出國報告（出國類別：國際研討會）

2013/4/08-15第八屆國際流體傳動技術研討會ICFP與會心得報告

服務機關：國立雲林科技大學機械系
姓名職稱：任志強，教授
派赴國家：中國、杭州
報告日期：2013/4/25
出國期間：2013/4/08~2013/4/15

摘要：

本校（國立雲林科技大學）工學院機械工程系 任志強 教授於2013 年 4 月 08~15 日參加由中國浙江大學機電控制工程研究所所主辦之第八屆ICFP國際流體傳動技術研討會，本人在這次研討會中主要是在 04/10 上午 Session E phase 1 中口頭發表一篇論文 (Virtual Instrument to Acquire the P/Q Characteristic Curve of a Pneumatic Pressure Regulator)，並應邀擔任 Session E phase 2 之主持人。並於會中與來自近20國之學者共聚一堂，交換彼此之心得與經驗；此外，也參觀了浙江大學機械系實驗室以及流體傳動國家重點實驗室等設施。另外，本人利用04/12~04/13二個整天，應台商亞德客（Airtac）之邀請，赴亞德客寧波總部進行參訪。

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一、目的與會議名稱簡介：

第八屆ICFP國際流體傳動技術研討會（The 8th ICFP Int. International Conference on Fluid Power Transmission and Control, Hangzhou 2013）於 2013年4月09~11日由中國浙江大學機電控制工程研究所（Institute of Mechatronic Control Engineering）所主辦，協辦單位如 Chinese Fluid power Transmission and Control Society、The State Key Laboratory of Fluid Power Transmission and Control Ministry of Education, China、Chinese Mechanical Engineering Society等等計有3個國際及9個中國本土組織參與，本研討會的舉辦是以四年為一周期，堪稱是國際流體傳動領域中頗具份量的大型研討會；另在研討會進行之同時，國際著名的數家流體傳動工廠，例如 SMC 等，也同步在現場展示其最新研發產品。本人在這次研討會中主要是在04/10 上午 Session E phase 1 中發表一篇論文 (Virtual Instrument to Acquire the P/Q Characteristic Curve of a Pneumatic Pressure Regulator)，並應邀擔任 Session E phase 2 之主持人。隨著流體傳動技術的不斷發展以及進一步應用的普及，各種諸如新型流體傳動元件與新的應用領域不斷地被開發出來，影響所及是更進一步加速了這一個領域的蓬勃發展。因而，藉由本次研討會，計有來自世界 18 國總計近 100 篇論文以及超過 200 位中外研究者齊聚一堂交換彼此之研究心得與成果。另外值得一提的是在這次研討會中，大會精心安排了在會議期間每日共5~6場 Keynote Lectures，其餘論文均安排在每日 3 場 Parallel sessions 中發表。

圖1: Conference Site and opening address

二、會議過程：

研討會開會地點座落在大陸杭州喜來登酒店國際會議廳，以下將就會議進行中的每日議程作一簡要介紹。

1. 報到：研討會的報到時間是 04/09日 PM 8:30-PM 9:00，地點在喜來登酒店大廳。當晚舉辦歡迎酒會（Welcome Party），時間及地點如下： 04/09日 PM:6:30-9:00 於大會舉辦地點杭州喜來登酒店會議廳，計有約 100位學者參加，筆者也利用此一機會與認識的外國學者交談，分享彼此近年來之研究領域與心得。

2. 第一天議程（04/09）：首日開幕典禮由陸甬祥教授開場後（圖1右）隨即安排了五場次 Keynote Speeches，計有 5位演講者分別來自德國中國日本英國及美國，分別針對 Energy-saving，Tunnel Boring Machine，Multivariable Control，Micro Hydraulics 與 Environmental Fluid Power 等主題發表專題演講，在下午則以3個 Parallel Sessions 進行了 4 小時，共 36 篇論文發表的第一階段論文發表。第一天的會議議程則是重頭戲，且今日實際參與大會的人數也最多，約達150人。大會並於當晚安排了大會晚宴（Conference Banquet）。

3. 第二天議程（ 04/10）：第二天的會議議程一開始則同樣是以 3 場 Parallel sessions 進行了 3 小時 30分鐘，共 34 篇論文發表的第二階段論文發表。下午大會並安排了六場 Plenary Speeches分別針對 Piston Pump，Pneumatic Control，Efficient Mobile Machine，Pressure Pulsation Active Control，Design 以及 Cavitation 等主題。而本人也在此日上午 09:30，Session phase 1- E4 中發表一篇論文（Virtual Instrument to Acquire the P/Q Characteristic Curve of a Pneumatic Pressure Regulator），並應邀擔任 Session E phase 2 之主持人，本次大會在 PM4:50 結束所有論文發表。大會並於當晚安排了Farewell Reception，計有約 100位學者參加。

