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Novel multi-way microvalve with ease of fabrication and integration for microfluidics application

Shao Qiang Tang^a, King Ho Holden Li^{a*}, Sheung Jien Lee^a, Jiamin Jasmine Zeng^b, Sum Huan Ng^{b*}

^aSchool of Mechanical and Aerospace Engineering, Nanyang Technological University, 50 Nanyang Avenue, Singapore 639798

^bA*STAR's Singapore Institute of Manufacturing Technology (SIMTech), 2 Fusionopolis Way, #08-04, Innovis, Singapore 138634

*Corresponding: HoldenLi@ntu.edu.sg (King Ho Holden Li), shng@simtech.a-star.edu.sg (Sum Huan Ng)

Abstract: We present a miniaturized active multi-way microvalve, with a rotor, measuring only 3.7 mm in diameter and having a footprint of only about 14 mm², and that is designed for integration into microfluidics chips, allowing for overall size reduction for microfluidics platforms. It has a zero dead volume, and an overall internal volume of only 268 nL. Its operation is based on a rotary mechanism, with compressible rotor and complementary stator seat valve specially designed to have five-degree rotational symmetry, and in which the rotor is made of Viton elastomer to provide a seal and switch between outlets precisely. The microvalve, which can be manually, as well as automatically actuated, is able to withstand pressures up to 370 kPa with no leakage and has a switching torque of under 0.03 Nm.

Keywords: microvalve, multi-way valve, zero dead-volume, small sweep-volume

1. Introduction

Fully integrated microfluidic devices such as Micro Total Analysis Systems (μ TAS) or Lab-on-Chip (LOC) devices [1] is in demand, due to its advantages of faster on-site analysis for applications such as clinical diagnostics [2,3] and chemical [4] and cellular analysis [5,6]. The popularity of such an integrated solution demands advancement in microfabrication techniques, as well as innovative designs of microfluidic device components, enabling smaller footprints and effective integrations. One such component is the microvalve, which are categorized into the active and passive ones. Active microvalves can be switched on and off, by some manual or automatic actuation methods, such as pneumatic [17, 37-40], magnetic [9, 37-40], electrostatic [54, 55], piezoelectric [52, 53], or phase changes [42-47]. Passive microvalves are used as check valves, enabling one-way flow in a fluid channel. The design of microvalves faces challenges of reducing the sizes and leakage flow, lowering fabrication complexity and cost, while ensuring ease of use and operational robustness against wear and tear and leakage. The use of valves ranges from the simple need to control flow rate and direction, to switching between one or more inlets to one or more outlets. The switching speed is also important in automated applications. Microvalves are categorized into mechanical and non-mechanical types. The mechanical type requires more energy to move bulkier structures against the sealing membrane during valve switching, while the non-mechanical type uses diaphragm made of 'smart' materials that undergo changes in its physical or chemical properties under some low-energy stimulation from the environment, which result in the modification of the fluid flow field [7]. One such mechanical microvalve, which uses the solenoid for actuation, such as the first active microvalve developed by Terry et al [9], or the electromagnetically driven ball-in-seat valve designs by Krusemark et al [34], Oh et al [35], and Fu et al [36], have the disadvantage of having a bulkier and structurally complicated design, making it a challenge to fabricate and integrate into LOC devices. Among the non-mechanical microvalves, that use phase-change materials, are those that use paraffin [42, 43], hydrogels [44, 45] and ferrofluids plugs [46, 47]. Although microvalves based on phase change materials have a much simpler construction and occupy a smaller footprint, they generally have slower switching speeds, ranging from seconds, for the paraffin-based ones [42, 43], to tens of seconds, for the ferrofluid-based ones [46, 47]. In this respect, the mechanical microvalves perform better, with switching times on the order of tens of seconds, for the ball-in-seat valve types [34-36], to hundreds of milliseconds, for the rotary type, such as those by Hasegawa et al [19] and Luharuka and Hesketh et al [14]. While being able to achieve much faster switching speeds than the phase-

change materials versions, the above mechanical active microvalves can also maintain a sufficiently high pressure tolerance, of above 50 kPa before leakage sets in. Nominal pressure tolerances of above 3 kPa are required for microfluidics applications [7]. Piezoelectric microvalves are usually more suitable for channeling gas, or as micropumps, since piezoelectric crystals have low strain and are able to generate large forces of above 100 N. Electrostatic-actuated microvalves are also not suitable for liquid channels, as the high voltages used may cause the electrolysis of the fluids [54, 55].

