EFFECT OF ACOUSTIC RESONANCE ON FLUIDELASTIC INSTABILITY IN NORMAL TRIANGULAR TUBE ARRAYS

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ABSTRACT

The interaction between fluidelastic instability and acoustic resonance in two normal triangular tube arrays has been quantified. Two possibilities are examined which are based on the quasi-steady framework proposed by Price & Paidoussis (1984) to model fluidelastic instability. The effect of acoustic resonance on the steady fluid forces on a static cylinder is examined as well as the effect of acoustic resonance on the time delay between tube motion and the resultant flow reorganisation close to the measurement cylinder.

1. INTRODUCTION

An array of circular tubes subject to fluid cross flow may exhibit large amplitude self-induced vibration known as fluidelastic instability (FEI). The vibration occurs at the natural frequency of the structure and the fluid force depends on tube motion, thus the tube motion itself causes the excitation. The phenomenon has been classified under two distinct mechanisms by Chen (1983): fluid damping; and fluid stiffness controlled instability. This study is concerned with fluid damping controlled instability and it occurs when the net linear damping goes to zero. In gas flows, as the tube bundle is enclosed in a duct, the system may also experience acoustic resonance (AR) at a frequency which is several orders magnitude greater than the natural frequency of the structure. One could reasonably expect that fluidelastic instability and acoustic resonance to be largely independent due to the large separation in frequency. However, Price & Zahn (1991) and Meskell & Fitzpatrick (2003) reported an apparent interaction between the phenomena. The interaction was quantified by Mahon & Meskell (2008). They reported that the imposed acoustic field reduced the vibration amplitude of a single flexible cylinder in a normal triangular tube array (P/d=1.32). The current paper briefly recalls the quantification of the effect of acoustic resonance on fluidelastic instability for P/d=1.32 extending the analysis to also include P/d=1.58 before proceeding on to a more comprehensive analysis of the interaction between the phenomena.

2. EXPERIMENTAL SETUP

The experimental facility consists of a draw down wind tunnel with the tube array under investigation installed in the test section. The configurations under test were two five row normal triangular (NT) tube arrays with pitch ratios of 1.32 and 1.58 subject to air cross flow. The flow velocity in the wind tunnel test section ranged from 2m/s to 14m/s with a free stream turbulence intensity of less than 1%. The tubes in the array are rigidly fixed, except for one tube which will be referred to as the instrumented cylinder. For the vibration tests, the instrumented cylinder (position 1 in Fig. 1) was free to oscillate in the lift, y, direction only. The tube is rigid in construction; however, it is mounted on a flexible cantilevered support outside the wind tunnel. The tube oscillation was measured using an accelerometer mounted on the tube support as shown in Fig. 2. The structural viscous damping of the system is controlled by a simple non-contact electromagnetic damper (EMD). This arrangement was also used for forced vibration tests by applying a voltage across the coil. For the force measurement tests, the instrumented tube had 36 surface pressure taps at 10^o intervals located along the centre span of the tube which was mounted on an $x-y$ traverse (located outside the wind tunnel). Each tapping was monitored with a Senotec differential pressure transducer with the reference vented to atmosphere. In effect the gauge pressure was measured. The readings from these instruments were digitised and logged using an NI 8 channel, 24 bit data acquisition frame. Each channel was simultaneously sampled and automatically low pass filtered to avoid aliasing.

Artificial excitation of acoustic resonance in the tube array was achieved using two 225W speakers located on both side walls of the test section as shown in Fig. 1.

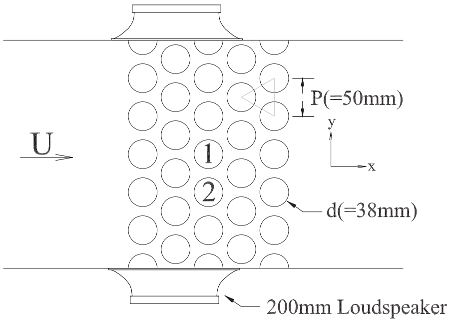

Figure 1: *P/d=1.32; Test section schematic.*

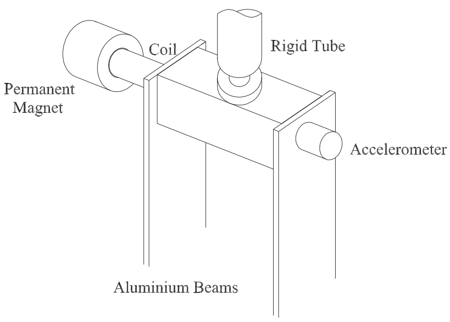

Figure 2: *Flexible tube with the EMD in situ.*

3. RESULTS

In the first instance, it was verified that a single flexible cylinder in P/d=1.32 and 1.58 became unstable due to FEI. It was also observed that the current results compared favourably with the data available in the literature.