4. 第三天議程（04/11）：本日大會安排參訪浙江大學機械系實驗室以及流體傳動國家重點實驗室，隨後並同時參觀杭州著名的西溪濕地國家公園，計有約100位學者分乘3部遊覽車

參加，之後結束所有行程，互道珍重再見。

另外，本人利用04/12~04/13二個整天，應台商亞德客（Airtac）之邀請，赴亞德客寧波總部進行參訪，除了參觀總部各個加工廠房之外，我個人對於亞德客積極引入自動化感到十分佩服，也對於亞德客公司不論是在規模或廠房面積等均留下深刻印象，驗證了行前台灣流體傳動公會高鳴祥總幹事給本人的提示：亞德客是台灣氣動廠商中最耀眼的一顆星。由於涉及保密，無法公布其他參訪細節。

圖2 :拜訪亞德客

<table>
<tr><th colspan="3">ICFP2013 Program Tuesday April 9, 2013</th></tr>
<tr><td colspan="3">08:30-09:00 Opening Ceremony (XIXI 1&2)</td></tr>
<tr><td colspan="3">09:00-11:50 Keynote Speeches (XIXI 1&2)</td></tr>
<tr><td colspan="3">12:00-13:20 Lunch (Front Hall of XIXI Ballroom)</td></tr>
<tr><td>13:30-17:50 Parallel Session A
(Room XIYUAN 1)
"Development of Hydraulic Components"</td><td>13:30-17:50 Parallel Session B
(Room XIYUAN 3)
"Innovation in Hydraulic Systems"</td><td>13:30-17:50 Parallel Session C
(Room XIYUAN 4)
"Simulation"</td></tr>
<tr><td>13:30-15:30 Phase 1: A1-A6
Chairmen:
Prof. Hua Zhou (China)
Prof. Andrew Plummer (UK)</td><td>13:30-15:30 Phase 1: B1-B6
Chairmen:
Prof. Guofang Gong(China)
Prof. Kim A. Stelson (USA)</td><td>13:30-15:30 Phase 1: C1-C6
Chairmen:
Prof. Songjing Li (China)
Prof. Mao-Hsiung Chiang (Taiwan)</td></tr>
<tr><td>15:50-17:50 Phase 2: A7-A12
Chairmen:
Prof. Hong Ji (China)
Prof. Radovan Petrovic (Serbia)</td><td>15:50-17:50 Phase 2: B7-B12
Chairmen:
Prof. Dingxuan Zhao (China)
Prof. Jouni Mattila (Finland)</td><td>15:50-17:50 Phase 2: C7-C12
Chairmen:
Prof. Canjun Yang(China)
Prof. Shinichi Yokota (Japan)</td></tr>
<tr><td colspan="3">18:45-21:30 Banquet (XIXI 1&2) Hosted by Prof. Huayong Yang</td></tr>
</table>

<table>
<tr><th colspan="3">ICFP2013 Program Wednesday April 10, 2013</th></tr>
<tr><td>08:30-11:50 Parallel Session D
(Boardroom)
"Mechatronics and Automation"</td><td>08:30-11:50 Parallel Session E
(Room XIYUAN 3)
"Advances in Pneumatics"</td><td>08:30-11:50 Parallel Session F
(Room XIYUAN 4)
"Fundamental Study of Fluid Power"</td></tr>
<tr><td>08:30-10:30 Phase 1: D1-D6
Chairmen:
Prof. Xiangdong Kong (China)
Prof. Heikki Handroos (Finland)</td><td>08:30-10:30 Phase 1: E1-E6
Chairmen:
Prof. Xuanyin Wang (China)
Prof. Toshiharu Kagawa (Japan)</td><td>08:30-10:30 Phase 1: F1-F6
Chairmen:
Prof. Long Quan (China)
Dr. Heinrich Theissen (Germany)</td></tr>
<tr><td>10:50-11:50 Phase 2: D7-D9
Chairmen:
Prof. Dianrong Gao (China)
Prof. Petter Krus (Sweden)</td><td>10:50-11:50 Phase 2:E7-E9
Chairmen:
Prof. Jyh-Chyang Renn (Taiwan)
Prof. So-Nam Yun (Korea)</td><td>10:50-11:50 Phase 2: F7- F9
Chairmen:
Prof. Yaobao Yin (China)
Prof. Kyoung Kwan Ahn (Korea)</td></tr>
<tr><td colspan="3">12:00-13:20 Lunch (Front Hall of XIXI Ballroom)</td></tr>
<tr><td colspan="3">13:30-16:50 Keynote Speeches (XIXI 1&2)</td></tr>
<tr><td colspan="3">18:30-21:00 Farewell Reception (Feast Restaurant of Hotel West Wing) Hosted by Prof. Bingfeng Ju</td></tr>
</table>

<table>
<tr><th>ICFP2013 Thursday April 11, 2013</th></tr>
<tr><td>09:30-Noon Tour to Xixi National Wetland Park or West Lake</td></tr>
</table>

Tips: Shuttle buses will be arranged by the organizer to pick up the participants from conference venue to the State Key Lab of Fluid Power Transmission and Control every one hour during parallel sessions.