Another type of active microvalves are the ones driven by external systems, such as the pneumatically actuated, diaphragm-based ones by Unger et al. [17], or the electromagnetically actuated ones by Cho and Ahn et al [37-40], which usually incorporate dead volumes. Terry's solenoid plunger for moving a membrane over a valve also generates a dead volume of around 4 nL [9]. Such designs may be more suited for flow regulation instead, such as those by Meckes [41] and Cho and Ahn et al [39]. The dead volumes, that are incorporated in the microvalve design, are dependent on the stiffness of the membrane material. Latex membranes [48, 49] used in some pneumatic microvalves were reported to have dead volumes of 50 nL, while the PDMS (Polydimethylsiloxane)-based ones have dead volumes of less than 8 nL [50, 51].

This article introduces a new variety of multi-way microvalve that can perform precise multi-way switching, on a small footprint of the microfluidic environment, using only a single actuating structure. Most microvalve designs, such as the electromagnetically actuated one by Terry et al [10,11], and the bistable diaphragm-based microvalve by Goll et al. [12] and Capanu et al. [15], have bulky constructions, that led to high complexity and cost of integration for realizing a multi-way application. The device presented here uses the rotary mechanism for aligning the inlet on the rotor with the desired outlet on the stator, which has been similarly used by Hasegawa et al [19], Godino [20], Gransee [22], Schaeffer and Stein [21]. Schaeffer and Stein's design incorporates a simple elastic element to provide sealing pressure between the inlet and outlets, in comparison to a pressurized spring in Godino's design. However, Schaeffer and Stein did not implement any construction that allows for precise positioning of the inlet relative to the outlets, while the use of alignment pins in Godino's design and notches and steel balls in Hasegawa's design, may pose a challenge to integration into a microvalve. The design in this paper consists of a rotor that is made of elastic material and of a pentagonal shape, allowing a biased fit to the similarly shaped valve seat on the stator, such that a precise positioning is achieved. The omission of the alignment pins reduces the complexity of fabrication, while the design still allows precise positioning. Although some of the past pneumatic and phase change-based microvalves designs have footprints as small as below 10 mm², to achieve a five-way switching, as in our design, would require stacking 5 of such microvalves laterally or vertically, resulting either, in an overall larger footprint than that of our design, or in an increase in the complexity and number of steps of fabrication. Table 1 provides a comparison of the collection of the above mentioned works.

Table 1.List of past microvalves and the new microvalve described in this paper (first item)

Device/ Material type	Group	Year	Switching Mechanism	Function	Dead Volum e (nL)	Swept Volum e (nL)	Total Internal Vol (nL)	Multi- way	Mode	Guiding structures	Sealing structures	Precise Switching	Switching time (s)	Switching power	Leakage Pressure (for fluids) (kPa)	Material	Device dimensions (footprint) (mm ²)
Pentagonal-shaped rotor-fitted valve seat	Ng, Li et al	2016	Manual or Servo-actuated	Multi-way Switch	0	268	268	Yes	Bi	Matching geometries of rotor and stator	Hyperelastic rotor lower ribs around channels and perimeter	Yes			373	Viton (Rotor lower), PMMA (others)	14
Rotary	Schaeffer and Stein, Gransee [21]	2011		Multi-way Switch				Yes	Bi	Clamping element	Elastic sealing ring	No				EPM, SBR, Viton, PEEK, POM, ABS	
Rotary	Godino et al [20]	2010	Manual	Multi-way Switch	0	6.40E+05	6.40E+05	Yes	Bi	Brass pins	Elastic rotor and stator	No				PDMS (stator, rotor), PMMA (case)	65
Rotary	Hasegawa et al [19]	2008	Solenoid	Multi-way Switch	0	338	338	Yes	Bi	steel balls and notches	Silicon rubber rings, Polymer springs	Yes	0.1	12 W	500	UV polymer	156
Rotary	Luharuka and Hesketh [14]	2008	Variable reluctance stepper motor	Switching	32.89	0.03	32.92	Yes	Bi	Flexible links	NiFe gate mass (with seat gap of 3.6 to 6 μ m)	Yes	0.01		41	PDMS, NiFepermalloy	25
Pneumatic	Kazuo Ryutaro [18]	2000	Pneumatic	On-off Switch	0	0.125	0.125	Yes	NC							PDMS channels and membrane	400
Membrane valve	Terry et al [9]	1979	Solenoid plunger	On-off Switch	4	10	14	No	NC	NA	Ni diaphragm					Pyrex membrane, silicon channels	

