

Vibration in Heat Exchangers

FRANZ MAYINGER and HANS GÜNTHER GROSS

Institute für Verfahrenstechnik
der Universität Hannover
Hannover, FRG

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1. INTRODUCTION

To improve the thermal efficiency heat exchangers are commonly equipped with baffles. These devices produce a cross flow around the tube bundles which is favourable for the heat transport, however, which also may induce vibrations. If the amplitudes of the vibrations become too high, fretting corrosion and erosion of the tubes at the position of the baffles may occur.

Since more than 30 years research activities are underway to study the vibration phenomena in cross flow bundles of heat exchangers. Mostly it is assumed that oscillations are excited by vortices departing from the tubes, then the strongest vibration should be observed if the departure frequency of the vortices and the resonance frequency of the tubes are identical. A safe layout of the tube banks would then be not too difficult, one just has to avoid the coincidence of these both frequencies.

2. MECHANISMS EXCITING VIBRATIONS

If the vibrations are excited by the vortices, which again are produced by the flow around a tube, the exciting frequency can be easily predicted by the Strouhal-number /1/

$$S = \frac{f_w \cdot d}{w} \quad (1)$$

which gives the relationship between the vortex frequency f_w , the diameter of the tube d and the velocity w of the flow, in front of the tube or the cylinder. In the literature it is clearly stated /2 - 4/ that there is a linear connection between the vortex frequency and the flow velocity, which means that the Strouhal-number is constant. After a careful literature survey Chen /5/ found that the value of the Strouhal-number should be between 0,17 and 0,21, for Reynolds-numbers from 300 up to more than $2 \cdot 10^5$.

Buffeting

Owen /15/ studied the turbulence behind the tubes and found that this may cause vibration. He called this aeroelastic exciting phenomenon buffeting. These turbulent and stochastic velocity fluctuations are mainly due to the perturbation of the boundary layer on the rear side of the tube. These fluctuations have a very wide frequency spectrum.

A special case is the resonant buffeting, which also has a statistical energy distribution, however, in addition a periodical velocity fluctuation is superimposed. If the frequency of this periodical fluid dynamic exciting force coincides with the resonance frequency of the tubes, vibrations of large amplitudes may be the consequence.

Galloping - Wake Galloping

In civil engineering another vibration exciting phenomenon is wellknown, which is called galloping - for a single obstacle - or wake galloping - for a group of obstacles like cylinders -. Galloping was observed with chimneys or with the cables of high voltage transportation lines /16, 17/. As shown in fig. 1 for the example of a tube or rod bundle with 3 rows galloping can produce a lifting force due to the partial deflection of the flow. This phenomenon mainly in the second and in the third row may perform a vibration rectangular to the direction of the inlet flow.

In civil engineering galloping was mainly observed with non-cylindrical obstacles, like prismatic rods, where one has to distinguish between a lift force

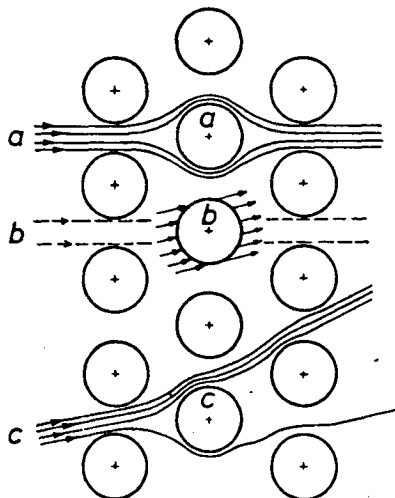


Fig. 1: Flow paths in rod bundle

$$L = \frac{1}{2} \cdot c_L \cdot H \cdot \rho \cdot w_{rel}^2 \quad (2)$$

and a drag force

$$D = \frac{1}{2} \cdot c_D \cdot H \cdot \rho \cdot w_{rel}^2 \quad (3)$$

In tube banks of staggered arrangement the lift force may be due to the mentioned deflection of the flow. The critical velocity can be formulated by solving the differential equation, describing the exciting and damping effects as well as the inertia forces

$$w_{krit} = - \frac{2 \cdot d^*}{\rho \cdot H \cdot K} \quad (4)$$

In equation (4) d^* represents a damping factor, H a characteristic length of the profile and K a factor describing the combination of lift and drag forces. Compared to the before mentioned phenomena - exciting by vortices and by turbulent buffeting - where only at a single frequency high amplitudes of vibration occur, galloping or wake galloping can produce strong vibrations above a certain flow velocity over a wide spectrum of frequencies.

