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EXPERIMENTS ON THE ACTIVE CONTROL OF BROADBAND SOUND RADIATION FROM A LIGHTWEIGHT PARTITION

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1. INTRODUCTION

For many aircraft, a dominant source of interior noise at cruise is the noise generated by the turbulent boundary layer on the skin of the aircraft. The low frequency component of this noise is difficult to deal with by using passive sound insulation measures without adding considerably to the mass of the aircraft. It has been suggested that the feedback control of the fuselage vibration could be used to reduce the transmission of this noise into the aircraft interior [1]. Vibration sensors on the fuselage skin could be used to provide the signals which, when passed through an appropriately designed compensator, could provide the inputs for piezoceramic actuators bonded to the fuselage or inertial actuators attached to the fuselage skin. An alternative to using piezoceramic actuators or inertial actuators has also been suggested [2]. In this alternative approach one utilizes actuators placed between the fuselage and stiff lightweight trim panels. If the motion of the panel is restricted by its mounting to be predominatly along an axis perpendicular to its surface, then at low frequencies (below the first bending mode frequency of the panel), a reduction in the velocity of the panel produces a reduction in the sound radiated from the panel. This is not the case for a flexible panel (i.e., one which has its first bending mode somewhere in the frequency range of interest) where a reduction in the vibration of the panel does not necessarily produce a reduction in the sound power radiated [3, 4]. In this work, the results are presented of some preliminary experiments which have been used to evaluate the possibility of reducing broadband transmission through feedback control of the motion of a pair of stiff lightweight panels. It is concluded that the approach shows considerable promise.

2. DISCRETE TIME LQG CONTROL

Optimal reductions in the panel velocity can be obtained by first estimating an ARMAX model [5] of the system: i.e., a model of the form

$$A(q^{-1})y(t) = q^{-k}B(q^{-1})u(t) + C(q^{-1})w(t), \tag{1}$$

where q^{-1} is the delay operator, $y(t)$ is the system output, $u(t)$ is the control input and $w(t)$ is a Gaussian white noise sequence. Given such a model, the plant (panel dynamics,

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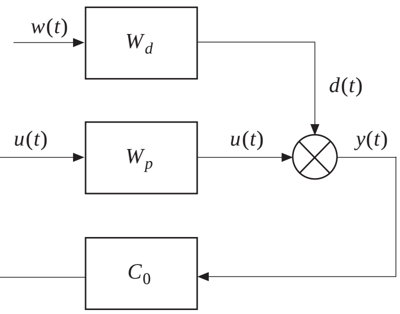

Figure 1. A block diagram of the feedback control problem.

actuator dynamics and dynamics of the ADC and DAC) is represented by the rational z-transform *

$$W_p = z^{-k}B(z^{-1})/A(z^{-1}), \tag{2}$$

and the disturbance (accelerometer output due to the Ling shaker input) is modelled as Gaussian white noise passed through the shaping filter with the rational z-transform

$$W_d = C(z^{-1})/A(z^{-1}). \tag{3}$$

Given such a model for the system the feedback control problem can be illustrated by the block diagram given in Figure 1. An appropriate cost function to minimize is the linear quadratic Gaussian cost which for a time invariant plant and stationary disturbance can be written as

$$J(u) = \mathrm{E}[\, y^2(t) + \beta u^2(t)] \tag{4}$$

where E is the expectation operator and β is a scalar constant. It can be shown [6, 7] that, in the case in which the plant is stable, a stabilizing compensator that minimizes the cost given by equation (4) can be found by obtaining the minimal degree solution of the Diophantine equation

$$D_c(q)G(q^{-1}) + q^g F(q^{-1})A(q^{-1}) = q^k B(q)D_f(q^{-1}), \tag{5}$$

where $G(q^{-1})$ and $F(q^{-1})$ are the unknowns, $g = \max\,(n_{dc}, n_b \ldots, n_b + k)$ and $D_c(q^{-1})$ and $D_f(q^{-1})$ are spectral factors given by

$$D_f(q^{-1})D_f(q) = Q_w C(q^{-1})C(q) + Q_v A(q^{-1})A(q), \tag{6}$$

$$D_c(q)D_c(q^{-1}) = B(q)B(q^{-1}) + \beta A(q)A(q^{-1}). \tag{7}$$

Note that a function of the form $B(q)$, for example, simply implies the mirror image "anti-casual" counterpart of the function $B(q^{-1})$. The scalars Q_w and Q_v are the variances of $w(t)$ and the measurement noise respectively. Given a minimal degree solution of equation (5), the optimal compensator is given by

$$C_0(q^{-1}) = \frac{-G(q^{-1})A(q^{-1})}{D_c(q^{-1})D_f(q^{-1}) - q^{-k}G(q^{-1})B(q^{-1})}. \tag{8}$$

