{M_{bonnet} \times 6}{tb^2}$	39.422	MPa
2.1) Calculation of stress in flange of bonnet/body, caused by internal pressure			



Flange thickness of bonnet at pitch diameter	tf	12	mm
Flange width	Fw	20	mm
Pitch diameter of bonnet screws	Tn	90	mm
Number of bonnet screws	n	6	
Size of screw holes in bonnet	Sb	10.5	mm
Values To Be Calculated			
Pitch between two screws	$Ts = Tn \times sin\frac{180}{n}$	45	mm
Force per section due to internal pressure	$Ft = \pi \times D^2 \times \frac{P}{4n}$	1761.386	N
Stress in Bonnet/body at screw holes	$\sigma_{bos} := \frac{F_t \cdot T_s}{2 \cdot (F_w - S_b) \cdot t_f^2}$	28.710	MPa
Result: Maximum bending stress at bonnet/body flange is τt < Mat_{bon} ; value is below the maximum allowable stress			

Design has been evaluated through analytical calculation and stress are below the maximum allowable stress at higher working pressure i.e. 20 bar. Direct acting Pressure regulator at higher pressure range, which generally not easily available in market. So considering pressure vessel aspect Valve is Safe.

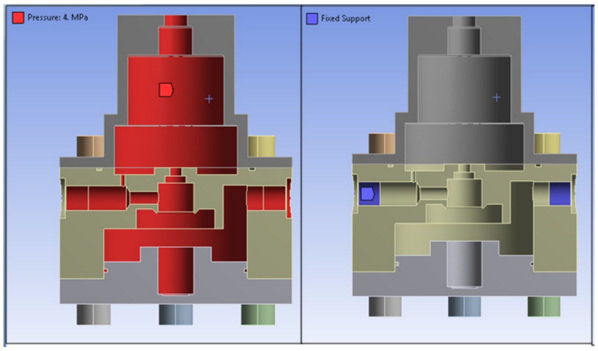


Fig. 4 Boundary conditions Internal Pressure=40 Bar & Constrained Pipe Location in body.

This new Design has been done to perform valve function at higher pressure condition up to 20 bar internal pneumatic pressure.

3. Results and Discussion

3.1 Stress Analysis

3D CAD model is created in Pro-Engineering to define the Functional dimension analysis and Fit function analysis. FEA is performed in ANSYS workbench.

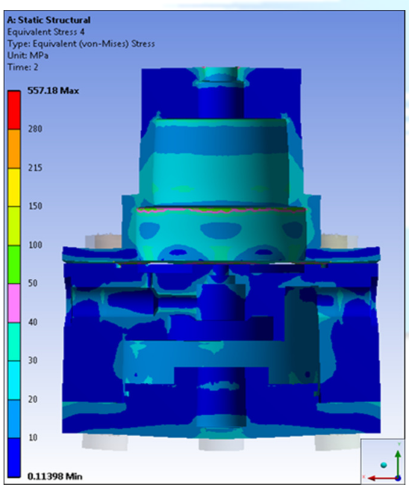


Fig. 3 FEA Simulation stress distribution

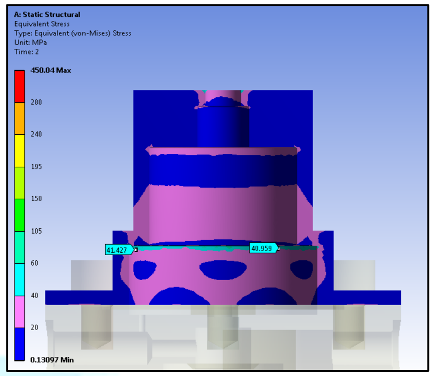


Fig. 5 Component wise stress distribution on the Bonnet at 40 bar internal pressure condition

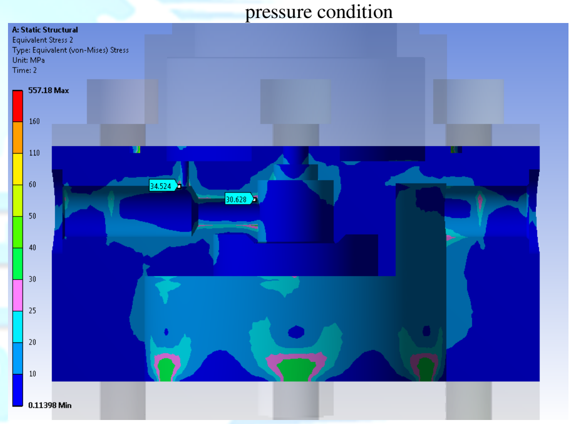


Fig.6 Component wise stress distribution on the body at 40 bar internal pressure condition

Table 3
Comparison table

	Calculated (MPA)	FEA (MPA)
σ_e = Sress in body by internal pressure	34.191	34.628
σbo=Maximum bending stress at bonnet wall	45.661	39.422

3.2 Force Analysis to evaluate Torque requirement

A pressure regulator Fig 7 has a sensing element piston which, on one side, is subjected to a load force (Fs5) created by a spring (as shown below force diagram) or can be a gas pilot pressure. On the other side, the sensing element is subject to the force (Fp4) of the system fluid.