圖3: Conference Program

三、與會心得與建議：

流體傳動（一般又稱之為：液氣壓控制）相關研究領域在國內並非顯學，但這項技術卻廣泛地應用在各式工作母機，射出成型機以及行走機械上，而這些產品每年也為國家創造不少外匯收入。相對的，在國外先進國家中，包含德國、日本、美國、英國、意大利、芬蘭、瑞典等甚至是對岸的中國大陸，均有相當強的研究團隊持續從事著流體傳動技術的研發，他們可能是大學中一個獨立的研究所或是在大學中成立的一個技術研發中心。反觀國內，從事流體動傳技術研發的學者專家本來就不多，而且又分散在各個大學或研究機構，故整合起來的力量是不夠強的。以流體傳動相關的國際研討會來看，上述先進國家幾乎每二至四年均會舉辦一次，就連中國大陸在這方面的發展也不容忽視，他們每二年均會在杭州、武漢或哈爾濱等地舉辦流體動力國際研討會；但是在台灣，由於力量分散再加上流體傳動並不是顯學，所以過去幾乎沒有舉辦國際流體傳動研討會的紀錄。因此，對本人而言，參與這次會議之後所獲得的最大心得是：經由論文發表及討論的方式，可以與來自全世界相關領域專家學者一起切磋，廣結善緣並增廣不少見聞，也學習到很多的新的專業知識以及未來值得投入的研究領域，收穫良多。在此，要特別感謝國立雲林科技大學以及國科會所給予經費上的資助，方得以成行，不勝感激。

四、附錄I：攜回資料：

此次會議本人總計攜回以下三份資料，分別是：

1. 大會論文發表日程（紙本）一本。
2. 大會論文集摘要集一冊以及全文光碟CD一份，總共收錄了近100篇論文。
3. 大會參與者之名錄以及聯絡方式一份。

伍、附錄II：發表論文：

Virtual Instrument to Acquire the P/Q Characteristic Curve of a Pneumatic Pressure Regulator

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[2] Graduate Student,
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Abstract

In this paper, a new concept of virtual instrument for acquiring the pressure/flow (P/Q) characteristics of a pneumatic pressure regulator is proposed. This concept is actually based on the well-known CFD simulations and dynamic modeling. Generally speaking, to measure the actual P/Q characteristics of a pneumatic pressure regulator, a real test bench according ISO 6953 has to be utilized. However, such a real test bench may be complicated and expensive. Therefore, a software virtual instrument is presented in this paper with the intention to replace the hardware test bench. In details, the commercial CFDRC software is chosen to find the sonic conductive C and the critical pressure ratio b for the main metering restrictor in the pneumatic pressure regulator according to ISO 6358. In addition, the ISO 6953 standard is also adopted for further acquiring the P/Q characteristic curve. An efficient MatLab/Simulink dynamic model for the pneumatic pressure regulator is also established to simulate the P/Q characteristic curve continuously according to ISO6953. Finally, a real test bench for the pneumatic pressure regulator is constructed to verify the validity of the proposed concept of virtual instrument. After experiments, it is proved that the concept of virtual instrument is feasible and may be applied to other fluid power components in the future.

Keywords：Pneumatics, CFD, Pneumatic Pressure Regulator, ISO6953, ISO 6358

1 INTRODUCTION

For a pneumatic pressure regulator, the pressure/flow (P/Q) curve is perhaps the most important characteristic. However, to measure the actual P/Q characteristics of a pneumatic pressure regulator, a real test bench according ISO 6953 has to be utilized. Such a real test bench may be complicated and expensive. In this paper, therefore, a software virtual instrument is proposed with the intention to replace the expensive hardware test bench. In details, the commercial CFDRC software is chosen to find the sonic conductive C and the critical pressure ratio b for the main metering restrictor in the pneumatic pressure regulator according to ISO 6358. In addition, the ISO 6953 standard is also adopted for further acquiring the P/Q characteristic curve. An efficient MatLab/Simulink dynamic model for the pneumatic pressure regulator is also established to simulate the P/Q characteristic curve continuously according to ISO 6953. Figure 1 shows the schematic of the chosen pneumatic pressure regulator. The main metering restrictor is shown in Fig. 2. In this paper, the well-known ISO 6358 standard is utilized to model the mass flow-rate characteristics. Figure 3 indicates a typical mass flow-rate characteristic of the airflow through a flow restrictor according to ISO 6358 [1]. It is observed that the value of the maximal mass flow-rate of the airflow is a constant and depends on the value of C, which is generally called the sonic conductance. When the ratio of the outlet pressure to the inlet pressure is small, meaning that the pressure drop across the flow restrictor is quite large, the airflow through the flow restrictor become a jet with high velocity. The acoustic velocity, however, physically limits this velocity. This is precisely the reason why the maximal constant flow-rate exists in the airflow property shown in Fig. 3.