Planar Solenoid on membrane	Meckes et al [41]	1999	Electromagnetic	Flow regulation	0	14	14	No	NO			No		25 mA			48
NiFe on membrane	Choi, Ahn et al [37-39]	2001	Electromagnetic	Flow regulation				No	NC			No				Silicon membrane	15
Ball in valve seat	Oh et al [35]	2001	Electromagnetic	Flow regulation		3.13E+06	3.13E+06	No	NC			No		500 mA	2.1	Ni (ball), Teflon (housing)	5.95
Ball in valve seat	Fu et al [36]	2003	Solenoid	Flow regulation	9.00E+03	3.15E+03	1.22E+04	No	NO			No	0.01	300 mA	50	Fe (ball)	100
PDMS membrane	Grover et al [51]	2003	Pneumatic	PCR	8	33	41	No	NC			No			75	PDMS	2.88
Latex membrane	Legally et al [48]	2000	Pneumatic	PCR	50	156	206	No	NO/NC			No				Latex	6.25
PDMS membrane	Hosokawa and Maeda [56]	2000	Pneumatic	Multi-way Switch	18.90	0.38	19.28	Yes	NC			No			10	PDMS	400
Pneumatic (Inline)	Quake et al [17]	2000	Pneumatic	On-off Switch	0	0.1	0.1	No	NO			No	0.01			PDMS	0.01
Polymer	Shao et al [52]	2004	Piezoelectric	On-off Switch	6	1.5	7.5	No	NC			No	0.0007	300V	50 (N2)	Silicone	0.03
Hydrogel	Liu et al [44]	2002	Phase Change	On-off Switch	45	12	57	No	NO			No	12		390		0.36
Hydrogel	Richter et al [57]	2004	Phase Change	On-off Switch			50	No	NC			No	2.3	200mW	840	PNIPAAm	0.25
Hydrogel	Baldi et al [45]	2003	Phase Change	On-off Switch	2199	98	2297	No	NO			No	300 - 2220			PDMS spacer, glass channels, silicon midlayer	15
Paraffin	Pal et al [42]	2004	Phase Change	PCR	18	12	30	No	B			No	1	7.5mW	1725	Glass channels	4.8
Paraffin	Liu et al [43]	2004	Phase Change	On-off Switch	3000	3000	6000	No	B			No	20	100mW	275	PDMS membrane, PC channels	900

Ferrofluid	Hartshorne et al [46]	1998, 2004	Phase Change	On-off Switch	576	24	600	No	NO			No	150		12	Glass channels	150
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Abbreviations: a) Function: PCR (Polymerase Chain Reaction), b) Mode: NC (Normally closed), NO (Normally opened), Bi (Bistable), c) Material: PMMA (poly(methyl methacrylate)), PDMS (Polydimethylsiloxane), EPM (Ethylene-propylene rubber), SBR (Styrene-butadiene rubber), PEEK (Polyether ketones), POM (Polyoxymethylene), ABS (Acrylonitrile-butadiene-styrene), PNIPAAm (poly(N-isopropylacrylamide)), PC (Polycarbonate)