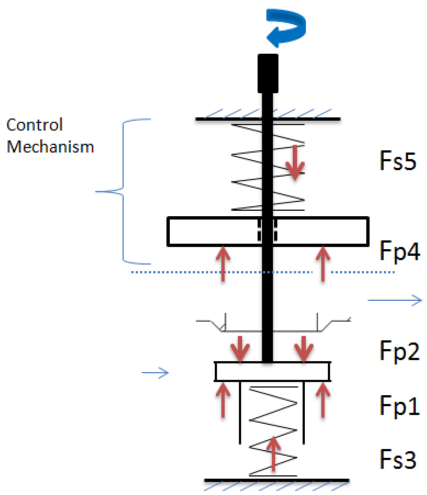


Fig.7 Force Balance Analysis

The function of a pressure-reducing regulator is to reduce a pressure and to keep this pressure as constant as possible while the inlet pressure and the flow may vary. This is accomplished by the fluid force (Fp4) being equal to or slightly lower than load force (Fs5) causing the poppet to open by overcoming Fs3.

Table 4
Force evaluation under dynamic condition

	Pressure (Bar)	Piston Diameter (mm)	Effective Piston Ares (mm2)	Force (N)
$P_{Piston}\ (Fp4)$	1	58	2622.4	262.2
$P_{Piston}\ (Fp4)$	2	58	2622.4	524.5
$P_{Piston}\ (Fp4)$	4	58	2622.4	1049
$P_{Piston}\ (Fp4)$	10	58	2622.4	2622.4
$F_{Spring}(Fs5)$	Piston Spring Stiffness = 264 N/mm			

Spring Designed with stiffness= 264 N/mm with respect to pressure increment from 1 to 20 bar to have dynamic balance. With this new design forces acting on the knob is zero. That result in the constant force applied on the knob by manual finger force.

4 Test Set Up

Test set up has been made to evaluate the Torque required towards knob to adjust the outlet pressure. Fig 8

Operating Pressure range 0 to 20 bar

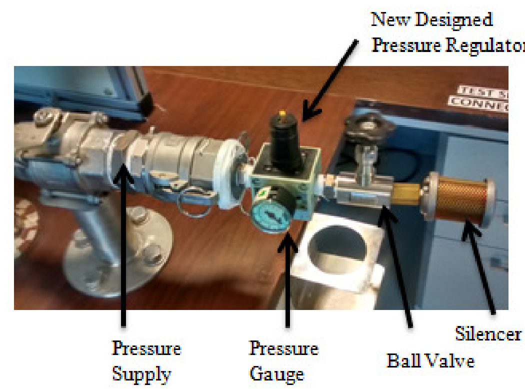


Fig.8 Experimental Test Setup

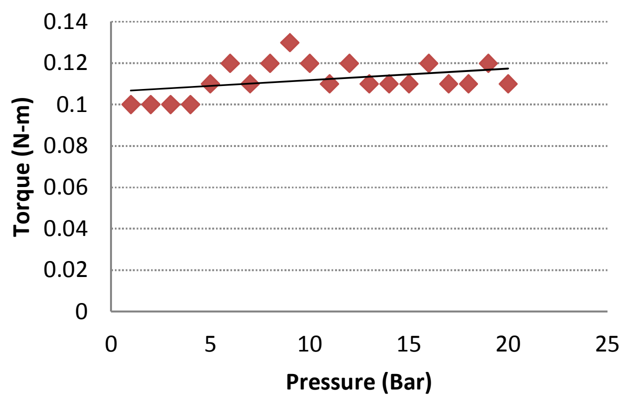


Fig.9 Torque vs. Pressure Test Observation by digital torque range

5. Conclusions

A systematic approach of failure mode of different components has been studied at higher operating condition. Good correlation in the theoretical calculations and simulation results is achieved for determining valve Functional dimensions and comply the safety norms.

A Design with reducing higher pressure forces towards adjusting knob device, new design with effortless operating on safety conditions by systematic study of dynamic Force inside the Pressure regulator component.

Design delivers a manual effort required to adjust knob will be constant throughout the operating range of pressure regulator and which can be controlled and easily adjusted by manual finger force.

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