3.1. Effect of Acoustic Resonance

The effect of acoustic resonance on fluidelastic instability was quantified for P/d=1.32 in a previous paper (Mahon & Meskell, 2008). This was achieved by artificially exciting the duct acoustics at various loudness levels and for a range of flow velocities and levels of structural damping. Acoustic resonance was found to modify the fluidelastic vibration amplitude. Fig. 3 shows a time trace of tube displacement at δ_{st}=0.088 and U=4.5m/s with and without acoustic excitation. At t=0s acoustic excitation was applied with a speaker power of 64W (SPL=140dB). It was seen that the effect of acoustic resonance was to reduce the vibration amplitude by more than 50%.

The fluidelastic vibration amplitude reduces with increasing speaker power as illustrated in Fig. 4. It was also apparent that the effect of acoustic resonance on fluidelastic instability was dependent on structural damping and flow velocity. It was also observed that the applied acoustic field increased the critical velocity, delaying the onset of fluidelastic instability.

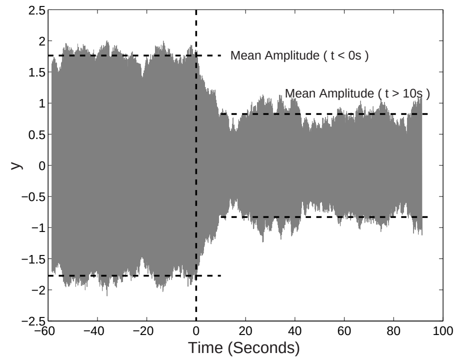

Figure 3: *P/d=1.32; Time trace of tube displacement. Acoustic excitation applied at t=0s.*

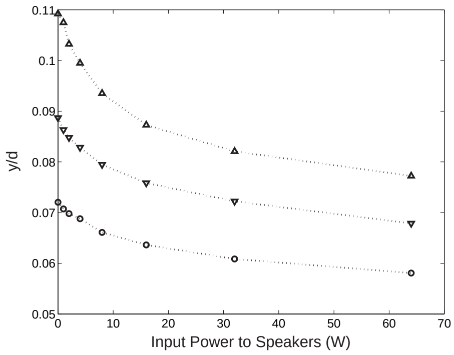

Figure 4: *P/d=1.32; Vibration amplitude against input power to speaker:* Δ, $\delta_{st} = 0.077$; Δ, $\delta_{st} = 0.098$; $\circ$, $\delta_{st} = 0.077$.

In terms of the system dynamics, acoustic resonance adds damping reducing the apparent negative fluid damping associated with fluidelastic instability. Further results on the interaction as well as the identification technique for obtaining the fluid damping can be found in Mahon & Meskell (2008).

The effect of acoustic resonance on fluidelastic instability in the pitch ratio of 1.58 was also examined. In this instance the effect of acoustic resonance was negligible. Fig. 5 shows a sequence of tests conducted at $\delta_{st} = 0.030$ and $U = 9m/s$. The test sequence moves from left to right with the input power to the speaker varying accordingly (left to right: 0, 2, 0, 32, 0, 64, 0W). It was observed that the acoustic resonance had no significant effect on the vibration amplitude. The amplitude varied from test to test independently of whether forced acoustics was applied or not. This was also shown to be the case at

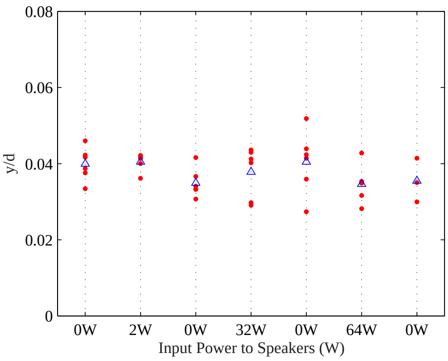

Figure 5: P/d=1.58. Sequence of tests from left to right showing vibration amplitude against speaker power at $U = 9m/s$. $\bullet$, individual test; Δ, average of the five tests