This form of solution of the LQG control problem was originally suggested by Kučera [8] and was discussed in some detail by Grimble and Johnson [6] and, in the context of the active control of sound and vibration, by Thomas and Nelson [7].

* The delay operator (q^{-1}) and the complex variable z^{-1} are used here interchangeably.

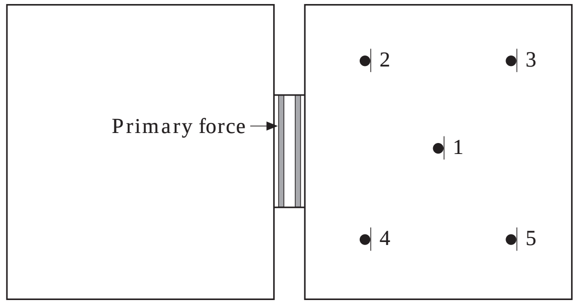

Figure 2. A plan view of the reverberation suite showing the five microphone positions.

The experiment described here was intended as a preliminary test of the transmission characteristics of the active double panel arrangement shown below in Figure 3, in which the compensator was designed using the approach outlined above.

3. EXPERIMENTAL DETAILS

The experiment was carried out in a small reverberation suite, which is illustrated diagrammatically in Figure 2. A double partition consisting of two aluminium honeycomb panels (dimensions 340 mm by 460 mm by 26·6 mm thick) mounted on flexible gaskets in a wooden frame was sealed into the aperture between the two chambers. The double partition is illustrated in Figure 3 together with the feedback control arrangement used in the experiments. A modified loudspeaker unit was used to provide a control force between the two panels. The control actuator was mounted on an axis passing through

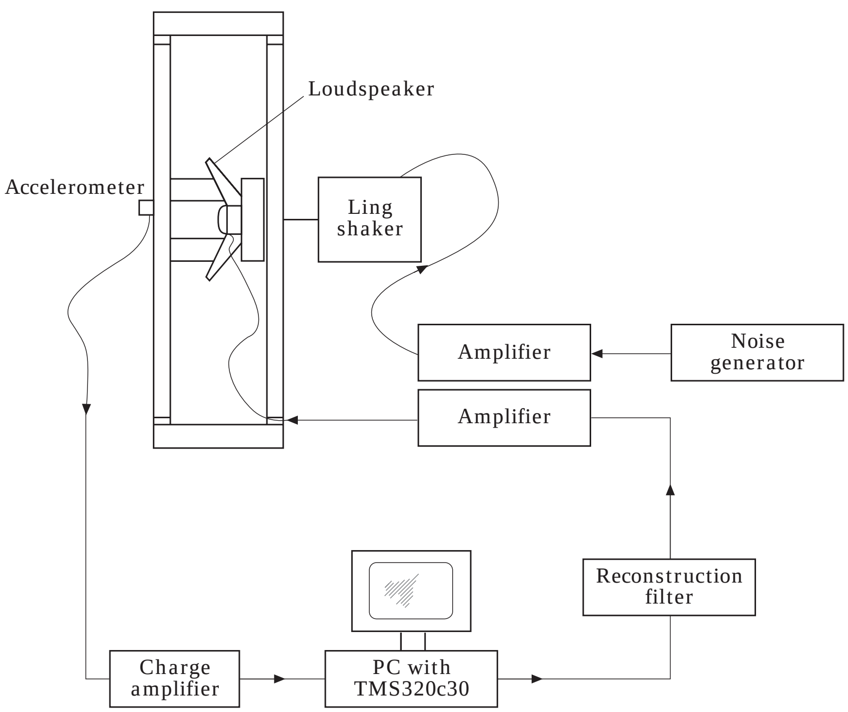

Figure 3. A diagram of the double panel and feedback control system.

the centres of the two panels. The signal from an accelerometer mounted at the centre of the receiving room panel is fed, via a charge amplifier to the input of a digital signal processing card incorporating a Texas instruments TMS320c30 processor. A feedback compensator is implemented on the DSP card as an infinite impulse response filter. The output of the card is fed via a reconstruction filter and necessary amplification to the input of the control actuator. The panel in the source room was excited by shaker attached at an off-centre position. Initial attempts to use an acoustic primary source were unsuccessful, since the transmission loss of the partition in the absence of control was sufficiently large to make the output of the accelerometer too low to be used as the control signal.