Aeroelastic Coupling

Another mode exciting vibrations in cylindrical tube banks may be the aeroelastic coupling, first mentioned by Livesey /18/ and later by Connors /13/. In contrast to the vortex or galloping induced vibration, the tubes not only move rectangular to the flow but also in the flow direction.

The aeroelastic coupling is a consequence of the movement of the rods. If the rod leaves its original or stationary position, the fluiddynamic forces around the rod, which are influenced by the relative position to the neighbouring rods, change. So a new exciting force for vibration may be created. The frequency of the vibration, however, is then not only depending on the flow velocity, but also on the resonant frequency of the surrounding rods. The main influencing factor with aeroelastic coupling is the movement of the neighbouring tubes, which means that a small vibration of a few rods in the tube bank may excite other rods by fluiddynamic forces and therefore this phenomenon is called aeroelastic coupling. Contrary to the exciting modes discussed before - like buffeting or galloping - with aeroelastic coupling no favoured vibration direction can be observed. The vibration movement of each rod is depending on and influenced by the movement of its neighbouring rods.

Spivack /6/, Dumpleton /7/ and Gregorig /8 - 10/ found that the vortices cannot be the only and main reason for inducing vibrations in tube banks. Most of the experiments in the literature studying vibrations in heat exchangers used tube banks where only one tube could freely move and all others were fixed /11/. Owen /12/ was one of the first who doubted about the influence of the vortices on exciting vibrations and found that freely moving and interfering tubes may have other mechanisms to produce vibration. Also Connors /13/ observed vibration phenomena, which could not be explained by the vortex theory. He introduced the idea of aeroelastic coupling between the different tubes in the bank.

The vibration induced by vortices was described by Magnus /14/, starting from the equation of movement

$$m\ddot{y} + d^*\dot{y} + ky = L \sin \Omega t \quad (5)$$

where

L is a lifting force

Ω the departing frequency of the vortices

d^* a damping factor

k a constant, representing the reacting force.

Introducing the damping factor of Lehr D^* and the resonance frequency ω_0 of the tube one can predict the maximum vibration amplitude

$$|y_{\max}| = \frac{L}{k} \cdot \frac{1}{\sqrt{\left(1 - \frac{\Omega^2}{\omega_0^2}\right)^2 + 4D^* \frac{\Omega^2}{\omega_0^2}}} \quad (6)$$

by solving equation (5). From equation (5) and (6) one can calculate the velocity w_{res} of the flow at which resonance of the tube is to be expected

$$w_{\text{res}} = \frac{\omega_0 d}{2\pi S} = \frac{f_0 d}{S} \quad (7)$$

Practical experience, however, shows that vibrations in heat exchangers occur sometimes far before this velocity is reached and sometimes also much later.

3. FLOW PULSATIONS AND ENERGY DENSITY DISTRIBUTIONS

In a turbulent flow fluctuations are superimposed to the mainflow direction which may be of periodic or stochastic nature. Using the root-mean-square value

$$\sqrt{\overline{f(t)^2}} = \sqrt{\frac{1}{2T} \int_{-T}^{+T} f^2(t) dt} \quad (8)$$

of these fluctuations the grade of turbulence

$$Tu = \frac{\sqrt{1/3 (\overline{u_x^2} + \overline{u_y^2} + \overline{u_z^2})}}{W} \quad (9)$$

can be defined as wellknown in the literature.

Like the kincetic energy of the flow in the main direction, also the turbulence represents a certain fluctuation energy. This energy can be diversificated according to the frequency spectrum of the turbulent flowpulsations. If in addition one refers the energy of a special pulsation frequency to the mean or integral value of the energy of the whole spectrum, one can define a reduced energy density function

$$F_{red}(f) = \frac{\overline{E^2(f_m)}}{b \overline{E^2}} \quad (10)$$

for a given frequency f , where E is the voltage measured at a hot wire anemometer. In equation (10) b is the range of the frequency under consideration for this function.