2. Device design and fabrication

The microvalve, presented here, consists of 4 simple components (Figure 1a), the stator upper, stator lower, rotor upper and rotor lower. The lower rotor is of an elastomeric material-Fluoroelastomer (Viton), with a nominal Shore A Hardness of 75 ± 5 , and high chemical resistance against a wide variety of organic solvents, chemicals and oils [22, 24]. The other components are fabricated with a thermoplastic, polymethyl methacrylate (PMMA). The stator upper and lower are bonded together, with the rotor sitting within and free to rotate independent of the stator. The rotor lower contains a channel that connects the inlet, located in the centre of the stator lower, to one of the five outlets located at the periphery of the stator lower. The stator lower has a pentagonal-shaped valve seat that provides a complementary fit for the rotor lower, which is also similarly shaped (Figures 1). Such a construction allows the entire volume in the channel to be swept during switching, resulting in a zero dead volume microvalve. Whereas several other microvalve designs make use of silicon rubber gaskets [12, 19] or PDMS valve seats [17,18] to provide leak-proof seals around the inlet and outlet, the viscoelastic Viton-based rotor lower in the current design provides dynamic sealing between itself and the stator lower, and around the inlets and outlets, preventing leakage. The rotor is only 3.7 mm in diameter, allowing easier integration into Lab-on-chip devices. Its height is marginally greater than the depth of the valve seat in the stator so as to result in compression of the rotor after the bonding of the two parts of the stator (Figure 1b and figure 2b). Figure 2 shows detailed mechanical drawings with dimensions of critical features of the microvalve components which contribute to the sealing and guiding functions. The valve seat on the stator lower has a depth of 1.5mm (Figure 2c), while the rotor lower has a height of 1.6mm (Figure 2b). The compressive force imposed by the rotor against the stator lower, provides a seal. Protruding ribs, of about 0.1mm in width (Figure 2b), around the channel in the rotor lower, the outlets in the stator lower and along the perimeter of the rotor lower provide additional protection against leakage. As the rotor lower rotates relative to the valve seat, the corners of the pentagonal structure deform temporarily, returning to their original shape when the rotor lower and valve seat are aligned again after every execution of 72-degree angular displacement. As the positions of the outlets in the stator lower coincide with the corners of its pentagonal shape, the elastic behavior of the rotor lower provides a guiding force to precisely align the inlet, channel and one of the five outlets with each 72-degree displacement. The rotational displacement of the rotor assembly is achieved by applying a torque on the recessed slot on the rotor upper. An opening in the stator upper allows access to the recessed slot on the rotor upper. The rotor can be manually rotated via the use of a screw-driver, or automatically actuated by a stepper motor.

The rotor lower is fabricated by injection-molding of Viton, while the rotor upper is fabricated via machining of pre-cast PMMA sheets at SIMTech's facility (Figure 3a). The boss on the rotor upper is mated with the complementing recess on the rotor lower with a minuscule amount of Loctite 401 cyanoacrylate-based instant adhesive, and held firmly and promptly while quick curing occurs. The stator lower consists of two layers of layers of PMMA, which are bonded thermally in an oven at a stable temperature of 120°C, after an initial ramping period of 0.5 hr. After the bonding process has been completed, the components are cooled down over 1 hr. Before placement in the oven, the PMMA layers are degassed in a degassing chamber overnight, cleaned with isopropanol to remove dust from the surface, aligned with the help of a microscope and finally restrained with a clamping jig under 0.9 Nm delivered by a torque wrench. The stator upper, stator lower and pre-assembled rotor are assembled together with an M2 screw and nut (Figure 3c).

a)

b)

Fig. 1. a) Isometric exploded view of the microvalve components; b) Side view of the assembled microvalve; c) Top view of assembled microvalve; d) Detailed side view of assembled microvalve, showing location and relation of fluid passages

Fig. 2. Detailed mechanical drawings of the microvalve components: a) rotor upper; b) rotor lower; c) stator lower.