In order for an optimal compensator to be designed, the $A(q^{-1})$, $B(q^{-1})$ and $C(q^{-1})$ polynomials in equation (1) need to be identified. Acceleration data was obtained with the primary source in operation and the loop open. This data was used to form an ARMA model of the disturbance with a 19th order numerator and a 20th order denominator. This was achieved by using the ARMAX function in the MATLAB System Identification Toolbox, which uses an iterative Gauss–Newton algorithm to minimize a quadratic prediction error criterion [5]. The power spectrum which results from driving the disturbance model with Gaussian white noise is compared with the measured disturbance in Figure 4. A Gaussian white noise signal was then passed through an all-pass filter implemented on the DSP card, the output of which was made to drive the reconstruction filter and hence the control actuator. Input white noise data and acceleration data were simultaneously acquired and from this data an auto-regressive model of the plate (i.e., the transmission path from the DSP input to the charge amplifier output) was identified by using the ARX function in the MATLAB System Identification Toolbox (ARX treats the least squares estimation problem as an overdetermined set of linear equations). This model has a 15th order numerator and 15th order denominator. The plant and disturbance models were combined to give an ARMAX model of the system. The frequency response of the plant model is compared with the measured frequency response of the plant in Figure 5. The polynomials that form this model were used in equations (5)–(8) to obtain a discrete time compensator that would minimize equation (4) with $\beta = 0{\cdot}0003$. The

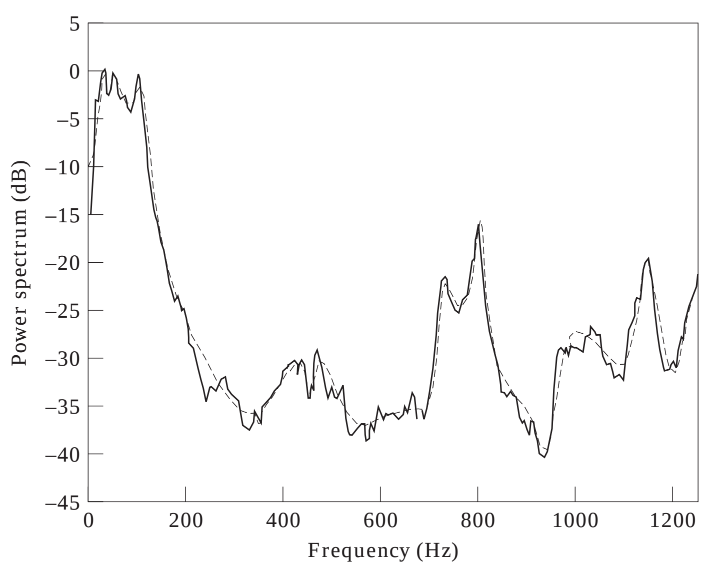

Figure 4. A comparison of the power spectrum of the output of the disturbance mode (---) when driven by white noise with the measured disturbance power spectra (——).

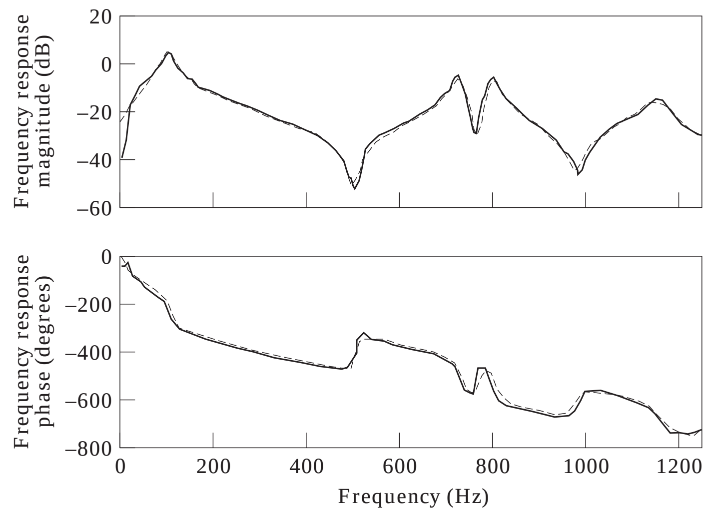

Figure 5. A comparison of the frequency response of the plant model (---) with the measured plant frequency response (——).