In the flow there are turbulent and periodic components of velocities

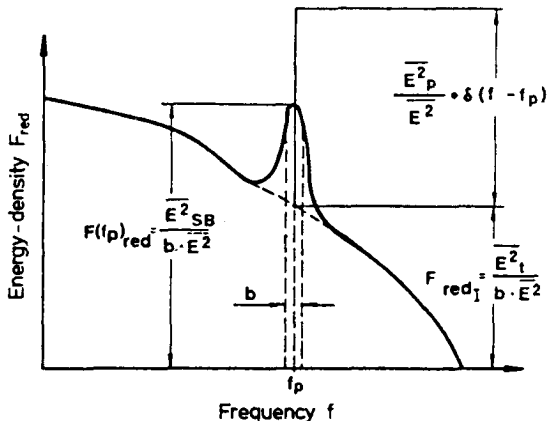


Fig. 2: Energy - density - spectrum (schematic)

$$\overline{U^2} = \overline{U_t^2} + \overline{U_p^2} \quad (11)$$

and the total energy exciting vibrations is the sum of both components.

$$\overline{U^2} \cdot \int_0^\infty F_{\text{red}} df = \overline{U_t^2} \cdot \int_0^\infty F_{\text{red}_t} df + \overline{U_p^2} \cdot \int_0^\infty F_{\text{red}_p} df \quad (12)$$

In equation (12) $\overline{U_t^2}$ represents the kinetic energy of the turbulent fluctuation and $\overline{U_p^2}$, that of the periodic fluctuations of the frequency f_p . A schematic distribution of the energy density of a flow with turbulent and periodic fluctuation is illustrated in fig. 2. The energy distribution shows a pronounced peak at the frequency of the periodic fluctuations. From this energy peak the vibration of the rods may obtain its exciting forces.

4. EXPERIMENTAL OBSERVATIONS

Detailed experiments studying the vibration phenomena in tube banks and also in tube or rod arrays of one row only were performed by Gross /19/. The main aim of these research activities was to get information which of the above mentioned exciting phenomena may have the main influence on the development of vibrations. In the experimental setup, staggered and inline arrangements of tube banks in airflow were tested. Contrary to the usual habit in the literature all tubes in the bank were movably suspended. The reacting forces in the suspension device were made rather weak and large vibration amplitudes were allowed to be able to study the influence of even small exciting forces and to get a well readable signal of the rod movement. For a single row of rods the course of the movement - i.e. the vibration pattern - is illustrated in fig. 3. There is a clear alternation in the direction of the vibration course. From this observation one can conclude that the vibration is mainly excited by aeroelastic coupling. This assumption is supported by comparing the critical velocity, when the vibration started, with the Strouhal-criterion in equation (1). The measured critical velocity was by a factor of 2 higher than that

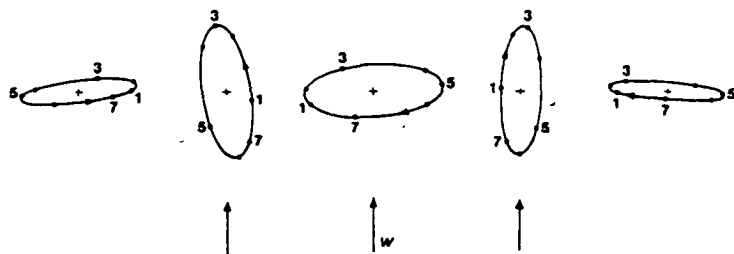


Fig. 3: Vibration pattern in a single row

one predicted by Chen /5/ with the Strouhal-criterion. Therefore vortices leaving periodically the lee-side of the tubes could not be the main reason for inducing these vibrations.