The parameter b, called the critical pressure ratio, is defined as the intersection of the straight flow-rate of so-called choked flow and the elliptic flow-rate curve of subsonic flow as shown in Fig. 3. For a flow restrictor with a fixed cross-sectional area, the corresponding two parameters, the sonic conductance, C, and the critical pressure ratio, b, are constants [2, 3]. In the following, the CFD-modeling of the mass flow-rate characteristic is firstly outlined.

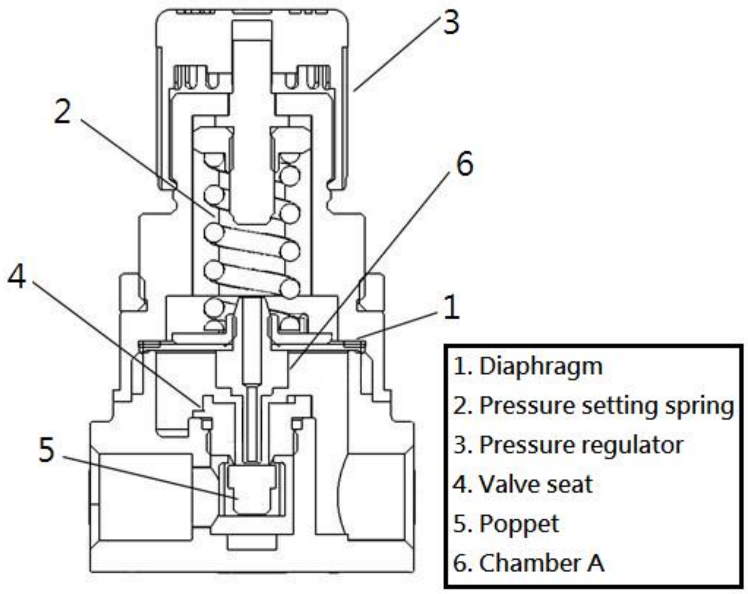

Fig. 1: *The chosen pneumatic pressure regulator*

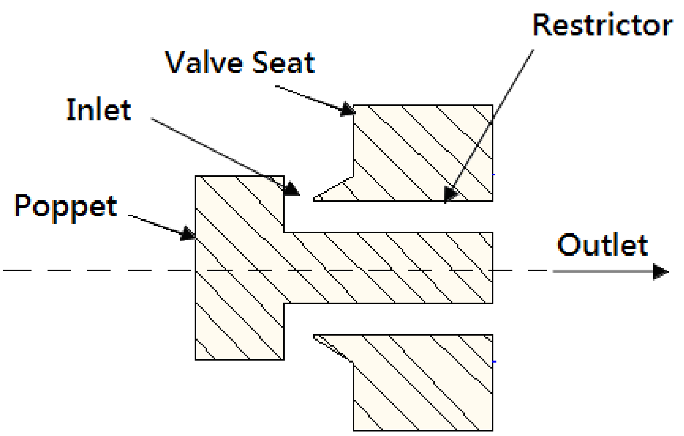

Fig. 2: *The geometry of the main metering restrictor*

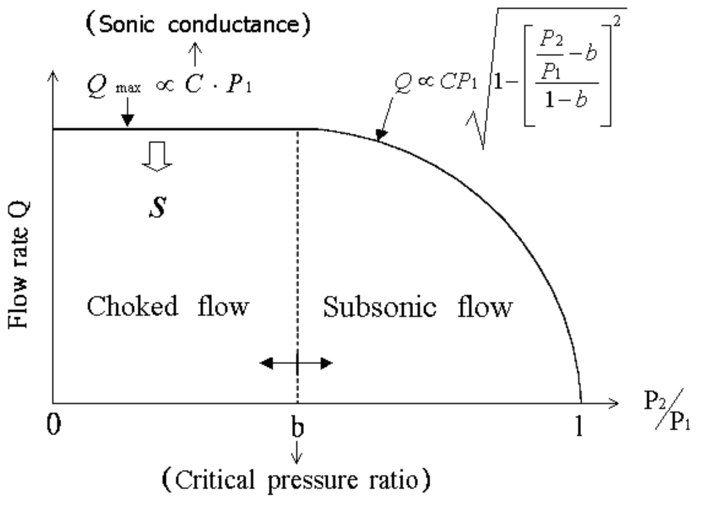

Fig. 3: *The typical airflow property through a pneumatic flow restrictor*

2 CFD-MODELING OF THE MASS FLOW-RATE CHARACTERISTIC

To implement the software virtual instrument accurately, the airflow property through the main metering restrictor (shown in Fig. 2) has to be determined in advance. In this paper, the commercial CFDRC software is used as a tool to determine the above mentioned air-flow property. The mass flow-rate of airflow through a pneumatic flow restrictor can be described by the following generalized formulas. [1-3]

Choked flow：

$$\dot{m}_{\max} = CP_1\sqrt{\frac{T_2}{T_1}} \quad for \quad \frac{P_2}{P_1} \le b\,, \tag{1}$$

Subsonic flow：

$$\dot{m} = CP_1\sqrt{\frac{T_2}{T_1}}\sqrt{1-\left(\frac{P_2/P_1-b}{1-b}\right)^2} \quad for \quad \frac{P_2}{P_1} > b\,. \tag{2}$$