$U = 11m/s$ for the same level of damping and also for $U = 7m/s$ and $9m/s$ at $\delta_{st} = 0.017$. This suggests a fundamental difference in the fluidelastic behaviour between the two arrays tested. However, it is likely that the jet switching observed in the pitch ratio of 1.58 interfered with the convection process (time delay mechanism) destroying any subtle changes in the convection process caused by acoustic resonance. This will be discussed in detail below.

The effect of acoustic resonance on fluidelastic instability has been quantified above. However, it is difficult to envisage the physics of a true interaction between fluidelastic instability and acoustic resonance given that the phenomenon of fluidelastic instability typically occurs at a frequency of approximately $6.6Hz$ while the acoustic resonance frequency is two orders of magnitude larger, at 1092Hz (P/d=1.32). Alternatives to an interaction between the two phenomena which could explain the observations reported were discussed. It was reported that acoustic resonance adds positive damping. Tests in quiescent fluid showed that the sound field did not provide an additional damping force independently of the flow, implying that superposition of independent phenomena can be excluded. It has also been reported by Blevins & Bressler (1987) and Feenstra et al (2004) that acoustic resonance can cause a change in the pressure drop across the array, hence a change in the mean flow velocity. Such findings were not observed in this study. The effect of acoustic particle velocity (APV) was also examined as it was thought it may affect the local fluid mechanics in the vicinity of the flexible cylinder. This was done by relocating the flexible cylinder position from position 1 were the APV was a minimum to position 2, where the APV was higher, the effect of acoustic resonance on fluidelastic instability was still observable. Changing the flexible cylinder position did not result in a significant change in behaviour suggesting that the effect of APV was small.

4. INTERACTION BETWEEN FLUIDELASTIC INSTABILITY & ACOUSTIC RESONANCE EXPLORED

Two possibilities are examined which are based on the quasi-steady framework proposed by Price & Paidoussis (1984) to model fluidelastic instability. In simple terms this separates the fluidelastic force into a magnitude dependent on the steady fluid force and a phase dependent on the time delay. It was reported above that acoustic resonance was found not to have an effect on fluidelastic instability vibration amplitude in P/d=1.58, so the discussion in this section is restricted to P/d=1.32.

The effect of acoustic resonance on the surface pressure distribution around a static cylinder in the third row of the array was examined and hence determines if the acoustic resonance alters the force magnitude on the cylinder resulting in the change in vibration amplitude. Tests were conducted for a number of flow velocities ($U = 2, 4, 6, 7, 8, 10m/s$); and at a range of tube displacements ($y/d = 0, 1, 3, 5, 7, 10\%$); at various speaker input power (0, 16, 32, 64W). The set up and testing procedure are more rigorously discussed in Mahon (2008).

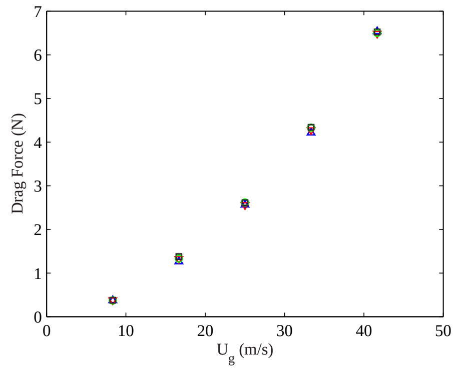

Figure 6: Drag force with and without imposed acoustics. $\triangle$, 0W; ∇, 16W; $\circ$, 32W; $\square$, 64W.