Similar observations could be made with a two-row arrangement. Measurements with this configuration were made under two different conditions, namely with all rods freely suspended and with the first row kept firmly in position. Fig. 4 gives information about the vibration amplitude of the rods in the first and in the second row as a function of the flow velocity. With all rods movably suspended, the second row starts earlier - i.e. at a lower flow velocity - to vibrate then with the first row fixed. Also the vibration amplitudes are larger then in the partially fixed case. The fact that with fixed rods in the first row, the onset of the vibration in the second row could be largely delayed to higher flow velocities, seems to be a clear hint that aeroelastic coupling plays an important role, which, however, refers only to the second row. For the first row there must be an additional exciting mechanism, for example galloping or wake-galloping.

Staggered Arrangement of Tube Banks

Besides the flow velocity and the suspension of the rods also geometrical parameters may have an influence on the beginning and the amplitudes of vibrations. A significant geometrical parameter is the ratio τ between the distances of the rods and their diameter d . This ratio was varied in the tests of Gross /19/ in the range 1,1 to 1,5. Also the arrangement of the rods - staggered or in line - certainly plays an important role. The velocities in the narrowest gap between the rods were varied between 10 and 60 m/s corresponding to Reynolds-numbers from 20.000 to 120.000. The diameter of the rods to which this Reynolds-number is referred to was 30 mm. The suspension of the rods was made in such a way that the elasticity was approximately the same as that of heat exchanger tubes between two baffles. With up to 10 mm the absolute amplitude A of the possible rod movement was rather large.

In the fig. 5, 6, 7 the observed vibration amplitudes for s over d ratios of 1,1, 1,3 and 1,5 are illustrated.

Only with the narrowest spacing the rods in the first row begin to vibrate before a movement of the other rods can be observed. With larger spacing - 1,3 and 1,5 - the second and the third row are mostly effected by vibration. Rows downstream and especially the last row in the bundle, which in these tests was the 6th one, have much lower amplitudes. The velocities where the vibration starts are by a factor 5 higher than the Strouhal-criterion would predict. Although calculations based on resonant buffeting, i.e. exciting by turbulent fluctuations, would result in a much lower critical velocity. Galloping may have influence in the lower velocity range on the vibration as can be concluded from stroboscopic observations giving information about the course of the rod movement. When the first vibration starts the rods in the second row show a oscillation movement almost exclusively rectangular to the main flow direction. With

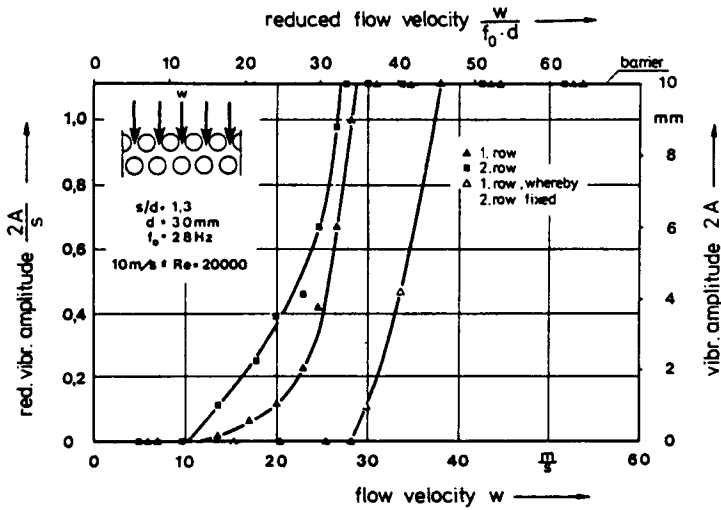


Fig. 4: Vibration amplitude of rods in a 2 rows arrangement

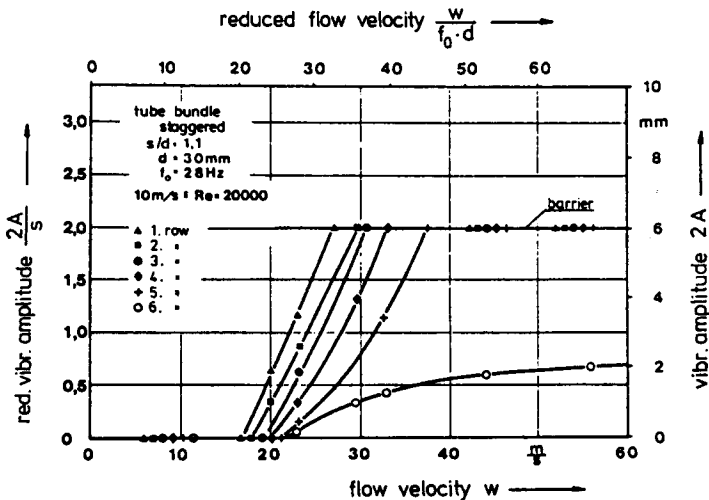


Fig. 5: Vibration amplitude of a staggered cross flow bundle ($s/d = 1,1$)

