SEMI-ACTIVE CONTROL OF WIND IMPACT EFFECTS BY MEANS OF PNEUMATIC SYSTEM

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1 SUMMARY
Possibilities of new applications of smart devices arise with the development of new generation wind turbines. Rotor up-scaling implies a need to control dynamic characteristics of loads transferred from blades to the hub and tower. The motivation for this work was to demonstrate an approach to control dynamic reaction force resulting from a wind blow. A controllable gas spring was chosen as a 'smart bond' between two sub-structures. The goal of the control was to keep the value of force in 'smart bond' below some critical level throughout the blow duration. Dynamic numerical analysis was performed with real-time feedback between a sensor and semi-active control device. A small-scale experimental set-up was developed with piezo-actuated valve to control the air flow between a pneumatic piston and an air reservoir. The aim for the tests will be to meet the control goal, while keeping the fixed amount of medium in the pneumatic system, i.e. to work in a repetitive and semi-active manner.

2 INTRODUCTION
One of major technological limitations of up-scaling of modern wind turbines is the load bearing capacity of blade materials. Connection between blade and hub has to withstand very large bending moments in particular during wind blows. An alternative for developing new blade materials is to introduce a semi-actively controlled connection between blade and hub with controllable characteristics. This work demonstrates the idea of controlling the value of peak dynamic force that is transferred to the support by means of pneumatic system with controllable on/off piezo valve. Pneumatic pistons together with controllable valves and pressure supply systems are used as smart actuators, typically as handling or positioning devices [1,2,3]. On contrary to these applications in present approach pressure is withdrawn from the cylinder with use of smart valve and the whole system works in semi-active way without applying external energy.
Piezo materials are used in pneumatic systems either to allow for very fine positioning precision [2] or as piezo actuator for micro valve opening [4]. Problems encountered for common pneumatic valves are long response times and, especially for low pressures, need to use a pilot valve. In case of present solution of mechanically amplified piezo which drives the valve orifice there is no need for pilot valve and the response time is of magnitude of single milliseconds. In the first sections of the paper a numerical study is presented to demonstrate the idea and asses the effectiveness of solution. Then experimental set-up is presented.

3 NUMERICAL SIMULATION

3.1 Model description

The numerical model of the system consists of two major elements, i.e. slender, elastic part which can be viewed as a demonstrator 'blade' and a pneumatic piston which transfers the dynamic force to the support (cf. Fig.1). The pneumatic spring together with a controllable outlet forms a 'smart bond' between the blade and the support. Further it is assumed that the rising time of a wind blow is slow enough, so that the piston velocity does not exceed 1 m/s and the heating effects are negligible.

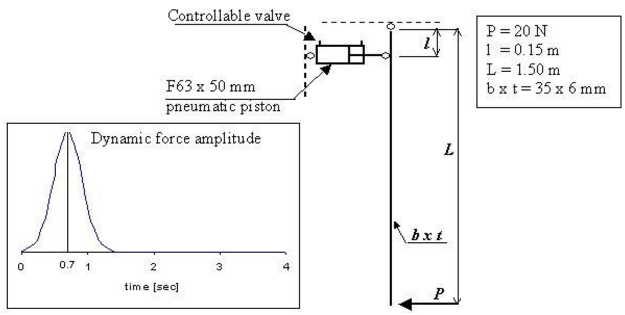


Figure 1: Numerical model features

Also, the working gas pressure in cylinder should not exceed 1 MPa. Given the above, the process within the cylinder was assumed as an adiabatic transformation of an ideal gas, at constant temperature T = 21ºC, according to the equation:

$$p \cdot V = n \cdot R \cdot T \tag{1}$$

It follows that the piston behaves like a non-linear spring with a hyperbolic characteristic. From the controllability standpoint it is preferable to use piston with possibly big diameter due to the fact that given force could be obtained at smaller stroke and thus possibly long stroke could be used during the active phase of the process. ϕ 63 mm piston with the stroke of 50 mm was modeled in the simulation.

