Accuracy and Scalability of MARS-KS Modeling for Swing Check Valve Behavior

Young Seok Bang [a*], Ju Yeop Park [a], Yong Suk Choi [a]
[a]*Korea Institute of Nuclear Safety, 62 Kwahak-ro, Yuseong, Daejeon, Korea*
**Corresponding author: k164bys@kins.re.kr*

Keywords : swing check valve, hydraulic torque coefficients, similarity, MARS-KS code

1. Introduction

Swing check valves are components that play an important role in establishing the direction of flow in the piping network of both active and passive safety systems [1]. In the check valve, the opening angle and the resulting through flow rate are determined by the balance of torques due to fluid momentum and the weight of the valve disk and the torque due to disk movement, so fluid-structure interaction is very important. Especially, in a passive safety system, the driving force required for flowing decreases over a long period of time after the start of an accident or transition [2], so it is necessary to carefully monitor the behavior of the flow decreasing as the check valve start to close.

In general, system thermal-hydraulic codes such as RELAP5 [3], MARS-KS [4], and TRACE [5] which are commonly used for thermal-hydraulic analysis of accidents or transients, have models describing check valves. In the previous study [6], the authors have assessed the experiment simulating check valve closure using the built-in check valve model of the MARS-KS code and the model proposed by Li [7] with the author's modeling scheme. According to the authors' experience, although the Li-proposed theory-based model clearly gives better prediction results than the built-in model, the accuracy of the prediction results is still dependent on stationary and rotational hydraulic torque coefficients, and determining those coefficients is difficult.

In the present study, it is discussed the authors' improvements of the accuracy of the modeling scheme since their previous study by showing the effect of hydraulic torque coefficients on prediction results. It also discusses the scalability of the known hydraulic torque coefficients to be applied to the check valves in big size of actual pipe networks.

2. Calculation Model

2.1 Basic Model

The check valve model presented by Li [7] and used in this study is based on the disk opening angle obtained through the solution of the following equation:

$$I\frac{d^2\theta}{dt^2} = -T_W + T_{HS} \mp T_{HR} + T_F \qquad (1)$$

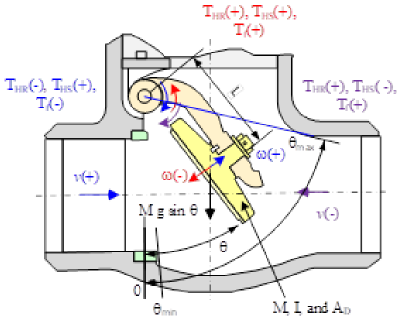

Fig. 1. Swing check valve model and notation

The right side of this equation refers to a torque due to the weight of the disk, a torque due to fluid flow, a torque due to rotational movement of the disk, and a torque due to friction of the valve shaft, respectively.

$$T_W = MgL\sin\theta\,, T_{HS} = C_{HS}\rho A_D L\frac{v^2}{2}, \\ T_{HR} = C_{HR}\rho A_D L\frac{(L\dot{\theta})^2}{2}, T_F = a - b\dot{\theta} \qquad (2)$$

In the equations above, θ means angle of disk ranging from θ_{min} to θ_{max}, $\dot{\theta}$ means angular speed of the disk, M, I, ρ, A_D, L mean mass of the disk and the arm (kg), moment of inertia of disk (kgm^2), through flow velocity (m/sec), area of disk (m^2), length of the arm (m), respectively. C_{HS}, C_{HR}, a, b mean stationary and rotational hydraulic torque coefficients, static and dynamic frictional torque coefficients. Of the double signs of each term in the equation, the above is for forward motion and the lower is for backward motion of the disk.

Those equations are modeled in the input using the control variables of the MARS-KS code and solved over time. That is, the through velocity of fluid, v, which is determined through the MARS-KS calculation from the given initial condition is used. Here, when the check valve is opened or closed, the directions of the frictional, stationary and rotating hydraulic torques should be appropriately considered. In addition, the variation of valve opening area depending on the angle of the disk is calculated using the following curve which was obtained through a complex integration process [8]. In this figure, the two curves presented by Turesson [9] and Lim [8] are compared. One by Turesson was used in the previous study, which is different from the projected area variation for the movement of the disk derived by Lim's study [8].