spectral factors given by equations (6) and (7) were obtained by first finding the roots of the polynomial on the right-hand side of each equation by using the MATLAB function ROOTS, and then taking the stable roots to be those of the spectral factor and using the POLY function to obtain the spectral factor polynomial from the roots. The variances Q_w and Q_v appearing in equation (6) were set to 1 and 0·01 respectively. The frequency response of the compensator which was used in the experiment is shown in Figure 6.

The results obtained by using this compensator can be seen in Figure 7, which shows the sum of power spectra at the five microphone locations with and without the control loop closed. In Figure 8 are shown the open and closed loop responses up to 300 Hz, and from this it can be seen that there is a significant reduction in transmission in this region. The piston-like rigid body mode dominates the radiation for a large part of this

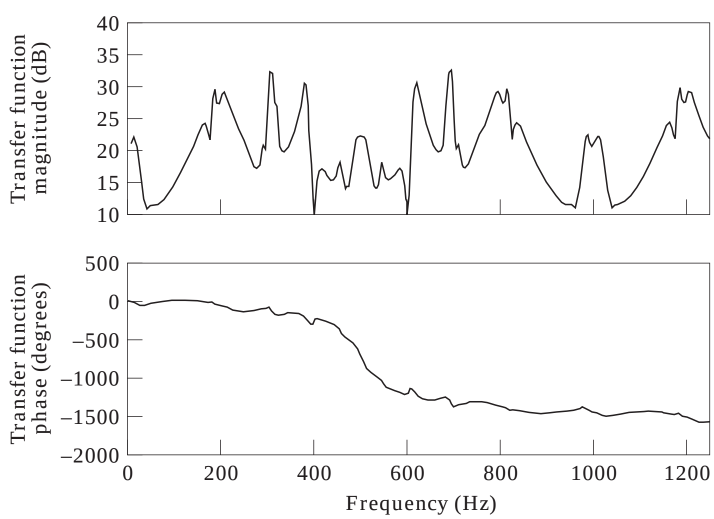

Figure 6. The frequency response of the compensator used.

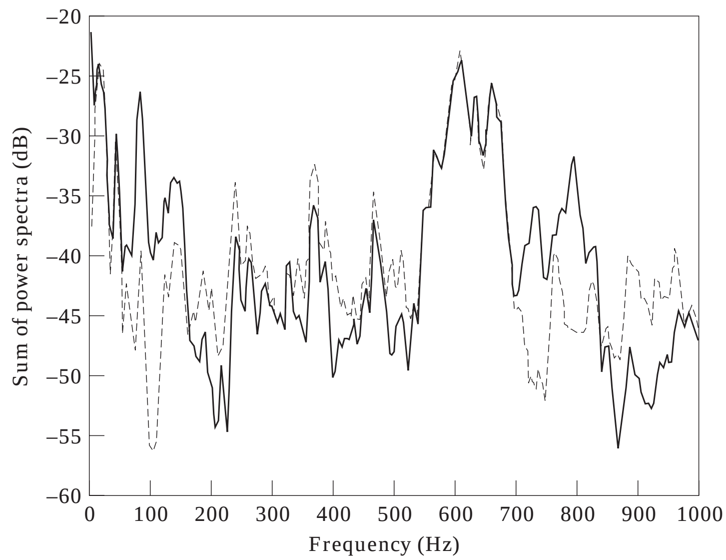

Figure 7. The sum of the power spectra of the microphone outputs with the feedback loop open (———) and the feedback loop closed (- - -) with $\beta = 0{\cdot}0003$.

region. Above 150 Hz there is a frequency region in which the feedback control of the acceleration at the centre of the panel actually increases the sum of the power spectra at the microphones. Measurement of the acceleration at different points on the receiving room panel indicated that the rotational rigid body modes have a resonance in this frequency region. Uneven stiffness in the mounting gasket (required to seal the internal air cavity) is likely to have caused the control force to be coupled into these modes which are weakly excited by the Ling shaker. Measurement at a number of points on the panel demonstrated that at around 550 Hz the bending modes of the panels become significant