At the lower velocities of 2 and 4m/s, acoustic resonance has a small effect on the mean pressure distribution. This is reduced further at the higher velocities. When the tube was displaced similar findings were observed. The small changes in pressure distribution at lower velocities were not translated into any significant effect on the lift and drag force. Fig. 6 plots the drag force at $y/d = 5\%$ were subtle changes in the forces were observed but these were consistent with the spread in the data. Thus, the effect of the second acoustic mode on the cylinder was minimal, suggesting that acoustic resonance was not affecting the steady forces on the cylinder surface. It was also not surprising that as the flow velocity increases, and hence the mean pressure on the surface of the cylinder increases, that the effect of acoustic pressure becomes less significant as the relative magnitude difference (between flow velocity and acoustic particle velocity) increases. A maximum sound pressure level (SPL) of 140dB was used which corresponds to a pressure of 200Pa. However, it is thought that at higher SPLs that the effect of acoustic resonance on the mean surface pressure would be more significant as reported by Fitzpatrick et al (1978). They reported that artificially excited acoustic resonance (160dB) modified the pressure distribution around cylinders in the thirteenth row of a twenty six row in-line array (P/d=1.73). Acoustic resonance altered the velocity gradients across the array thus modifying the force on the cylinder. In this current study where the SPL was an order of magnitude less the effect on the steady fluid force was small. Hence, this was not the cause of the observed interaction between fluidelastic instability and acoustic resonance.

4.1. Time delay

When fluidelastic instability is discussed in the literature, a time delay between the tube motion and the resulting fluid forces is thought to be at the root of fluidelastic instability. The exact nature of the time delay is unclear and has yet to be measured directly. To further explore the observed interaction between acoustic resonance and fluidelastic instability, the time delay with and without acoustic resonance was measured.

An attempt to measure a time delay between tube motion and a point in the flow located near the flexible cylinder is discussed. In an ideal setup a time delay between tube motion and fluid forces would be measured. This was not achievable due to limitations in the setup. The justification for the current approach stems from the fact that the fluid forces on the cylinder are as a direct consequence of what is happening in the flow around the cylinder. Hence a relationship between the fluid flow and fluid forces are closely related. It therefore seems reasonable to measure the fluids response instead of the fluid force as a first attempt to measure the time delay. This conceptual approach is also consistent with the assumption of the Lever & Weaver model.

The flexible cylinder was forced to vibrate at its natural frequency of 6.6Hz. This was achieved using the electromagnetic shaker (EMS) system described previously. The input signal was generated using a HP35665A dynamic signal analyzer via a USA 370 amplifier. The excited vibration amplitude chosen corresponded to a RMS value of 1.8% tube diameter. Using the electromagnetic damper the maximum level of damping achieved was $\delta_{st} = 0.205$. In an effort to reduce the effect of turbulent buffeting additional damping ($\delta_{st} = 0.410$) was added. This was achieved by adhering lengths of rubber to the cantilever support. At the new level of damping the tube did not go unstable due to fluidelastic instability for the velocity range of the wind tunnel. Tests were conducted for three flow velocities: 4, 7 and 10m/s. The local velocity around the cylinder was measured using a single hot-wire probe. The positions around the cylinder are shown in Fig. 7. The local flow velocity was measured at $\theta = 15, 30, 60, 90, 120, 150$ and 165^o in both the in-flow (u) and cross flow (v) directions. Each test was conducted for 15 seconds at a sample rate of 8192Hz. With the excitation frequency of 6.6Hz this translates to 99 averages thus improving the signal-to-noise ratio by a factor of 10.

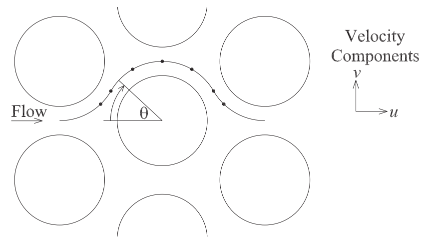

Figure 7: Hot-wire positions around the instrumented cylinder

For each test, tube motion, local flow velocity (using hot-wire anemometry) and the output signal from the amplifier (input signal to EMS) was acquired. This signal from the amplifier was used as a reference in the analysis as it produced a clean sinusoid, whereas the flow velocity and tube response measurements includes a random component as both were subject to turbulence in the flow. The reference signal was differentiated using a central difference method. The original and differentiated signals were normalised and the inverse tangent taken on the resultant of the normalised original signal divided by the normalised differentiated signal. This process presents the reference signal in the form of an angular position. The flow velocity and tube motion can now be related to an angular position. As the tube motion was forced using a sinusoidal form at the natural frequency of the structure it might be expected that this would also be observed in the flow surrounding the cylinder. It can be seen that this is the case (Fig. 8) but there are significant cycle-to-cycle variations due to turbulence in the flow. The underlying behaviour was extracted by fitting a series of harmonic sinusoids.

$$v_M = \sum (A_M \sin M\theta + B_M \cos M\theta) + c \qquad (1)$$

where v_M is the velocity, θ is the angular position of the reference signal, A_M and B_M are constants. The constants A_M and B_M were obtained using a pseudo-inverse method which yielded a least squares fit for an over-determined set of equations (Keays & Meskell (2006)).