The simulation was performed with Abaqus/Standard code. Piston and remaining part of the pneumatic system was modeled with Fluid Cavity elements. Before adding control, results were compared with those obtained with non-linear Spring elements (cf. Fig. 2).

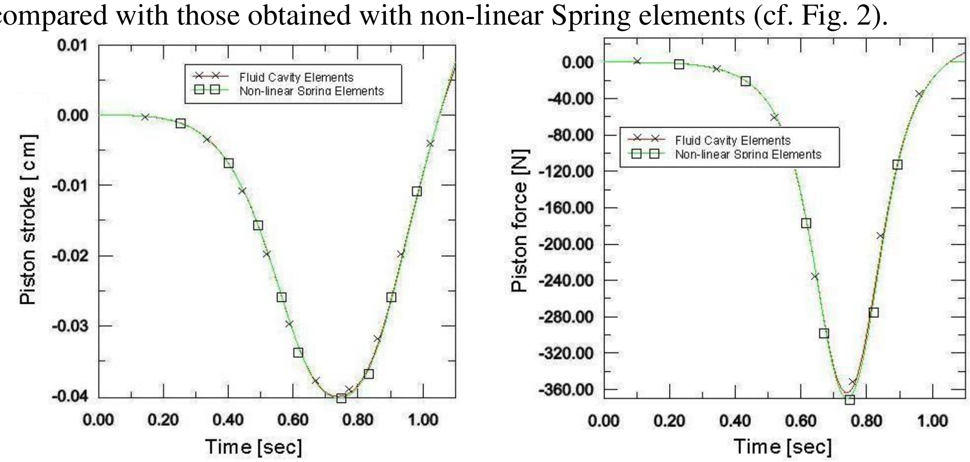


Figure 2: Fluid Cavity and Non-linear Spring elements comparison

3.2 Controllable valve

Air flow through the controllable valve was modeled with the equation:

$$\Delta p(c,t) = C_V(c) \cdot q(t) \tag{2}$$

Pressure difference Δp between the pneumatic cylinder and the environment is proportional to the rate of mass flow $q(t)$. The flow resistance $C_V(c)$ depends on the control variable c:

$$C_V = \begin{cases} 1.000e+19, & \textit{for } c=-1 \quad (\textit{valve closed}) \\ 0.155e+9, & \textit{for } c=1 \quad (\textit{valve open}) \end{cases} \tag{3}$$

Flow resistance for the open valve was chosen so that it corresponds to the flow rate of 30 l/min at the pressure difference of 0.1 MPa.

3.3 Aim of control

The goal for the control procedure is to keep the value of piston force F_P below some critical level F_{cr}. This is achieved by means of opening the valve whenever $F_P \geq F_{cr}$ and closing it if F_P drops below the critical value. At some point during the blow decay sign of F_P is changed and underpressure starts to build up in the cylinder. At this point the valve is opened again in order for the air to flow back into the cylinder. Control procedure described above can be summarized in the diagram shown on Figure 3.

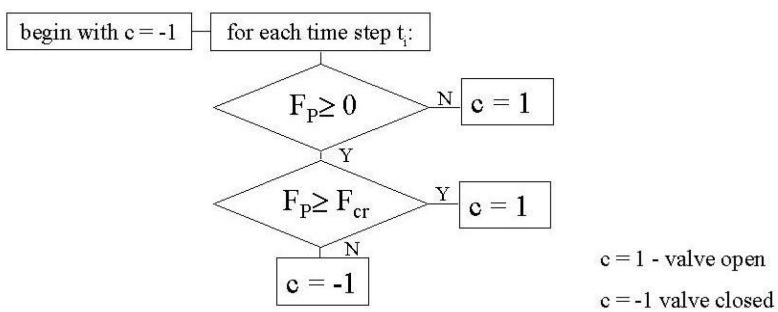


Figure 3: Control procedure

3.4 Numerical results

Use of piston stroke is depicted on Figure 4. The maximum stroke is 43.5 mm, that means 87% of allowable stroke is used. Active phase of the process begins at stroke of app. 22 mm, that is at about 44% of allowable stroke.

