MICROMACHINED HYDRAULIC ASTABLE MULTIVIBRATOR

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1 Abstract

The paper presents a hydraulic astable multivibrator system. The system consists of hydraulic resistors, hydraulic capacitors and hydraulic pressure controlled valves. The system is designed, realised and tested. The measured system behaviour agrees well with the model simulations. The free running frequency of the multivibrator is 0.18 Hz and the output pressure swing is 90% of the supply pressure.

2 Introduction

An important application area for micro systems is in the field of micro analysis systems. Important components in those systems are channels, junctions, filters, mixers, valves, pumps and other micro liquid handling components. For the pump action, the reciprocating pump is a widely used pump principle [1,2,3]. The pump action is performed by (periodically) changing the pump chamber volume. These kinds of pumps can be actuated with a periodically varying pressure. The presented hydraulic multivibrator is one way to generate (high) actuation pressures for micro-liquid pumps and fluid compressors. Key-component in the presented system is a pressure controlled valve. The valve is basically a pressure controlled resistor element [4]. The hydraulic circuit function of this valve can be well compared with the p-MOS transistor functionality in electronic circuits. The hydraulic astable multivibrator system has been designed using a lumped element approach.

3 Astable Multivibrators

Figure 1 shows a basic block diagram of a multivibrator.

Figure 1. Basic block diagram of a multivibrator with inverting amplifiers (A) and coupling networks (CN).

Blocks A1 and A2 are both inverting amplifiers, composing one positive-feedback amplifier. Blocks CN1 and CN2 are coupling networks. Depending on

the choice of the coupling networks: a) both resistive, b) one resistive and one capacitive, c) both capacitive, a bistable, monostable or astable multivibrator is composed respectively [5]. Astable multivibrators are examples of oscillating systems, they generate a block shaped periodic signal.

The basic schematic circuit of an electrical astable multivibrator is given in figure 2. The multivibrator has p-MOS transistors as active element and simple resistor-capacitor elements as coupling networks. The input signal of T_1 , assuming an ideal switching behaviour of the p-MOS transistors at $V_{in} = 3/4 \cdot V_{supply}$, is given in figure 3. If the input voltage of T_1 is $V_{in} = 1/4 \cdot V_{supply}$ then the transistor T_2 is at its switching point.

Figure 2 Circuit diagram of astable multivibrator with p-MOS transistors. C_1, R_1, R_2 and C_2, R_3, R_4 are the coupling networks and T_1, R_5 and T_2, R_6 are the inverting amplifiers.

Figure 3. Top: The output voltage of transistor T_2 available over resistor R_6 . Bottom: Input signal of the p-MOS transistor T_1 (see figure 2). The signal relaxes to $V = 1/2 \cdot V_{supply}$, defined by the resistors R_1 and R_2 .

4 Hydraulic Multivibrator Design

The hydraulic multivibrator is based on an existing electronic astable multivibrator. We apply the analogy between the electrical and the hydraulic domain, known from the physical system theory. The electrical effort *voltage* is replaced by the hydraulic effort *pressure* and the electrical flow *current* is replaced by the hydraulic flow *volume flow*. Figure 4 shows the so called Ideal Physical Model (IPM) of a hydraulic astable multivibrator.

Figure 4. Ideal Physical Model (IPM) of hydraulic astable multivibrator (compare figure 2).

In the hydraulic astable multivibrator the active elements are pressure controlled valves (PCV's). The valve PCV₁ together with the resistor R₅, and the valve PCV₂ with the resistor R₆ compose the hydraulic inverting amplifiers. The two high pass filters are composed of capacitor C₁ together with resistors R₁ and R₂, and of capacitor C₂ with R₃ and R₄. A switching event can be initiated by both the invertors, because the change of state of the output of one inverter is passed to the input of the next inverter, by means of the series capacitors. The next switching event is triggered by the relaxation process of the high pass filters. Therefore, the oscillation frequency can be controlled by means of the time constants C₁•(R₁//R₂) and C₂•(R₃//R₄).

5 System Components

The resistors, capacitors and pressure controlled valves are all realised in a simple glass-silicium-glass sandwich structure (see figure 5).

- Resistors

Neglecting entrance and exit effects, the flow resistance of the hydraulic resistors, for laminar flow conditions is given by:

$$R = \frac{2 f Re l \mu}{D_h^2 A} \quad [\text{Nsm}^{-5}] \quad (1)$$

with D_h the hydrodynamic diameter of the resistor, A the surface, f the Fanning friction factor, Re the Reynolds

Figure 5. Cross sections of the glass-silicium-glass structures and IPM's of hydraulic multivibrator components.

number, l the length, and μ the liquid viscosity. According to Shah [6], for a laminar fully developed flow, the product of the Fanning friction number and Reynolds number is dimensionless number $fRe = k$, only depending on the shape of the cross section. For a triangular resistor channel with a top width of $2a$ (see figure 6) the resistance is given with

$$R = 17.4 l \mu / a^4 \quad [\text{Nsm}^{-5}]. \quad (2)$$

Entrance and exit effects in the resistor channel result in a non linear relation between pressure drop across and volume flow through the resistor channel. They can be neglected if the channel is long compared to the hydrodynamic entrance length [7]. For a circular tube, the ratio of the entrance length L_{hy} to the diameter d depends linearly on Reynolds number [6].

$$L_{hy}/d = 0.59 + 0.056 Re \quad (3)$$

The resistor channels are bulk-etched in a <100> silicon wafer and closed by bonding a glass wafer to the silicon.