Fig. 3. a) Components of microvalve test setup; b) Exploded view of all components of test setup, showing the method of assembly; c) Assembled microvalve test setup with 5 different output channels for one inlet

3. Device Evaluation

3.1 Simulation procedure

A detailed investigation on the optimum applied force to apply during actuation of the microvalve rotor while minimizing leakage and stress, strain on the surrounding structures, is done with a simulation using ANSYS, using a CAD model imported directly from Solidworks.

Middle range (10-250%) [28] deformation of the Viton-based rotor is expected to be encountered during the actuation of the microvalve. Mooney-Rivlin model [26,27] is used for determining the strain energy of the incompressible Viton-based rotor, where the strain energy is defined as

$$W = C_1(I_1 - 3) + C_2(I_2 - 3) \quad (1)$$

Where W is the strain energy,

I_1 and I_2 are the two strain invariants of the left Cauchy-Green deformation tensor,

C_1 and C_2 are the material constants.

Fig. 4 Simulation of pressing force onto rotor assembly

The above material constants ($C_1=5.805E+05$ Pa, $C_2=3.614E+05$ Pa) are obtained by a curve-fitting process performed by ANSYS, on the stress-strain curve, obtained via uniaxial compression testing on a sample of Viton material. An Instron 5548 micro tester is used for this purpose, and the test is performed five times to get stable results. The incompressibility parameter D is set to 0, to enforce the incompressible condition of the Viton material. The material response of PMMA [23, 29] for the other three components of the microvalve is modeled, using the preset properties for polyethylene. To simulate the compressive forces that are generated after the stator upper and lower are bonded over the rotor, a normal force, in the negative x-direction (Figure 4), is applied on the top surface of the rotor upper. The force is varied in steps, from 0.5 N up to a maximum of 12 N.

The preset material properties for polyethylene (Table 2) were used to model the behavior of the PMMA parts in the microvalve. The physical properties of PMMA are in the range of that of polyethylene, with PMMA typically having a higher tensile strength and Young's Modulus [23][29]. Also, the deformation of the PMMA material relative to that of the Viton material is insignificant in the microvalve, thus the behavior of the PMMA parts under loading is less important compared that of the Viton part in the microvalve assembly.

Since PMMA has a much higher tensile strength and Young's modulus than Viton, the stator lower would experience insignificant deformation, relative to that experienced by the rotor lower. The deformation of the stator lower is therefore set to zero, while the bottom surface set as a fixed support. Resulting values, such as total deformation, equivalent stress, equivalent elastic strain, gap and contact status, from the simulation, are taken and analyzed. The friction coefficient for the surface of contact between the rotor and stator is set to 0.2, allowing the rotation of the rotor against the stator, under the applied normal force.

3.2 Experimental procedure of leakage tests

The final assembled device, with the microvalve integrated into the microfluidic platform (Figure 5, top), was tested for its leakage pressure, under forward and backward fluid flow, across a range of pressures (from 0.5 N to 7 N), imposed by a pressure gauge normal to the top surface of the rotor upper, and against the stator lower. First, the screw and nut in the testing chip holding the rotor in place are loosened, so that no force is applied onto the rotor. The microvalve testing chip is placed in the microfluidic test jig, and the rotor is switched to one of the five outlets. A tubing-to-chip clamping device is clamped onto the inlet of the testing chip and a sealing clamping device is clamped onto the selected outlet of the testing chip. The microfluidic test jig is securely attached to the manual lab jack with hex screws. The syringe pump is manually turned to fill it with distilled water. The tubing is connected from the tubing-to-chip clamping device to the pressure sensor and from the pressure sensor to the syringe pump. The lab jack is placed under the force gauge such that the force gauge presses directly onto the top of the microvalve. The lab jack is then adjusted so that the force gauge applies a force of 0.5N onto the microvalve. The syringe pump is manually turned to gradually increase the hydrostatic pressure to the microvalve. The pressure is increased in steps of 10kPa until the pressure reading no longer holds a steady value and begins to fall. The highest pressure value held steady is recorded. The force applied onto the microvalve is increased by 0.5N and the maximum steady hydrostatic pressure obtained before leakage occurs is taken again, as above. This is done up to a maximum applied force of 7N. The same experiment is performed twice, once with the water flowing in the forward direction, from

the inlet to the outlet channels, and another with the water flowing in the backward direction, from the outlet channels to the inlet.