It was found that M=5 was sufficient in all cases on the basis of minimising the normalised error between the fit and the raw data. However, the analysis technique employed to calculate the time delay between the tube motion and flow reorganisation requires the data to be represented using a single harmonic curve. Fig. 8(a) and (b) presents the tube motion and flow velocity against angular position, respectively. Also plotted is the respective single harmonic fits and it is observed that the single sinusoid captures the underlying trend in both cases. However, the flow field around the cylinder in a tube array is highly sheared and at some positions it was clear that the flow velocity does not respond linearly to the tube motion. It is therefore important to consider how the quality of the fit was determined using a number of criteria. The approach used in this study examined the energy contribution at each harmonic in conjunction with the auto-correlation between the actual data less the first harmonic fit. A good fit was deemed to have been achieved when the energy distribution at the first harmonic was greater than 95%. Below that threshold the fit was deemed to be not of the base line quality. The second criteria also had to be satisfied. This involved examining the auto-correlation of the raw data less the fit of the first harmonic. If the fit was good random noise should be all that remains. Viewing the auto-correlation of this signal determines if the resulting distribution was random or if it contained periodic artifacts.

The measured time delay was found to change slightly from test to test with the extent of the

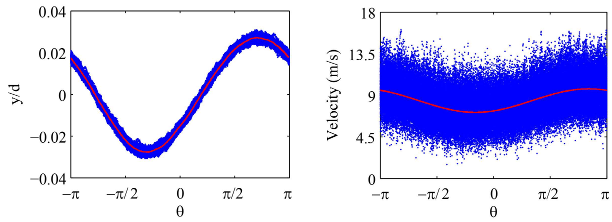

Figure 8: First harmonic fit of the tube motion and flow velocity data

deviation from the mean varying with measurement position. On average a deviation of $\sim 10\%$ was observed. This is not surprising given that the time delay has been obtained from a measurement in a flow which is highly sheared and turbulent. As the static surface pressure measurements showed smaller deviation from the mean compared to the velocity measurements in the array when all tubes were rigid, it is envisaged that the spread in the measured delay would reduce if the delay was directly measured from the fluid forces on the cylinder.

	U=4m/s		U=7m/s		U=10m/s	
Position		AR		AR		AR
15^o (v-dir)	3.0	0	3.5	3.7	4.4	2.3
30^o (u-dir)	5.1	1.4	3.4	5.7	3.5	3.6
30^o (v-dir)	6.6	0	4.9	5.4	5.8	0
90^o (u-dir)	42.0	45.9	52.4	51.1	-	-
150^o (u-dir)	-	-	5.5	9.1	4.6	3.4
165^o (u-dir)	9.3	7.9	7.9	8.1	7.0	6.0
165^o (v-dir)	10.3	8.1	8.8	9.1	8.0	6.8

Table 1: Time Delay (ms) at a range of positions in the flow field with and without acoustic resonance. Shading - illustrates the hot-wire positions and velocities where overlap between the individual time delays measured with and without forced acoustics occurs

A series of tests were conducted to measure the time delay with and without forced acoustics and each test was repeated five times. Tests were repeated for other flow velocities and also the other hot-wire positions. It was discussed previously that the measured time delay was found to wander slightly from test to test. The average (five tests) time delays with and without acoustic resonance are summarised in Table 1. In some instances a change in time delay with forced acoustics occurred but there was an overlap between the individual time delays measured with and without acoustic resonance. These are denoted by the shading in Table 1. In these instances no conclusive outcome as to the influence of acoustic resonance was realised. However, at the other positions a definitive phenomenon emerged: the acoustic resonance modified the time delay. In some instances the time delay was increased; more often the time delay was reduced. Further work is required to explore this result but this can only be rigorously examined when the time delay between tube motion and fluid forces is measured.