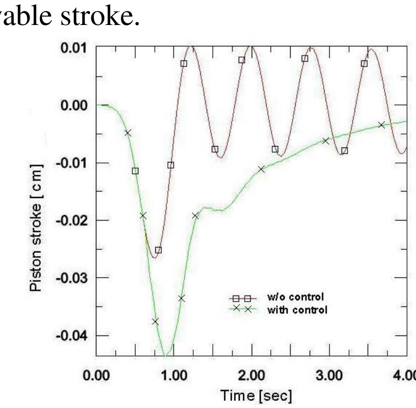


Figure 4: Piston rod displacement

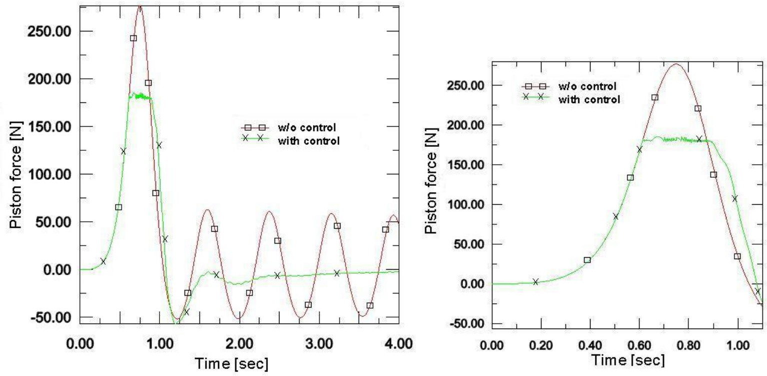
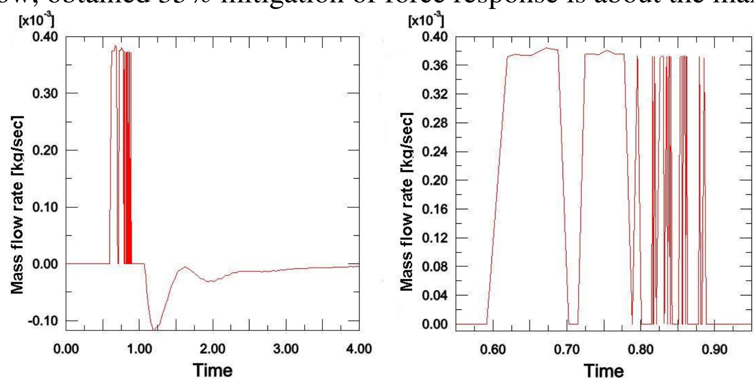


Figure 5: Dynamic force in pneumatic piston

Peak force for the reference case (w/o control) is 277 N and the critical value for the control procedure was set at 180 N. Maximum value obtained during simulation was 185 N, which means that peak dynamic force was mitigated by 35% (cf. Fig 5). If smaller piston force was chosen as critical, or the rising time of the blow was shorter, then the maximum force obtained during simulation would significantly exceed the critical value. Therefore, for given rising time of blow, obtained 35% mitigation of force response is about the maximum gain.



(a) (b)

Figure 6: Mass flow rate m_f *through the valve*

a) whole time domain, b) time window 0.5 – 1.0 sec

Zero value on Figure 6 indicates valve closed, whereas positive values correspond to flow out of the cylinder and negative values – flow into the cylinder. It can be seen that during the peak load valve has to stay open to prevent force from rising, and after the biggest load intensity the valve is opened in a sequence of short impulses (cf. Fig. 6b). In present considerations it is assumed that opening/closing of the valve is instantaneous. This is a reliable assumption since the response time of the piezo valve is of order of 1 ms.

The total mass flow calculated as:

$$mf_{TOT}(\tau) = \int_0^\tau mf(t)dt \,, \tag{4}$$

where *mf(t)* is the mass flow rate, is depicted on Figure 7. One can observe that final value is not equal to zero, which means that the amount of air in cylinder at the end of simulation is not equal to the initial amount. Since total mass flow asymptotically approaches zero, the time required for the system to return to its initial state would be too long.