3.3 *Experimental procedure for switching torque tests*

The switching torque for actuating the microvalve is also investigated, with the following procedures: first, the screw and nut in the testing chip holding the rotor in place are loosened, so that no force is applied onto the rotor. Then the testing chip is placed in the microfluidic test jig, after which the microfluidic test jig is securely attached to the manual lab jack with hex screws. The torque wrench is fitted with a hex socket and a precision screwdriver is attached. The torque wrench (Figure 4, bottom) is set to display the maximum torque applied. The lab jack is placed under the force gauge such that the force gauge presses directly onto the top of the microvalve. The lab jack is then adjusted so that the force gauge applies a force of 0.5N onto the microvalve. The torque wrench is used to switch the valve, and the maximum torque reading is recorded. The force applied onto the microvalve is increased by 0.5N and then the valve is switched again using the torque wrench. This is done repeatedly up to a maximum force applied of 7N.

Table 2. Preset material properties for polyethylene, being used for PMMA in the static structural simulation

Property	Value	Unit
Density	950	kgm ⁻³
Coefficient of Thermal Expansion	0.00023	C ⁻¹
Zero-Thermal-Strain Reference Temperature	22	C
Young's Modulus	1.10E+09	Pa
Poisson's Ratio	0.42	
Bulk Modulus	2.29E+09	Pa
Shear Modulus	3.87E+08	Pa
Tensile Yield Strength	2.50E+07	Pa
Compressive Yield Strength	0	Pa
Tensile Ultimate Strength	3.30E+07	Pa
Compressive Ultimate Strength	0	Pa

Fig. 5. Microfluidics jig setup for hydrostatic leakage test (top). Torque wrench used for the switching torque test (bottom).

4. Results and discussion

4.1 *Simulation results and discussion*

The microvalve is investigated for its structural stability and effectiveness in leakage prevention, during operation. To obtain the failure criterion for the rotor, we use the von Mises stress [28],

$$\sigma_v = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}} \quad (2)$$

becomes $\sigma_v = \sigma_I$, in the uniaxial tension test that is conducted on the Viton-based rotor, since the other two stress components go to zero. S_y is the yield strength of Viton. Hence, we get $\sigma_v = \sigma_I = S_y$. However, since there is no clear distinction on the stress-strain curve of Viton where plastic deformation starts to occur, the yield point is taken to be the maximum load point [31], and therefore, in the current analysis, the tensile strength of Viton, S_U is used to define the failure criterion. Given the following form of factor of safety (FS), which is given by the stated rationales and general rules in engineering design textbooks [32],

$$FS = FS_{material} \cdot FS_{Stress} \cdot FS_{geometry} \cdot FS_{failure\ analysis} \cdot FS_{reliability} \quad (3)$$

Where

$FS_{material} = 1.1$, (as the material yield strength is obtained from manufacturer testing.)

$FS_{Stress} = 1.3$, (as there is potential overload of stresses during switching operation of the valve when the rotor lower slides relative to the stator, which is not simulated in this model.)

$FS_{geometry} = 1.0$, (as the manufacturing tolerances are tight and have been closely held.)

$FS_{failure\ analysis} = 1.4$, (as the failure theory used in this analysis is simplified. Also, as the actual yield strength of the material may be lower than the tensile strength used in the failure condition, a higher safety factor is used here.)