Figure 6. Hydraulic resistor. a) cross section of the flow channel. b) Photograph with top view on the flow channel. The effective channel length is shortened due to under etching of the channel mask.

The resistors are characterised with the experimental set up shown in figure 7. A syringe pump generates a constant liquid flow rate. In the stationary state, the liquid level difference $h_{in} - h_{out}$, between the level in the capillary and the level in the beaker is a measure for the pressure drop across the resistor channel.

Figure 7. Experimental set up for determining the resistor parameters.

Typical results from the measurements on triangular resistor channels are shown in figure 8 and table 1. Table 1 shows the measured (according to the linear fit in figure 8), and the expected (in brackets) resistance values. The hydrodynamic entrance lengths, for an applied pressure of 8000 Pa are listed as well. The non-linear behaviour of resistor R_b can be explained from the relative long entrance length compared to the resistor length.

Figure 8. Measured pressure drop across resistors as a function of the volume flow.

Res.	l [mm]	a [μm]	Resistance $\cdot 10^{11}$ [Nsm ⁻⁵]	L_{hy}/l at 8000 Pa
R_a	2.8	85	9.7 [10]	0.11
R_b	0.8	85	2.9 [3.0]	1.1

Table 1. Measured resistor parameters. The expected resistance values based on eqn. 2 are given in square brackets.

- Capacitors

A cross section of the hydraulic capacitors is given in figure 5. The buffer characteristic of the capacitor is the relation between the pressure difference Δp across the membrane and the stored volume V . For small deflections the capacitor has a linear buffer characteristic: $V = C \cdot \Delta p$. In this case there is a linear relation between the membrane deflection and the applied pressure. Using the theory of pure bending of circular plates [8] we find:

$$C = \frac{\pi a^6}{192 D} \text{ [N}^{-1}\text{m}^5\text{]} \quad (4)$$

with D the flexural rigidity $D = Eh^3/12(1-\nu^2)$ and a the radius of the membrane. The capacitance for a silicon membrane having a diameter of 10 mm and a thickness of 30 μm is $C = 1 \cdot 10^{-12}$ [N⁻¹m⁵].

- Pressure Controlled Valves (PCV)

The pressure controlled valves are normally closed membrane valves. The circular valve membrane has a diameter of 8 mm and a thickness of 30 μm.

Figure 9. Pressure controlled valve. The resistance between port 1 and port 2 is controlled with the pressure at port 3.

The valve membrane is deflected due to the pressures forces resulting from p_1 , p_2 and p_3 . We assume that the membrane deflection $w(r)$ is a linear combination of the deflections resulting from the pressure differences p_{13} and p_{23} , $w(r) = w(r, p_{13}, p_{23})$. Therefore the membrane deflection at $r = a$ is a function of the pressure differences p_{13} and p_{23} and can be characterised with two spring constants [7]:

$$w(a, p_{13}, p_{23}) = \frac{p_{13}}{k_1} + \frac{p_{23}}{k_2} \quad (5)$$

with $k_1 = p_{13}/w(a)$ for ($p_{23} = 0$) and $k_2 = p_{23}/w(a)$ for ($p_{13} = 0$). For a silicon valve membrane with a outer radius $b = 4$ mm and a sealing ridge radius $a = 0.96$ mm we measured $k_1 = 2.1 \pm 0.2 \cdot 10^9$ [Pa m⁻¹] and $k_2 = 3.7 \pm 0.2 \cdot 10^8$ [Pa m⁻¹]. The hydraulic model is based on the membrane deflection from eqn. 5 assuming volume changes shown in figure 10.

Figure 10. a) Detail of the valve cross section. b) Approximation of the volume changes in the valve due to the membrane deflection.

The effective valve opening is less than the opening calculated on eqn. (5) due to a SiO₂ layer on the valve-ridge: $w_e = w(a, p_{13}, p_{13}) - d_{\text{SiO}_2}$. The hydraulic resistance of a gap with a height of w_e between the circular sealing ridge and the base plate, is given by van de Pol [9]. Using this expression we can write for the valve resistance:

$$R_{21}(p_{13}, p_{23}) = \frac{6 \mu \ln(a_2 / a_1)}{\pi w_e^3} \quad (6)$$

The resistance is experimentally investigated using an experimental set up similar to the one shown in figure 7 with the output pressure and port-3 pressure at zero, $p_1 = 0, p_3 = 0$.

Figure 11. Measurement set up for the measurement of the hydraulic resistance of the pressure controlled valve.

Figure 12. Measured and expected volume flow through the valve as a function of the applied pressure p_{23} .