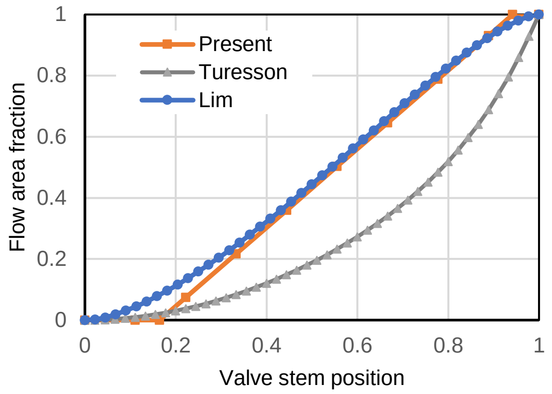

Fig. 2. Valve Stem position vs flow area fraction

Thus, a simplified curve based on Lim's results was used in the present study. The flow area fraction determined in this way is incorporated into the 'abrupt area change model' of the MARS-KS code to determine the loss coefficient. Changes in pressure drop and flow rate calculated with this loss coefficient are incorporated into the subsequent calculations and repeated until convergence.

2.2 Torque coefficients

The frictional torque coefficients were set as follows, based on Li's research result.

$$a - b\dot{\theta} = \begin{Bmatrix} 0.018 - 0.003\dot{\theta} \quad if \;\; \dot{\theta} < 0 \\ -0.018 - 0.003\dot{\theta} \quad if \;\; \dot{\theta} > 0 \end{Bmatrix} \tag{3}$$

The stationary hydraulic torque coefficient is a function of disk angle and through velocity. Li determined the stationary torque coefficient for various angles of the disk through several steady state experiments. Boqvist [10] showed that this coefficient can be determined through CFD (computational fluid dynamics) calculations. In the present study, adjustment of the coefficients was made at the disk angle of 1.0 rad or higher to match the disk opening angle in a steady state based on the experimental results of Li [7].

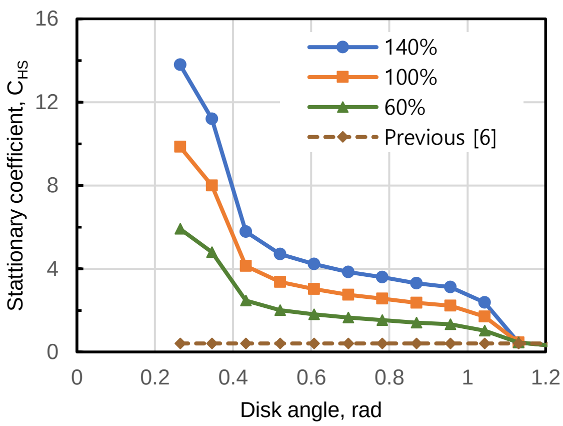

Fig. 3. Stationary hydraulic torque coefficient

Figure 3 shows the stationary hydraulic torque coefficient determined by those process as a function of the opening angle. The curves for the upper bound and the lower bound of this C_{HS} curve having ±40% margin are shown in the figure. The curve used in the previous study is also shown.

The rotational hydraulic torque coefficient is a function of disk angle, angular velocity, and through flow velocity. Li presented the rotational hydraulic torque coefficients in the range of several fluid velocities and disk angles through complicated process using the steady-state torque coefficients. However, as Boqvist pointed out [10], it was difficult to use those data in RELAP5 code. In the present study, among the curves of the rotational coefficient determined by Boqvist through CFD analysis, one close to the velocity condition of the present study was used. Figure 4 shows the rotational torque coefficient curve used in the present study for the several open angles. This figure compares the curves of upper bound and the lower bound to be used for the following sensitivity analysis and the curve used in the previous study.