Since the airflow in the metering restrictor is axis-symmetric, a two-dimensional Cartesian coordinate model of the flow field is established for the quantitative analysis of the mass flow-rate through the test restrictor (opening width: 1.0 mm) as shown in Fig. 4.

The initial and boundary conditions for the CFD simulation are:

(1) The input velocity of the airflow is determined by trial-and-error approach to meet the preset air pressure at the inlet.
(2) The airflow is compressible.
(3) The air density is assumed to be 1.189 kg/m^3 at room temperature.
(4) The dynamic viscosity of the air is set to be 1.789 e^{-5} Ns/m^2 at room temperature.
(5) The room temperature is set to be 20 $^\circ C$ or 293 *K*.
(6) The reference acoustic velocity of air is assumed to be 340 m/s.
(7) The boundaries of the flow field are considered as the wall, which means that no airflow across the boundaries is allowed.

The velocity vector diagram for the main metering restrictor is found and shown in Fig. 5. From this velocity vector diagram and the corresponding numerical velocity data at the outlet, the average mass flow-rate of the air can be derived. Finally, the mass flow-rate characteristic curves for different opening widths can be obtained as shown in Fig. 6. The critical pressure ration is found to be 0.291 and the sonic conductance, C, can be calculated by Eq. (3), where y denotes the opening width (unit: mm). Using the test device described in [5], it is found that the maximal deviation between simulation and experimental results is around 5.6 %.

$$C = -0.9608y^2 + 1.926y \tag{3}$$

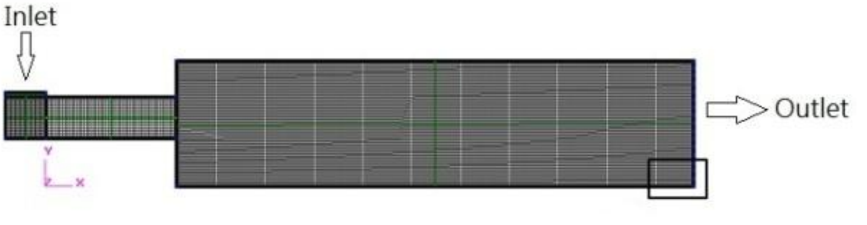

Fig. 4：*Two-dimensional Cartesian coordinate mesh model of the flow field (Opening width= 1.0mm)*