$FS_{reliability} = 1.1$, (as the required reliability of the part need not be very high due to it being designed for single-use or disposable devices.)

a factor of safety of 2.2 is obtained. Considering that the tensile strength of Viton is greater than 6.5 MPa and the relationship - $FS = S_U / \sigma_v$ - the von Mises Stress allowed would be 2.95 MPa. From Figure 6 (top graph), a pressing force of 5 N results in σ_v of 2.93 MPa, so a pressing force of 5N is appropriate for this microvalve. Considering an FS of 2.2, a design value of maximum equivalent elastic strain, ϵ_{eq} , of 0.68 is obtained. As the ultimate strain E_f for Viton is typically observed to be at 1.7, the use of FS of 2.2 is justified. Moreover, the static structural simulation data shows that with a normal force on the top of rotor of 7.5 N, a ϵ_{eq} of 0.675 is obtained (Figure 6, bottom graph). This implies that the material is stable under an elastic strain generated by up a vertical pressing force of up to 7.5 N.

Fig. 6. Simulated equivalent stress maximum with respect to applied normal force on rotor upper (Top); Simulated equivalent elastic strain maximum with respect to applied normal force on rotor upper (Bottom)

In the simulation studies, the contact forces on the ribs, which are important for improving leakage prevention, are also studied. With a higher compressive forces applied on the assembly, the contact pressures on the ribs on the perimeter of the rotor lower and around the channels and inlets, are reduced, due to the increase in contact area between the surface of the rotor lower and the stator (Figures 7a and 7b). This is undesirable as it may compromise the leakage prevention in the microvalve. The contact pressure on the ribs around the inlets and channels is about 3.28 MPa and about 1.11 MPa on the ribs around the rotor perimeter, with a 6 N compressive force imposed on the assembly (Figure 8a), while the contact pressure around the inlets and channels reduces to 2.64 MPa, and that around the perimeter goes to zero, under a compressive force of 12 N (Figure 8b). Hence, to optimize leakage pressures, the compressive force should not exceed 6 N.

Fig. 7a) Simulated gap at bottom surface of rotor lower at 6N pressing force; **b)** Simulated gap at bottom surface of rotor lower at 12N pressing force

Fig. 8.a) Simulated contact pressure at bottom surface of rotor lower at 6N pressing force; b) Simulated contact pressure at bottom surface of rotor lower at 12N pressing force

Convergence is checked and affirmed for total deformation, equivalent stress and equivalent elastic strain, with respect to changes to mesh size (Figures 9). The criteria for convergence is set as being equal or less than 0.1% of change in the quantity considered with increases to the mesh size. From the following results of changes of the order of negative 2, the mesh of 44312 elements is considered reasonable for achieving good convergence for the above quantities. In addition, reasonable convergence for force and displacement is achieved up to 19 iterations of the numerical computation via the Newton-Raphson method (Figures 10).

	Total Deformation (m)	Change (%)	Nodes	Elements
1	2.0705e-005		15338	44312
2	2.0685e-005	-9.6994e-002	14433	60704
3	2.0702e-005	8.1318e-002	31050	144397

	Equivalent Stress (Pa)	Change (%)	Nodes	Elements
1	3.0455e+005		15338	44312
2	3.0436e+005	-6.0134e-002	14433	60704
3	3.0445e+005	2.7305e-002	31050	144397

	Equivalent Elastic Strain (m/m)	Change (%)	Nodes	Elements
1	5.6212e-002		15338	44312
2	5.6269e-002	0.10221	14433	60704
3	5.6228e-002	-7.2719e-002	31050	144397

Fig. 9. Convergence of a) total deformation, b) equivalent stress and c) equivalent elastic strain, with respect to changing mesh sizes.

Fig. 10.Convergence of a) force, b) displacement, with respect to cumulative iteration (x-axis)