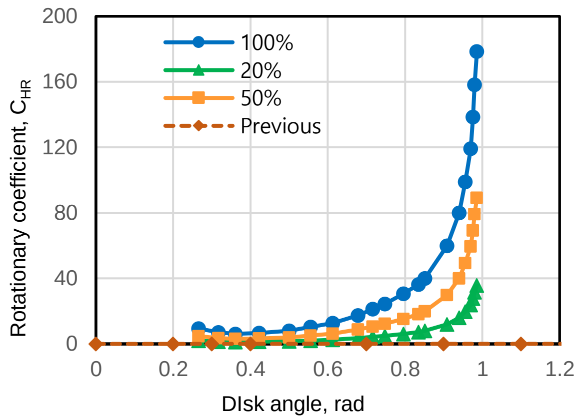

Fig. 4. Rotational hydraulic torque coefficients

2.3 Scalability

Equation (1) is a general form and is independent of the shape and size of the check valve. However, the torque coefficients included in the equation can vary depending on the shape and size of the check valve. In the present study, if the shape, size, and the torque coefficients are known, it is evaluated whether they can be applied to check valves of similar shape and different size. To this end, non-dimensional variables are introduced and non-dimensional equations is derived as follows. The non-dimensional form of the similar torque-inertial equation has been attempted in the literature [11]

$$\begin{gathered} I^* = I/I_R\,, \theta^* = \theta/\theta_R = \theta/(\pi/2)\,, t^* = t/(L_R/v_R), \\ M^* = M/M_R\,, L^* = L/L_R\,, \rho^* = \rho/\rho_R, \\ {A_D}^* = A_D/{A_D}_R\,, v^* = v/v_R\,, a^* = a/a_R, \\ b^* = b/b_R\,, {C_{HS}}^* = C_{HS}/{C_{HS}}_R\,, {C_{HR}}^* = C_{HR}/{C_{HR}}_R \end{gathered} \tag{4}$$

In the equation, subscript R means reference state, i.e. known condition. The reference states of disk angle and time were set $\pi/2$ and L_R/v_R, respectively. By substituting these definition equations into equation (1) and organizing them, the following non-dimensional torque-inertia equation can be obtained.

$$I^*\frac{d^2\theta^*}{dt^{*2}} = -\psi_{1R}M^*L^*\sin\left(\frac{\pi}{2}\theta^*\right) + \psi_{2R}C_{HS}{}^*\rho^*A_D{}^*L^*\frac{v^{*2}}{2} - \psi_{3R}C_{HR}{}^*\rho^*A_D{}^*L^*\left\{\frac{d(L^*\theta^*)}{dt^*}\right\}^2 - \psi_{4R}a^* - \psi_{5R}b^*\frac{d\theta^*}{dt^*} \quad (5)$$

In this equation, the definitions of five non-dimensional numbers on the right side are as follows:

$$\psi_{1R} = \frac{2M_RL_R{}^3g}{\pi I_Rv_R{}^2}$$
$$\psi_{2R} = \frac{2C_{HS_R}\rho_RA_{D_R}L_R{}^3}{\pi I_R}$$
$$\psi_{3R} = \frac{\pi C_{HR_R}\rho_RA_{D_R}L_R{}^3}{4I_R} \quad (6)$$
$$\psi_{4R} = \frac{2a_R}{\pi I_R}\left(\frac{L_R}{v_R}\right)^2$$
$$\psi_{5R} = \frac{b_R}{I_R}\frac{L_R}{v_R}$$

Here, subscript R means that the non-dimensional number is expressed as a value of the reference state. The meaning of this equation is that the torque-inertia equation for the desired state (T) can be expressed in five non-dimensional numbers and non-dimensional variables, and that the same non-dimensional solution can be obtained if these five non-dimensional numbers are the same for both the reference state and the desired state. From the condition of the same non-dimensional numbers,

$$\{\psi_{1R}, \psi_{2R}, \psi_{3R}, \psi_{4R}, \psi_{5R}\} = \{\psi_{1T}, \psi_{2T}, \psi_{3T}, \psi_{4T}, \psi_{5T}\} \quad (7)$$

Now, consider a check valve in the same shape as and different size from the known check valve. Assume the diameter of the pipe to be λ times, the mass of disk λ^3 times, the length of the arm λ times, and moment of inertia λ^5 times of the reference state.

$$\{A_{D_t}, L_t, M_t, I_t\} = \{\lambda^2A_{D_R}, \lambda L_R, \lambda^3M_R, \lambda^5I_R\} \quad (8)$$

By inserting those equations into equation (5), we can obtain the following result.

$$\{v_t, C_{HS_t}, C_{HR_t}, a_t, b_t\} = \{\lambda^{0.5}v_R, C_{HS_R}, C_{HR_R}, \lambda^4a_R, \lambda^{4.5}b_R\} \quad (9)$$

This result determines the relations between the desired state and the reference state regarding through flow velocity, friction torque coefficient, and hydraulic torque coefficient as necessary conditions for ensuring similarity between the behavior of the R-check valve and the behavior of the T-check valve. In this case, similarity is ensured only when the hydraulic torque coefficients of the reference state are the same as them of the desired state, so the hydraulic torque coefficients of the reference state can be applied as they are.