increasing velocity the movement of the rows has a more stochastic character, as illustrated in fig. 8, changing slowly with time. From fig. 8, it can be clearly seen, that the second row undergoes the strongest vibrations and that the first one is only weakly influenced by the vibration inducing effects.

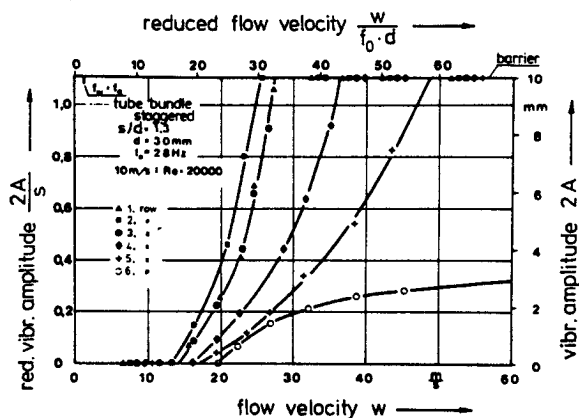


Fig. 6: Vibration amplitude of a staggered cross flow bundle ($s/d = 1,3$)

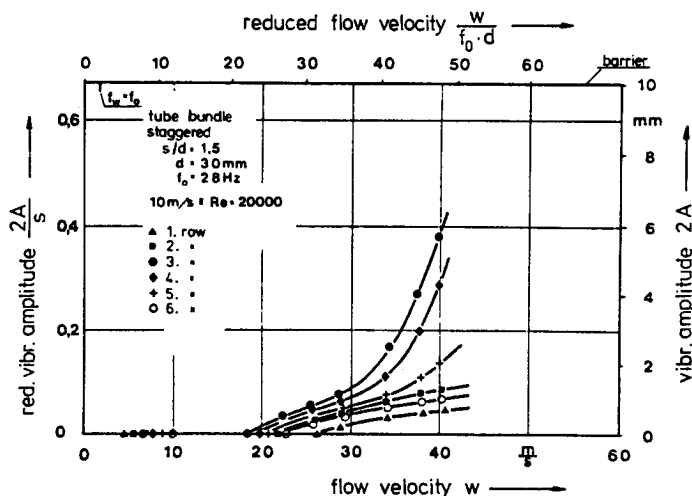


Fig. 7: Vibration amplitude of a staggered cross flow bundle ($s/d = 1,5$)

The s/d ratio τ has a pronounced influence not only onto the beginning of vibration but also onto the amplitudes with $\tau = 1,3$ vibrations were observed at the lowest flow velocities as demonstrated in fig. 9. This is valid not only for the critical row in the bundle - usually the second row - but also for the bundle as a whole.

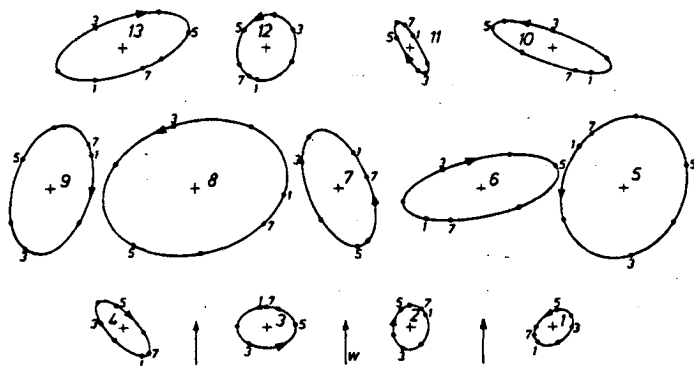


Fig. 8: Vibration pattern in a cross flow bundle (first 3 rows)