6 Simulations

The hydraulic multivibrator has been modelled and simulated using the bond graph method. The concept of bond-graphs provides a unified method for the description of lumped element models of physical systems in various physical domains [1,11]. Practical implementation of the modelling and the simulation is done using the CAMAS program package [12]. Before the final design is made, the system is modelled and its behaviour simulated to optimise the choice of the values of the different system components.

A bond graph model of the hydraulic multivibrator system is given in figure 13.

Figure 13 Bond graph representation of the hydraulic multivibrator system shown in figure 4.

This bond graph can be obtained from the IPM shown in figure 4 by applying the systematic method of Breedveld [13]. The hydraulic resistors and capacitors can directly be put in bond graph elements. The non-linear behaviour of system components as shown in figure 8 can also be implemented. The PCV's are modelled as a three port element (see figure 14). In bond graph notation this implies that there are three, for the system behaviour relevant ports, through which the valves exchange energy and liquid volume (the conserved quantity in the hydraulic domain).

Figure 14. a) Cross section of the PCV showing the positive directions of the flow. b) Three port bond graph representation of the pressure controlled valve.

The valve is implemented with parameters as discussed in section 4. The values of the resistors R_1 , R_2 , R_3 , R_4 and the capacitors C_1 and C_2 of the high pass coupling networks are chosen, such that we obtain a robust as possible oscillating behaviour; the output low and the output high levels in the oscillator pressure signal (p_5 and p_6 in figure 4) are as widely separated as possible (NB: $C_1=C_2$, $R_1=R_3$, $R_2=R_4$, $R_5=R_6$ and $PCV_1=PCV_2$).

- Frequency of oscillation

The cycle time of the multivibrator is proportional to the time-constant of the coupling network $R_1/R_2 \cdot C_1$. A simulation of the free running multivibrator is given in figure 15.

Figure 15. Simulation of the free running multivibrator, with $R_1=R_2=1 \times 10^{13} [Ns m^{-5}]$ and $C_1=1 \times 10^{-12} [N^{-1} m^3]$. The signals C and D represent the output pressures p_5 and p_6 respectively. A and B represent the input pressures p_1 and p_2 respectively. The oscillation frequency equals 0.16 Hz.

-Choice of C_1

In the case of a switching event, a pressure step applied to the input of the high pass coupling network, is divided between the coupling capacitor and the switching capacitor of the valve. Therefore C_1 should be taken large compared to the input capacitance of PCV_1 . The pressure drop across the coupling capacitor is visible in the simulation shown in figure 16.

Figure 16. Pressure across the coupling capacitor C_1 (curve E), and pressure on the control port of PCV_1 (curve A). Note the sudden pressure drop at a switching event.

7 System Realisation and Experiments

The hydraulic system is implemented using the MFS (Modular Fluid System) concept developed at the MESA research institute [10]. The multivibrator components are realised in one integrated realisation process (see figure 5), which consists of silicon bulk etching and anodic bonding of glass to the silicon [2]. The PCV is combined with the input-resistors to a module M_1 (PCV_1 with R_1 and R_2). The PCV load resistor is combined with the coupling capacitor to a module M_2 (R_5 with C_2).

Two sets of these modules M_1 and M_2 are mounted on a channel substrate which provides the necessary connections (see figure 17). Photographs of the modules are given in figure 18.

Figure 17. Layout of the channel substrate, showing the connections between the modules M_1 and M_2 , and to the outside world (supply and output signals). The scale is 1:1.

Figure 18. a),b) Top and bottom view of the M_2 module (R_5, C_2). c),d) Top and bottom view of the M_1 module (R_3, R_4, PCV_2) (see figure 4). Module size is $15 \times 15 \text{ mm}^2$.

Figure 19 shows the expected and measured relation between the static input and output pressure of the inverter.

Figure 19. Measured and expected pressure transfer characteristic of the hydraulic inverters.

From the figure it can be seen that there is a region (about 4000-7000Pa) where the input pressure variations are amplified. This is essential to obtain oscillation. The (dynamic) step response of the high pass filter in combination with the inverting amplifier is given in figure 20. It is clearly visible that the capacitor resistor combination acts as a high pass filter. Due to the negative amplification the positive input step leads to a negative output pulse. The dynamic pressures are measured with a pressure calibrated liquid flow sensor with a hydraulic input resistance of $5 \cdot 10^{12} \text{ [Nsm}^{-3}]$. Figure 21 shows the measured output pressure (P_0) of the free running multivibrator. The oscillation frequency is 0.18 Hz.

Figure 20. a) Measured step response of high pass filter-inverter combination. b) IPM of the coupling network with the inverting amplifier.

Figure 21. a) Measured output pressure of the free running hydraulic multivibrator. b) IPM of set up.

8 Conclusions

A hydraulic astable multivibrator has been realised and tested. The component and system properties are modelled using the bond graph method. The system behaviour is simulated with the CAMAS program package. Models for the component and system behaviour are developed and verified by experimental results. The results agree well with theory and simulations.

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