However, in general, when the pipe size increases, it is difficult to expect that the check valve of the increased size satisfies those conditions. In such case, it is required to obtain torque coefficients through specific experiment or CFD analysis. However, when referring to various literatures, the hydraulic torque coefficient can be regarded to have a generally similar shape, so it may be possible to apply upper and lower curves with sufficient margin based on known reference conditions.

3. Results and Discussion

3.1 Base Calculation

The aforementioned modeling was applied to Li's check valve closure experiment. The check valve was described as a servo valve component, and the modeling of calculating various torques using control variables is the same as in the previous study. The data described above is implemented as hydraulic torque coefficients. The experimental loop and MARS-KS noding are shown in Figures 5 and 6.

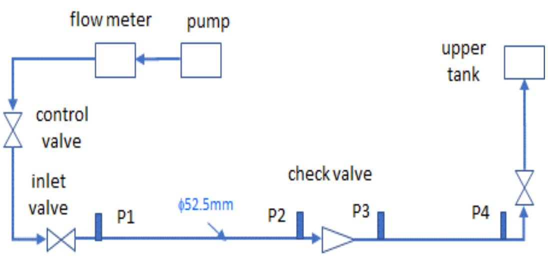

Fig. 5 Test loop for swing check valve test

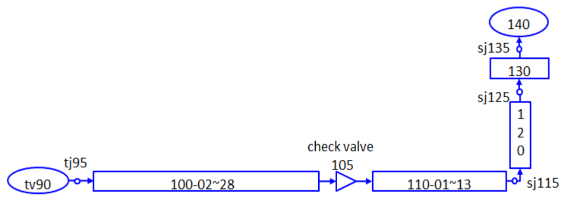

Fig. 6. MARS-KS nodalization of the test loop

Figure 7 shows the disk angle obtained as a result of the base calculation compared to the experimental data. The calculation result of the previous study was also compared. As shown in the figure, the calculation results of the present study are well matched to the experimental data. Those improved results are attributed to implementing the stationary hydraulic torque coefficient and the rotational hydraulic torque coefficient.

Valve reopening after 0.8 seconds was not predicted. It seems to be due to the failure to consider the vertical pipe upstream of the check valve of the experimental facility.

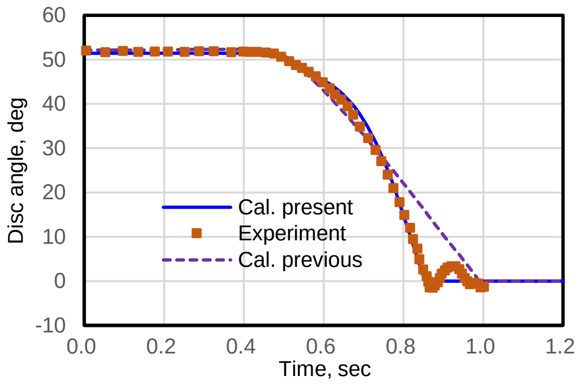

Fig. 7 Comparison of disk angle of base case

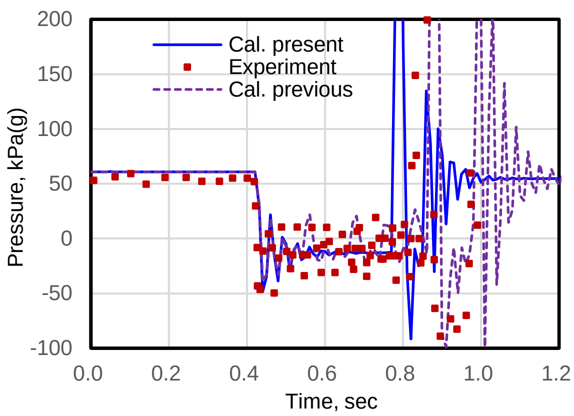

Fig. 8 Comparison of downstream pressure of base case

Figure 8 shows the calculated pressure behavior at the downstream of the check valve. Since the present calculation indicated the check valve closing time closer to the experiment than the previous study calculation, the pressure increase caused by the water hammer wave is also occurring earlier.