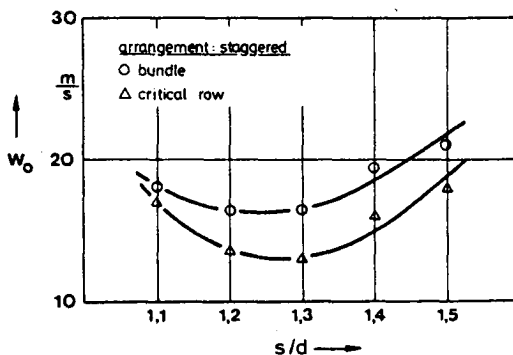


Fig. 9: Onset of vibrations as function of distance - diameter ratio

Rod Bundles with Inline Arrangement

Similar to the staggered conditions also for inline arrangement the amplitudes are increasing continuously with the flow velocity, and the vibrations were also mostly pronounced in the first 3 rows. From the fig. 10, 11 and 12 one can conclude that the position of critical row moves downstream with increasing s/d ratio. For the narrowest spacing the first row is undergoing the largest amplitudes and starts first with vibrations as illustrated in fig. 10. At a s/d ratio of 1,3 the vibration amplitudes of the first two rows are almost identical. This exponential sensitivity to vibrations is restricted to the second row only at the s/d ratio of 1,5. The first and the second row - as illustrated in fig. 12 - show first symptoms of damping effects which are clearly pronounced for the 4th and the following rows.

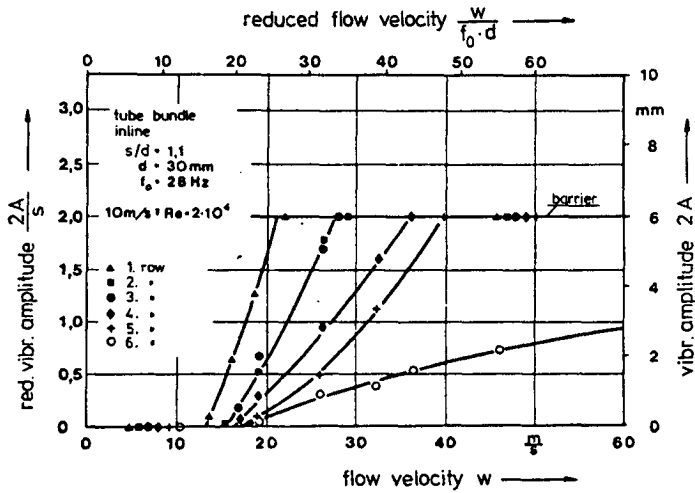


Fig. 10: Vibration amplitude of in-line cross flow bundle ($s/d = 1,1$)

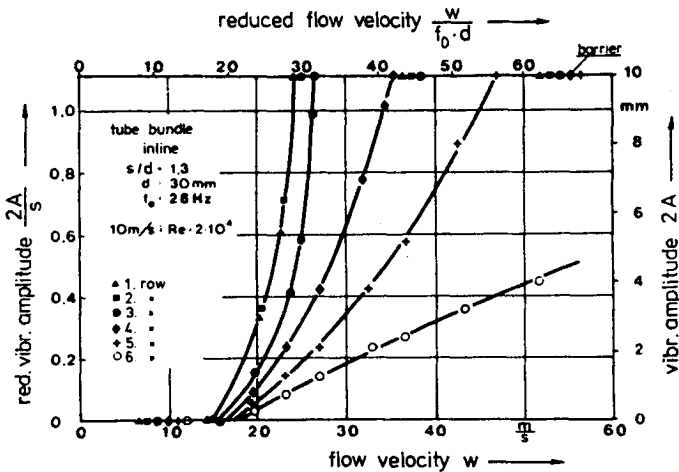


Fig. 11: Vibration amplitude of in line cross flow bundle ($s/d = 1,3$)

The course of the vibration movement is in the inline arrangement strongly different from that under staggered conditions. As illustrated in fig. 13 which shows the vibration course in the first three rows of an arrangement with a s/d ratio of 1,3,