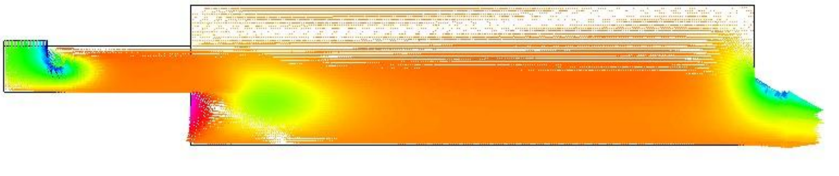

Fig.5：*The velocity vector diagram*

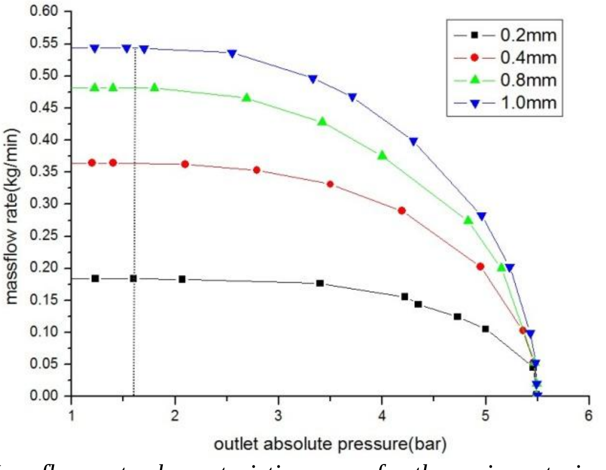

Fig. 6: *Mass flow-rate characteristic curves for the main metering restrictor*

3 EQUATION OF MOTION FOR THE PNEUMATIC PRESSURE REGULATOR

In the previous section, the mass flow-rate model for the main metering restrictor in the pneumatic pressure regulator was successfully established. In this section, the equation of motion for the poppet in the pressure regulator is derived. Figure 7 shows the scheme of the poppet together with the valve body geometry. It is worth mentioning that the pressure in chamber A (P_A) is actually the controlled output pressure of the pressure regulator. If the pressure P_A is smaller than the target preset pressure, then the poppet will move downwards to open the metering restrictor. Thus, the output pressure will increase because of the larger opening area. Similarly, the poppet will move upwards to close the metering restrictor gradually if the output pressure is greater than the target pressure setting. To change the target output pressure, the pre-compression of the pressure setting spring has to be adjusted by hand. From Newton's law, the following two equations can be obtained.

(1) If the metering restrictor is closed:

$$\begin{aligned} &-P_s \times A_2 - P_A \times A_1 + (y_{01} + y_i - y) \times k_1 \\ &-(y_{02} + y) \times k_2 + M \times 9.8 = M \frac{d^2 y}{dt^2} \end{aligned} \quad . \qquad (4)$$

(2) If the metering restrictor is still open:

$$\begin{aligned} &-P_s \times A_3 - P_A \times A_1 + (y_{01} + y_i - y) \times k_1 \\ &-(y_{02} + y) \times k_2 + M \times 9.8 = M \frac{d^2 y}{dt^2} \end{aligned} \quad . \qquad (5)$$

Where M: Mass of the poppet,
Ps: Supply pressure,
P_A : Pressure in chamber A,
A_1 : Area of membrane,
A_2 : Area of the poppet head,
A_3 : Area of the connecting rod,
y_{01} : Pre-compression of pressure setting spring,
y_{02} : Pre-compression of the recovery spring,
y : Displacement of the poppet or the opening width of metering restrictor,
k_1 : Pressure setting spring constant,
k_2 : Recovery spring constant.

The position of poppet decides which one of these two equations should be used. For example, if the metering restrictor is fully closed, then Eq. (4) should be utilized. Now, it is essential to build a mathematical model relating the pressure gradient in chamber A to the corresponding net mass flow-rate. To simplify the model, it is reasonable to assume that the heat-loss term and the kinetic-energy term are both negligible [2, 3]. In addition, the volume of chamber A is a constant regardless of the airflow. Thus, the simplified differential equation for the pressure gradient in chamber A is obtained as

$$\dot{P}_A = \frac{kRT_A}{V_A}(\dot{m}_1 - \dot{m}_2)\cdot \qquad (6)$$

Where $\dot{P}_A$: Pressure gradient in chamber A,

$\dot{m}_1$: Mass flow-rate into the chamber A,

$\dot{m}_2$: Mass flow-rate out of the chamber A,

R : Air constant (29.26 m/°K),

T_A: Room temperature (°K),

V_A: Volume of chamber A,

k : Adiabatic constant (k=1.4).

Combining Eq. (1) through Eq. (6) and using MatLab/Simulink software, the overall simulation block diagram can be obtained as shown in Fig. 8. The simulation and experimental dynamic pressure responses for two different output target pressure settings are shown in Fig. 9 and Fig. 10, respectively. It is observed that the simulation dynamic pressure responses agree quite well with the experimental ones. Furthermore, to acquire the P/Q characteristic curves of the pressure regulator, a stepwise variable input signal of opening width, *y*, to the dynamic simulation program is utilized as shown in Fig. 11. Such an input signal represents the stepwise adjustment of output target pressure by hand from maximal 1.6 bar to minimal zero bar. On the other hand, a real test bench for the pneumatic pressure regulator according to ISO 6953 standard is constructed as shown in Fig. 12. After some experiments, the comparisons between the experimental and simulation P/Q characteristic family curves of the chosen pneumatic pressure regulator is obtained as shown in Fig. 13. It can be seen that the simulation results agree very well with the experimental ones. This verifies the validity of the proposed concept of virtual instrument.