3.2 Sensitivity study

In the present study, it was evaluated how the closing behavior of the valve changes when the upper and lower bounds of the torque coefficients presented above are applied.

Figures 9 and 10 compare the difference in the behavior of the valve opening angle according to the change in the stationary hydraulic torque coefficient and the rotational hydraulic torque coefficient, respectively. As can be shown from those figures, as the torque coefficients increase, the time taken for full closure increases by up to 20%. Therefore, if the upper and lower bounds of the hydraulic torque coefficients obtained from existing researches are obtained and applied, the reliability of closing behavior prediction can be further ensured

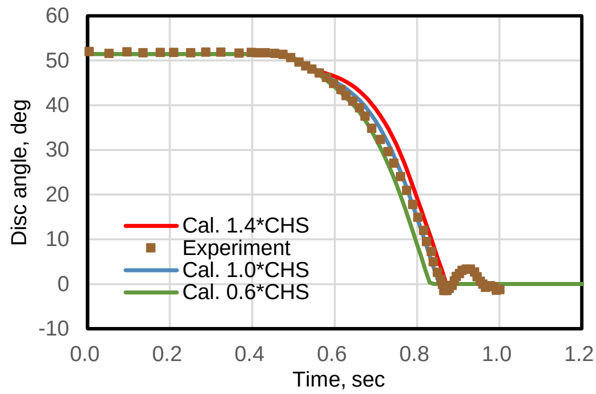

Fig. 9. Comparison of disk angle for changing stationary torque coefficient

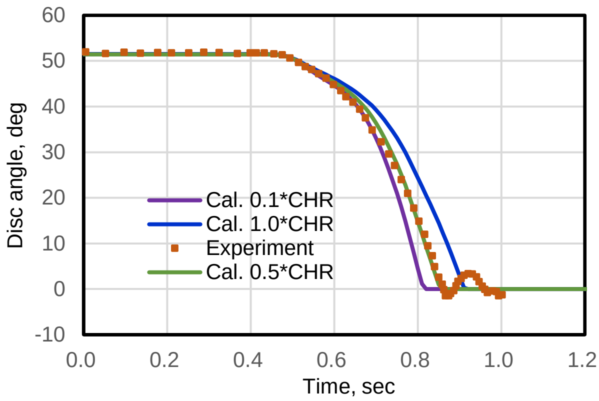

Fig. 10. Comparison of disk angle for changing rotational torque coefficient

3.3 Evaluation of scalability

As mentioned in the previous section, it was investigated whether the hydraulic torque coefficients used in the calculation of the 2-inch check valve in the present study could be applied to the bigger check valves. Figures 11 and 12 show the calculation results for the case where the diameter (2 in.) of the valve disk is doubled, tripled, and quadrupled ($\lambda = 2, 3, 4$).

Figure 10 shows that the four valves satisfy the similarity conditions, but the larger the valve, the longer the full closing time. Figure 11 shows this behavior in a dimensionless form for both time and opening angle. It can be found that all valves exhibit one closing behavior within a range of about 6.7% (based on 15 dimensionless time). The lack of complete similarity may be considered to be the effect of the difference in pipe frictional pressure drop by the change in the diameter of the pipe in the MARS-KS code.