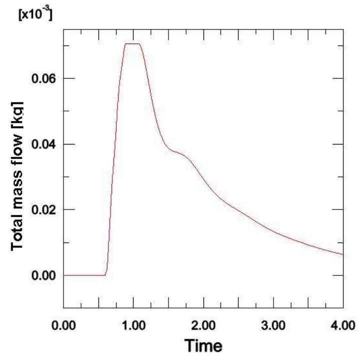


Figure 7: Total mass flow through the valve

4 EXPERIMENTAL SET-UP

System shown on Fig.1 has been manufactured in order to verify numerical results. Air outlet from the cylinder was connected to on/off valve (cf. Fig. 8). Control of the valve opening was obtained by means of Cedrat Technologies piezo actuator. Its integral part is mechanical amplifier of displacements which enables to obtain displacement of 0.2 mm. The response time of such piezo actuated valve is less than 1 ms. Initial tests indicated that blocking force of the actuator is enough to withstand forces of magnitude considered in the numerical simulations.

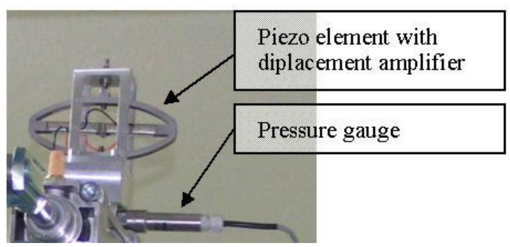


Figure 8: Cedrat Technologies piezo actuator as element of controllable valve

In order to overcome the problem of too slow rate of flow back into the cylinder (cf Fig. 7) system was additionally equipped with air reservoir connected with the cylinder outlet. During the rising phase of load air from cylinder is controllably pressed to the accumulator and returned into the cylinder during the decay phase of loading. This forms a closed system which can work in a sequence. General view of the experimental set-up is shown on Figure 9.

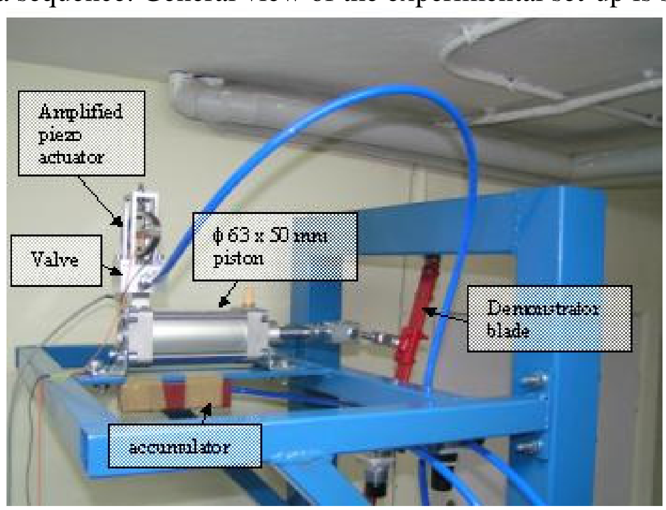


Figure 9: Experimental set-up

The feed-back was established between the pressure transducer monitoring the pressure/force in the cylinder and the piezo actuator opening/closing the orifice. Set-up was integrated using the Bruel&Kjaer PULSE system.

5 CONCLUSION AND FURTHER STEPS

Numerical study presented shows that controlling the pressure in a pneumatic piston subjected to dynamic loading by means of fast response time controllable on/off valve can be an efficient way to reduce the peak dynamic force in the support. For the studied case the gain was 35% compared to the reference case. Presented approach is currently being verified experimentally with use of piezo actuated valve. In the first step closed loop is established between pressure gauge and the opening/closing of valve. In the next step control variable will be a function of signal from the Bruel&Kjaer 4948 surface microphone monitoring the blow intensity from a wind generator.

8 AKNOWLEDGEMENTS

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- **SAFE-STRUCT** *New methods for design of safe structures with identification of hazards and active adaptation in critical states.*

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