Figure 11 shows the vectors of deformation with respect to the 5 corners of the pentagonal edges of the rotor lower. The deformation used is the maximum compression of the rotor material at the pentagonal corners as the rotor lower is rotated in the valve seat of the stator lower. The greatest deformation in the directions normal to the face of the chamfered corners is calculated by hand to be 0.3602mm, which occurs when the rotor lower is rotated to a position within the confines of the valve seat, where it is compressed to assume the smallest size. This value is input to ANSYS as 5 displacement objects for the faces at the 5 corners, and the solution for the reaction force due to this deformation at each corner is obtained using the ANSYS Static Structural solver. Using the friction coefficient of 0.2 and the ANSYS-generated corner reaction force of 2.6953N, the consequential static frictional force, at the interface between each corner of the rotor lower and the inner surface of the valve seat, is computed to be 0.54N for each corner. The total resulting frictional force during the rotation of the rotor in the valve seat is taken to be the summation of the frictional forces experienced at all the 5 corners of the rotor lower and that, exerted across the interface of the valve seat base and the rotor lower base, under a compressive force of about 5N normal to the base plane of the assembly. The total resulting frictional force experienced by the rotor, during rotation, is computed theoretically to be 3.7N. Using 3.7mm for the rotor diameter, it is theoretically deduced that a maximum torque of 0.0137Nm is required for turning the rotor in the valve seat. This value is well within the torque range of many small stepper motors.

Fig. 11.View showing vectors of deformation at the 5 corners of the pentagonal edges of the rotor lower (left). ANSYS solution visualization showing the rotor lower at the onset of deformation (middle) and at midway to maximum deformation (right).

4.2 Experimental results and discussions

Figures 12 show the evolution of the hydrostatic leakage pressures encountered in the microvalve, during forward (top graph) and backward fluid flow (bottom graph), with respect to the compressive force applied on the microvalve. Generally the greater the compressive force, the greater the maximum tolerable hydrostatic pressure before leakage, or the greater the leakage pressure, for both forward and backward fluid flows. Therefore, it is advantageous to impose a high compressive force within the microvalve assembly. However, as mentioned earlier, due to the presence of ribs on the perimeter and around the channel inlets and outlets, higher compressive forces lead to a greater area of contact between the stator and rotor, thus lowering the contact pressures around the inlets and outlets. Moreover, a higher compressive force would increase the switching torque required. Hence, it is important to implement an optimum compressive force, to achieve a sufficient leakage pressure, and also allowing for a feasible switching torque.

The optimal design of the microvalve is found to be one that compresses the rotor by 0.14 mm, to apply a pressing force of 5 N onto the rotor. The switching torque of the microvalve is below 0.03 Nm, which is within the torque rating of small stepper motors which are commercially available. One such suitable stepper motor would be the Johnson Electric Unipolar Stepper Motor (model UAG2) [33]. Under such a compressive force, the forward flow leakage pressure is 373 kPa, while the backward flow leakage pressure is 310kPa. The dead volume of the microvalve design is zero, as the entire volume in the channel is swept during switching. However, the swept volume might be an important consideration, especially for sensitive analysis applications where a large swept volume during switching may be problematic. A small microvalve in this case is advantageous as the swept volume is usually a function of the rotor diameter, as in Hasegawa's design [19]. In the microvalve presented in this project, the swept volume is simply the volume of the channel in the rotor lower. This volume is calculated to be 268 nanolitres.

Fig. 12. Measured leakage pressure with respect to compressive force imposed on microvalve assembly under forward flow (Top); Measured leakage pressure with respect to compressive force imposed on microvalve assembly under backward flow (Bottom)

5. Conclusions

A microvalve of simple construction which is more easily fabricated and integrated into Lab-on-chip devices, without complex additional guiding components, has been realized. The designed microvalve has a rotor that is only 3.7 mm in diameter, resulting in footprint of only 14 mm² (almost only half that of the multi-way microvalve design by Luharuka and Hesketh et al) and is able to withstand hydrostatic pressure of up to 373kPa in the forward flow direction and up to 310kPa of pressure in the backward flow direction, when operating under compressive forces of 5 N. The torque required to switch the microvalve between outlets was measured to be less than 0.03 Nm, making it feasible for automatic operation by commercially available small stepper motors. The novel use of a pentagonal-shaped rotor, allows for a precise switching among the outlets, for the inlet in the rotor, while eliminating the need for additional guiding structures which may be challenging to fabricate and integrate into the LOC device. This microvalve also features a zero dead volume, and a small sweep volume of only 268 nanolitres, achieved by the incorporation of a transfer channel, inside the rotor, with a cross sectional area that matches that of the inlets and outlets.

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