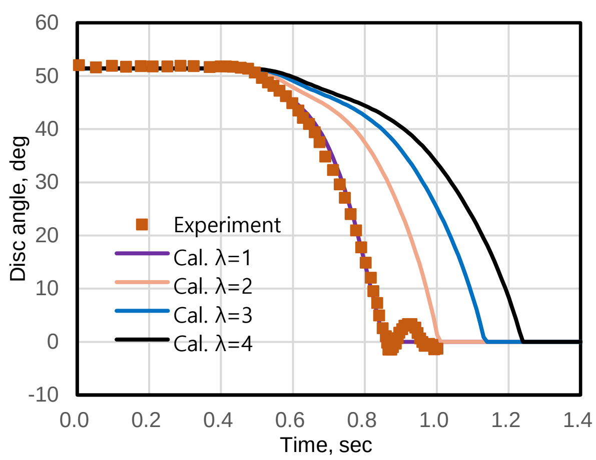

Fig. 11. Comparison of disk angle for different size of check valve

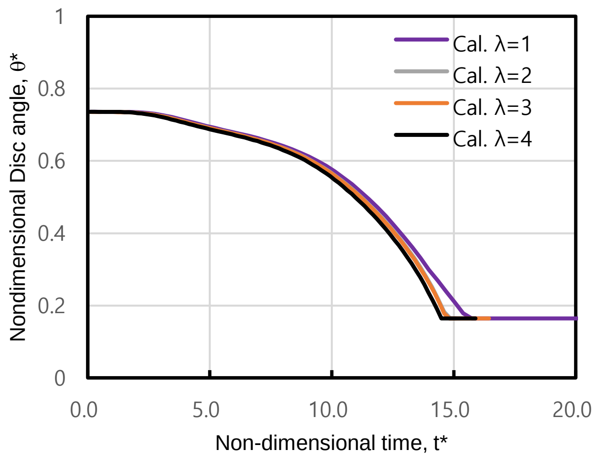

Fig. 12. Comparison of non-dimensional disk angle

4. Conclusions

In the present study, it was focused to show the method of modeling the check valve which is an important factor in evaluating the performance of the passive safety system, with the MARS-KS code, was improved, compared to the previous study. Especially, stationary and rotational hydraulic torque coefficients were determined from the available data and implemented into the MARS-KS code input. In addition, the scalability that the known hydraulic torque coefficient could be applied to the check valve of the actual large flow piping was evaluated. The conclusions obtained are as follows.

1) The stationary hydraulic torque coefficient and the rotational hydraulic torque coefficient applied in this calculation can give results close to the experimental data of Li. In addition, it was confirmed through sensitivity study that those torque coefficients can have a significant effect on the closing behavior of the check valve.
2) The conditions under which known hydraulic torque coefficients can be applied to check valves of different sizes could be determined through similarity analysis with non-dimensional numbers.
3) To analyze the behavior of check valves in large scale, the torque coefficient must be determined through valve-specific experiment or computational fluid dynamics analysis. However, if the upper and lower bounds of the torque coefficients can be found from the available data, application of those bounds can also be considered.

ACKNOWLEDGEMENT

The preparation of this paper was supported by the Nuclear Safety Research Program through the Korea Foundation Of Nuclear Safety (KOFONS), granted financial resource from the Nuclear Safety and Security Commission (NSSC), Republic of Korea (No. 2106002).

REFERENCES

[1] G. Welsford, Crucial Role Of Check Valves in Piping Systems, https://www.piping-world.com › Valves › Check Valve, 2023.
[2] NEA, Survey on the Regulatory Practice to Assess Passive Safety Systems used in New Nuclear Power Plant Designs, First stage report, NEA/CNRA/R(2017)3, 2019.
[3] Information System Lab, REALP5/MOD3 Code Manual, Volume II: Input Requirements, NUREG/CR-5535, 2009.
[4] KINS, MARS-KS Code Manual, Volume II: Input Requirements, KINS/RR-1822, Vol. 2, Daejeon, Korea, 2018.7.
[5] USNRC, Final - TRACE V5.0 Theory Manual - Volume 1: Field Equations, Solution Methods, 2012. and Physical Models, https://www.nrc.gov/docs/ML1200/ML120060218.pdf.
[6] Y. S. Bang et al, Modeling of Swing Check Valve in Support of Prediction of Passive Safety Systems Performance, Transactions of the Korean Nuclear Society Spring Meeting Jeju, Korea, May 18-19, 2023.
[7] G. Li, J. C. P. Liou, Swing Check Valve Characterization and Modeling During Transients, Journal of Fluids Engineering, Vol. 125, p1043, 2003.
[8] H.G. Lim et al, Development of a swing check valve model for a low velocity pipe flow prediction, Nuclear Engineering and Design Vol. 236, p 1051–1060, 2006.
[9] M. Turesson, Dynamic simulation of check valve using CFD and evaluation of check valve model in RELAP5, MS Thesis, Chalmers University of Technology, 2011.
[10] E. Boqvist, Investigation of a Swing Check Valve Using CFD, MS Thesis, Linkopings Univ., Sweden, 2014.
[11] J. Lynch et al, Dimensional analysis of spring-wing systems reveals performance metrics for resonant flapping-wing flight, Journal of The Royal Society Interface, Volume 18, Issue 175, Feb 2021.