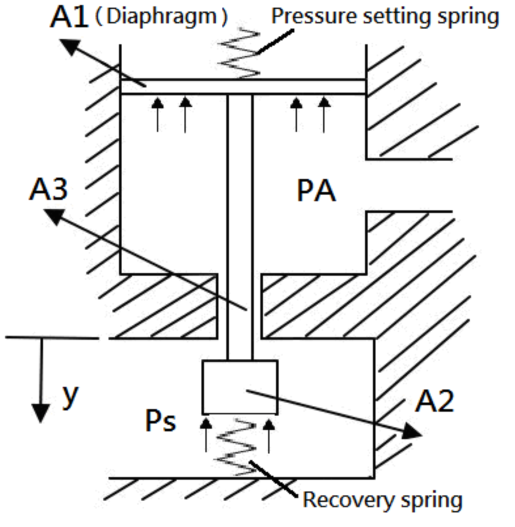

Fig. 7：*Scheme of the poppet and valve body geometry*

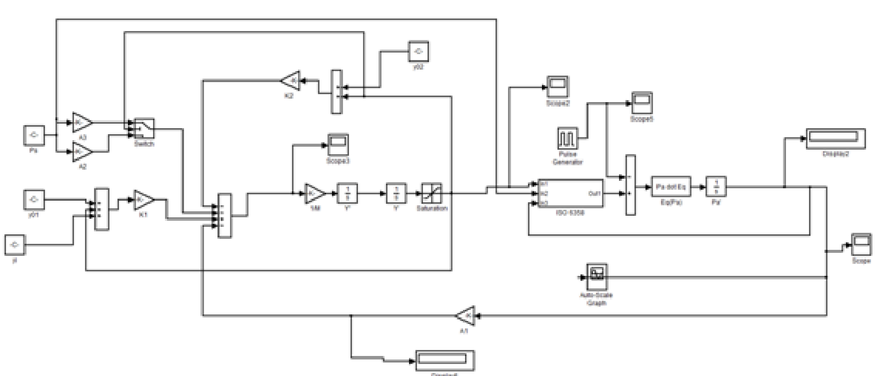

Fig. 8：*Dynamic simulation block diagram using MatLab/Simulink*

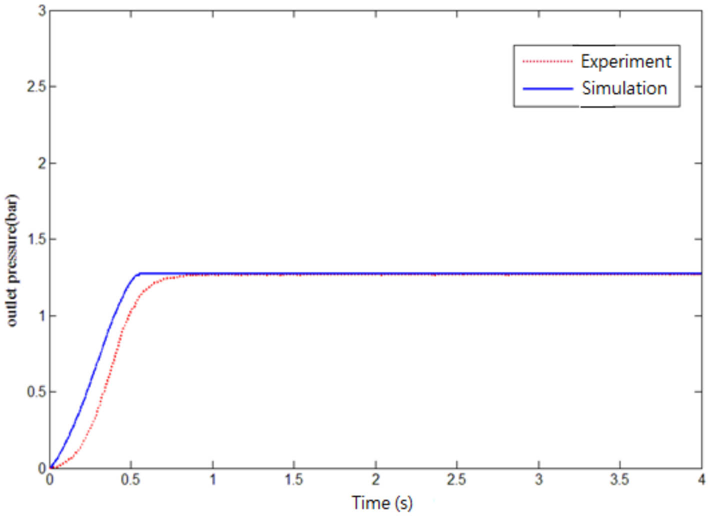

Fig. 9: *Simulation and experimental dynamic pressure response for 1.25 bar output pressure control*

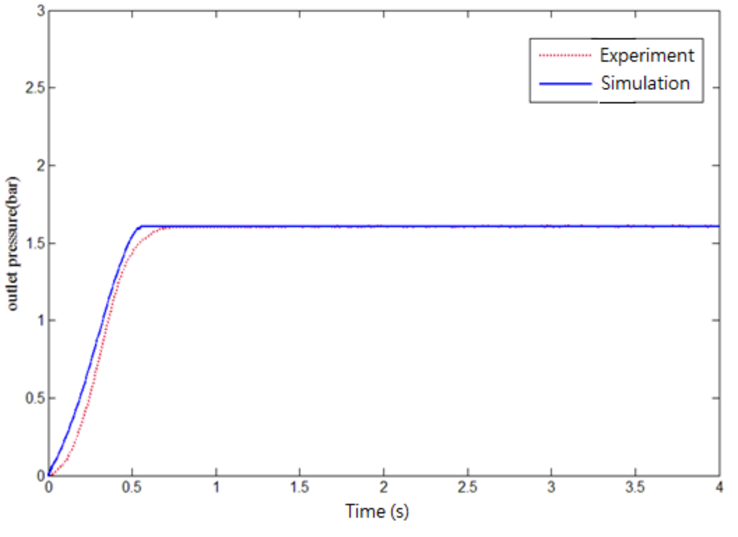

Fig. 10: *Simulation and experimental dynamic pressure response for 1.6 bar output pressure control*