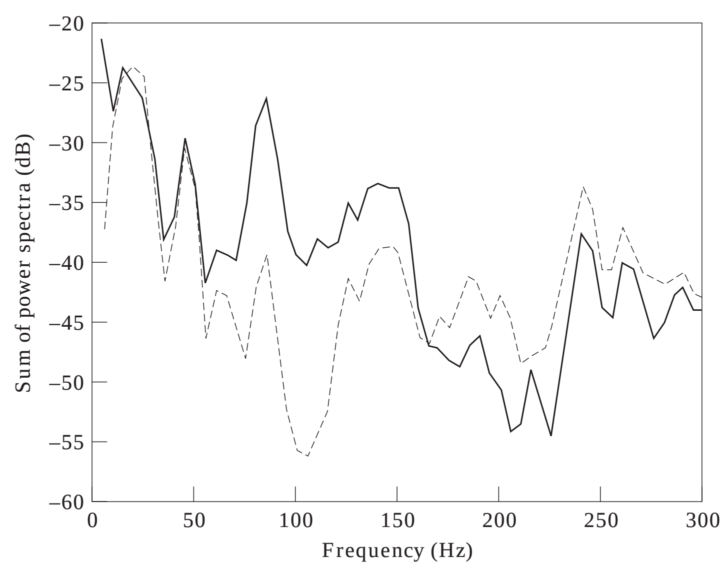

Figure 8. As Figure 7, but with $\beta = 0{\cdot}0003$ for 0–300 Hz.

and it can be seen from Figure 7 that above this frequency feedback control has variable effects depending upon which modes dominate radiation at any particular frequency.

To illustrate the effectiveness of the controller in minimizing the variance of the controlled variable at the centre of the panel, in Figure 9 is shown the acceleration at the centre of the receiving room panel with and without feedback loop closed.

When designing a feedback controller it is not sufficient to assess simply the performance of the system. It is also necessary to assess the robustness of the system. Of particular interest in this case is the robustness of the controller to changes in the plant or errors in the plant model used to design the compensator. One analytical technique developed for H_∞ control which can be used to assess robustness to plant changes requires a model for the uncertainties in the plant. It has been suggested [9] that an appropriate model for the uncertainties in an acoustic plant is the unstructured multiplicative output uncertainty model in which the plant is modelled as

$$W(\mathrm{j}\omega) = (1 + \Delta W(\mathrm{j}\omega))W_0(\mathrm{j}\omega), \tag{9}$$

where $W_0(\mathrm{j}\omega)$ is the nominal plant response and $\Delta W(\mathrm{j}\omega)$ is the fractional uncertainty in the plant. If the fractional uncertainty has an upper bound, $B(\omega)$, then robust stability is assured if the following expression is satisfied:

$$\| T_0(\mathrm{j}\omega)B(\omega) \|_\infty < 1, \tag{10}$$

where $T_0(\mathrm{j}\omega)$ is the complementary sensitivity function for the nominal plant. In this case the complementary sensitivity function is given by

$$T_0(z^{-1}) = z^{-k}G(z^{-1})B(z^{-1})/D_c(z^{-1})D_f(z^{-1}), \tag{11}$$

where $D_e(z^{-1})$ and $D_f(z^{-1})$ are given by equations (7) and (6) and $G(z^{-1})$ is found by solving equation (5). In Figure 10 is shown the magnitude of the complementary sensitivity for the compensator used in the experiment presented here. The maximum value of $|T_0(\mathrm{j}\omega)|$ is 2·05 which, from equation (10), gives an upper bound on fractional plant uncertainty