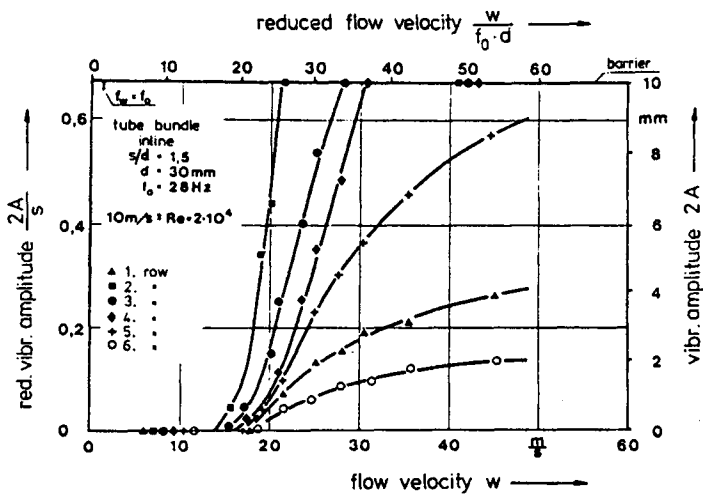


Fig. 12: Vibration amplitude of inline cross flow bundle ($s/d = 1,5$)

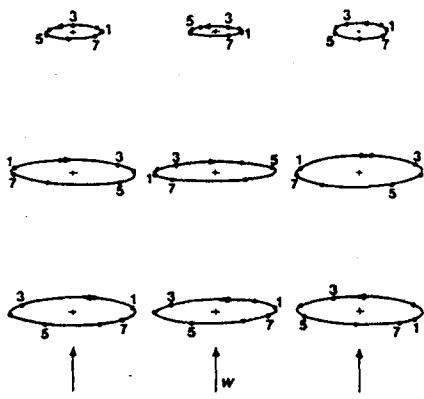


Fig. 13: Vibration pattern, in line arrangement (first 3 rows)

the rods are oscillating perpendicular to the flow. Within one row all rods are oscillating in phase. With respect to the next row, however, a phase shift of 180° was observed. The vibration phenomenon of a purely perpendicular movement as well as the range of velocities at which the first vibrations occurred give the hint that galloping is the main exciting effect. Fig. 13 also gives a good visuell impression that the first two rows are undergoing the largest amplitude which decrease tremendously already for the third row. Aeroelastic coupling seems to have no or negligible

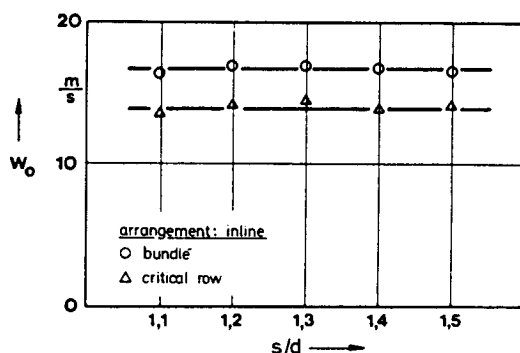


Fig. 14: Onset of vibrations as function of distance-diameter ratio

influence and for exciting by vortices the critical velocity is to high.

The critical velocity, i.e. the flow velocity at which vibrations start, is independent from the s/d ratio as shown in fig. 14. This applies for the first row as well as for the whole bundle. There is a simple explanation for this observation. As we learned from fig. 8 with a staggered arrangement the rods moved in all directions while vibrating within the bundle. This caused a continuous change of the gaps between the rods which again was the reason for aeroelastic coupling. The relative displacement and its consequences of the rods within the bundle as well as the sensitivity of the aeroelastic coupling to this displacement undergoes a maximum and therefore at a certain s/d ratio the bundle has the lowest stability against vibrations. In the inline arrangement aeroelastic coupling plays a subordinated role and therefore back fitting effects by the displacement of the rods are not present and so the s/d ratio does not influence the sensitivity against vibrations. In other words due to the in phase movement of the rods the gaps between the rods remain constant and no additional fluctuation energy exciting the vibrations is produced.