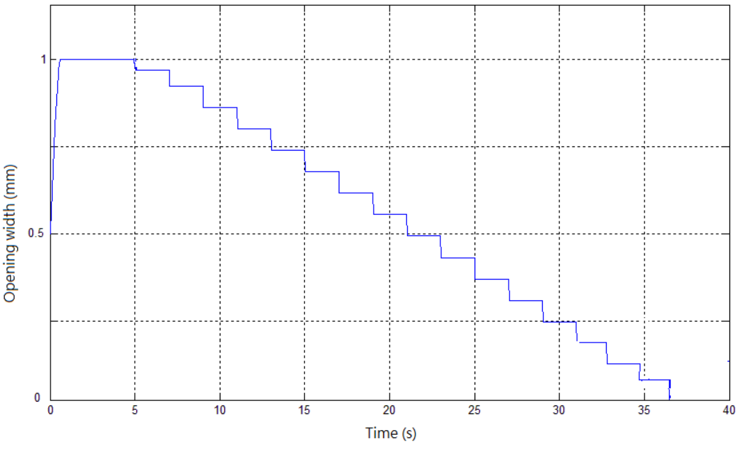

Fig. 11: *Stepwise variable input signal of opening width for the continuous measurement of the P/Q char. curves*

4 CONCLUSION

In this paper, the proposed new concept of virtual instrument to acquire the P/Q characteristics of a pneumatic pressure regulator is not only proved to be feasible but also successfully realized. After some simulations and experiments, it is observed that the simulation P/Q characteristic family curves agree very well with experimental ones. A most important feature of the proposed new concept is that the complex and expensive real test bench may no longer be necessary as long as the modeling of the pneumatic pressure regulator is accurate. Therefore, it is expected that such a new concept of virtual instrument may be applied to other fluid power components in the future.

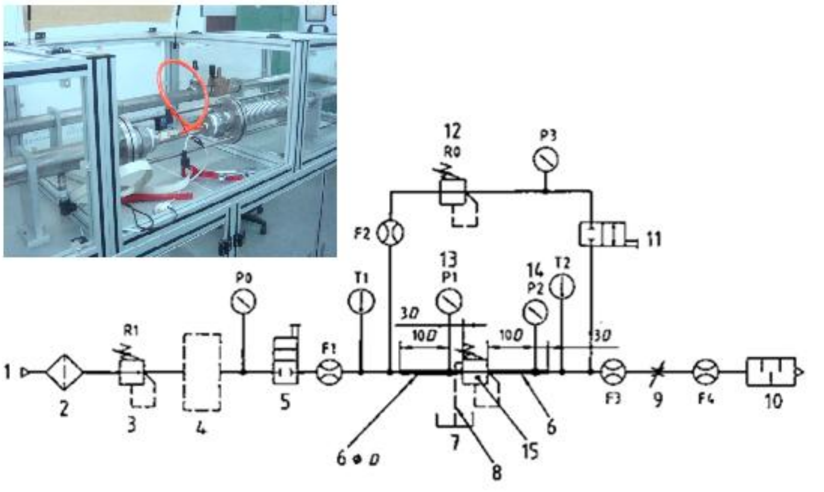

Fig. 12: *Real test bench for the pneumatic pressure regulator according to ISO 6953 standard*

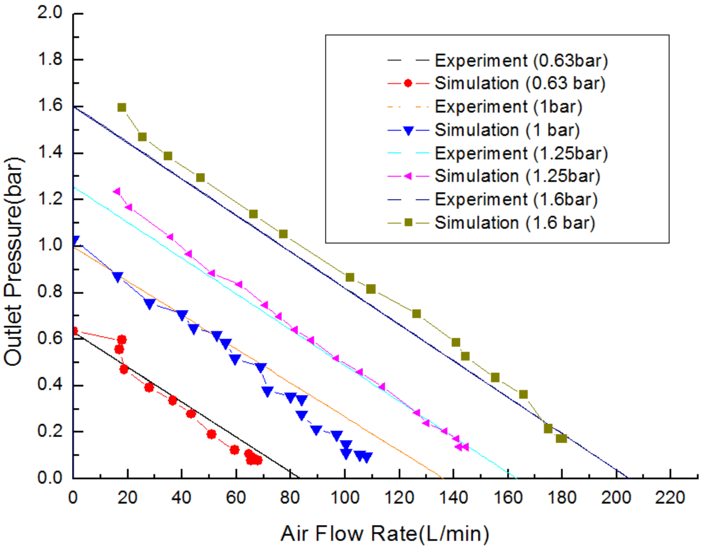

Fig. 13：*Comparisons between the experimental and simulation P/Q characteristic family curves*

5 ACKNOWLEDGMENT

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