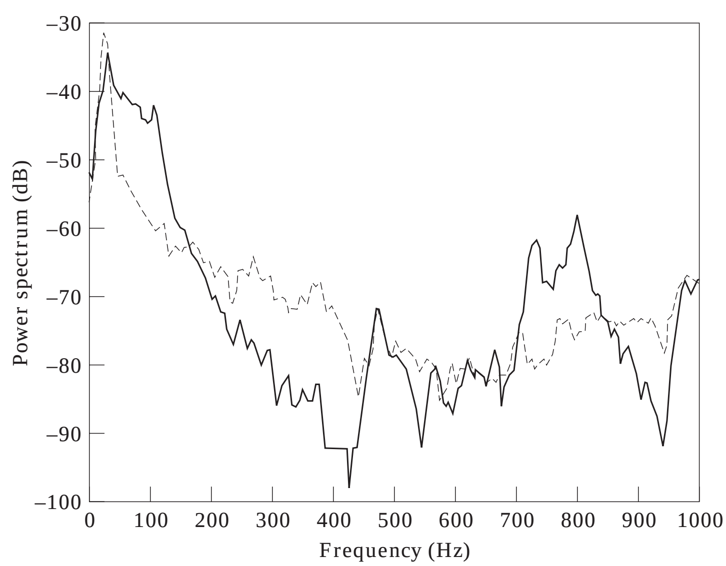

Figure 9. The power spectrum of the acceleration of the centre of the receiving room panel with the feedback loop open (——) and the feedback loop closed (---) with $\beta = 0{\cdot}0003$.

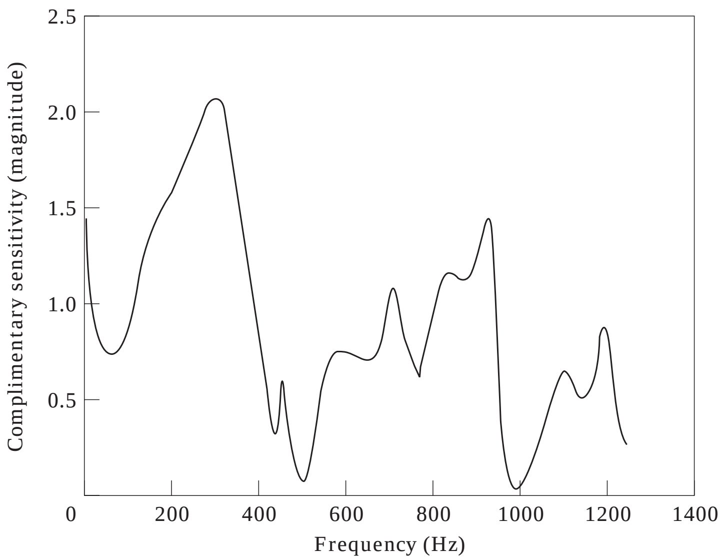

Figure 10. The magnitude of the complementary sensitivity function for the closed loop system.

of approximately 49%, which corresponds to an error in the plant model magnitude of 3·46 dB.

4. CONCLUSIONS

The experiments presented here have demonstrated that feedback control of broadband structural acoustic transmission through a double partition incorporating a control actuator can yield reductions in excess of 10 dB in averaged sound pressure over a useful frequency range. If excitation of the rotational rigid body modes can be prevented this frequency range could be extended up to the natural frequency associated with the first bending mode of panels.

REFERENCES

1. D. R. THOMAS and P. A. NELSON 1995 *Journal of the Acoustical Society of America* **98**, 2651–2662. Feedback control of sound radiation from a plate excited by a turbulent boundary layer.
2. D. R. THOMAS and P. A. NELSON 1994 *Proceedings of Internoise* '94, 1283–1286. Feedback control of sound transmission through stiff lightweight paritions.
3. C. R. FULLER, R. J. SILCOX, V. L. METCALF and D. E. BROWN 1992 *Journal of Sound and Vibration* **153**, 387–402. Active control of sound transmission/radiation from elastic plates by vibration inputs, II: Experiments.
4. D. R. THOMAS, P. A. NELSON and S. J. ELLIOTT 1995 *Journal of Sound and Vibration* **181**, 515–539. An analytical investigation of active control of the transmission of sound through plates.
5. L. LJUNG 1987 *System Identification—Theory for the User*. Englewood Cliffs, NJ: Prentice-Hall.
6. M. J. GRIMBLE and M. A. JOHNSON 1988 *Optimal Control and Stochastic Estimation*: *Theory and Applications*, *Volume*. 2 Chichester: John Wiley.
7. D. R. THOMAS and P. A. NELSON 1995 ISVT *Technical Memorandum* 754. A comparison of some strategies for discrete time feedback control.
8. V. KUČERA 1980 *IEEE Transactions on Automatic Control* **25**, 913–919. Stochastic multivariable control: a polynomial equation approach.
9. S. J. ELLIOTT 1993 *ISVR Technical Memorandum* 732. Active control using feedback.