Assuming that acoustic resonance modifies the time delay, how could this process be justified physically? Granger & Paidoussis (1996) formulation of a memory effect (cause of the time delay) refer to vorticity generated on the surface of the cylinder resulting from tube motion. This vorticity is diffused and convected downstream by the mean flow. When the vorticity is convected far enough downstream a new steady state is reached. It was shown that the effect of acoustic resonance on the steady fluid forces was negligible (i.e. the vorticity generation process). So, as acoustic resonance was observed to have modified the time delay it must be interfering with the vorticity diffusion-convection process. It was also observed that the acoustic resonance shifted the mean velocity (both increasing and decreasing) at some positions as well as the form of the distribution with reference to the angular position of the tube vibration. This is curious, as at the current tube position the acoustic particle velocity corresponds to a minimum in this region. In this instance it appears that acoustic resonance is causing streaming. It is not unreasonable to suggest that the acoustic streaming may be interfering with the diffusion-convection of vorticity process from the surface of the cylinder suggested by Granger & Paidoussis.

5. CONCLUSIONS

It has been shown that acoustic resonance effects fluidelastic instability and this has been quantified. In an attempt to better understand the interaction between fluidelastic instability and acoustic resonance for P/d=1.32 two possibilities are examined based on the framework proposed by Price & Paidoussis (1984) to model fluidelastic instability. Acoustic resonance does not change the steady fluid force. It has also been shown that at some hot-wire (local flow velocity) positions a definitive change in the time delay between tube motion and the flow field around the cylinder emerged as a result of acoustic resonance. It is also clear that acoustic resonance modifies the mean velocity at some positions in this region where it is thought this results from acoustic streaming. Further work is required to further explore the time delay mechanism and hence the effect of acoustic resonance on it.

6. ACKNOWLEDGEMENTS

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7. REFERENCES

Andjelic, M., Popp, K., 1989 Stability effects in a normal triangular cylinder array *Journal of Fluids and Structures* **3**(2): 165-185.

Blevins, R. D., Bressler, M. M., 1987 Acoustic Resonance in Heat Exchanger Tubbe Bundles-Part 1: Physical Nature of the Phenomenon *ASME Journal of Pressure Vessel Technology* **109**: 275-281.

Chen, S. S., 1983 Instability Mechanisms and Stability Criteria of a Group of Circular Cylinders Subject to cross Flow. Part I: Theory. *ASME Journal of Vibrations, Acoustics, Stress, Reliability and Design* **105**: 51-58.

Feenstra, P. A. et al, 2004, The effects of duct width and baffles on acoustic resonance in a staggered tube array. *Proceedings of the 8th International Conference on Flow-Induced Vibration* **1**: 459-464.

Fitzpatrick, J. A. et al, 1978, Some observations of the pressure distribution in a tube bank for conditions of self generated Acoustic Resonance. *International Symposium, Vibration Problems in Industry, Keswick, U.K.* **Paper 3:6**: 1-13.

Granger, S., Paidoussis, M., 1996 An improvement to the quasi-steady model with application to cross-flow-induced vibration of tube arrays *Journal of Fluid Mechanics* **320**: 163-184.

Keays, J., Meskell, C., 2006 A study of the behaviour of a single-bladed waste-water pump *Journal of Process Mechanical Engineering* **220**(2): 79-87.

Mahon, J., Meskell, C., 2008 Interaction between acoustic resonance and Fluidelastic Instability in a normal triangular tube array *ASME Journal of Pressure Vessel Technology*, In production.

Mahon, J., 2008 Interaction between acoustic resonance and Fluidelastic Instability in a normal triangular tube arrays *PhD Thesis, Trinity College Dublin, Ireland.*

Meskell, C., Fitzpatrick, J. A., 2003 Investigation of non-linear behaviour of damping controlled fluidelastic instability in a normal triangular tube array. *Journal of Fluids and Structures* **18**: 573-593.

Price, S. J., Paidoussis, M. P., 1984 An improved mathematical model for the stability of cylinder rows subject to cross-flow. *Journal of Sound and Vibration* **97 (4)**: 615-640.

Price, S. J., Zhan, M. L., 1991 Fluidelastic behaviour of a normal triangular array subject to cross-flow. *Journal of Fluids and tructures* **5**: 259-278.