The influence of the aeroelastic coupling can be obviously demonstrated by measuring the drag forces acting on the rod along its vibration course. The product of the drag force times the way of displacement is the energy absorbed by the rod due to the flow fluctuations. One can do this test twice the first time with movable suspended neighbouring rods and then with fixed ones. Under the first conditions - movable rods - one measures an absorbed energy which is by the factor 8 larger than in the fixed surrounding.

Cinematographic observations show that the vibrating tube takes another way during its upflow movement than at the downflow one. Due to this it can be easily understood that the absorbed energy especially from periodical fluctuations excited

by the neighbouring rods is different. The consequence is an aeroelastic coupling. Measurements with a hot wire anemometer showed that periodical vortices are only existing behind the rods in the first two rows whilst in the following rows only stochastic turbulent fluctuations could be observed.

5. CONCLUSIONS AND MEASURES TO REDUCE VIBRATIONS

There seem to be two effects mainly influencing the vibration of tube- or rod bundles, namely the wake galloping and the aeroelastic coupling. From this information the conclusion can be drawn that two measures could be taken in account to reduce the sensibility of a bundle against vibrations namely

increasing of the inlet turbulence

putting out of tune the resonance frequencies of neighbouring rods.

The inlet turbulence can be easily increased by placing a grid upstream of the first row of the bundle. Already Vickery /20/ found a reduction of the oscillating pressure onto a prismatic rod in the order of 100 % by increasing the inlet turbulence. Using a punched plate with wholes of 10 mm diameter and placed 20 cm upstream of the first row a remarkable improvement of the stability against exciting vibrations could be observed /19/. By this measure the grade of turbulence which in the tests before was in the order of 0,7 % could be raised up to 50 %.

The improvement was much more pronounced for the staggered arrangement than for the inline one and the onset of vibrations could be shifted to velocities which were almost twice of that under low turbulence conditions, as illustrated in fig. 15. The vibration behaviour shown there, has to be compared with the results presented in fig. 5, where the same geometrical of the large difference in turbulence. The dotted line in fig. 15 represents the amplitude of the critical row at low turbulence corresponding the conditions in fig. 5. Comparing both figures, one can see that the grade of turbulence mainly influences the vibration behaviour of the critical rows, which are usually the three inlet rows. Further downstream there is no effect of the turbulence promotor which can be easily explained by the fact that then the grade of turbulence is anyhow high enough due to the perturbation of the flow in the first rows. It can be assumed that the increased inlet turbulence affects the drag coefficient rectangular to the flow direction and reduces by this the onset of aeroelastic coupling.

Whilst the increasing of the inlet turbulence is certainly a measure of practical use, the mistuning of the resonance frequency of neighboured rods seems to be more of academic interest. Never the less it should be briefly pointed out here that this can reduce the vibration amplitudes remarkably. It, however, does not change the critical velocity as shown in fig. 16 for the most sensitive s/d ratio of 1,3 and a staggered arrangement. The measured results in this figure again have to

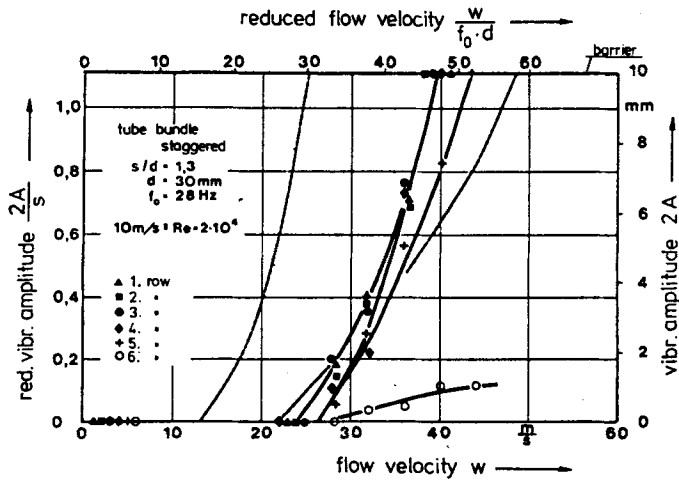


Fig. 15: Vibration amplitude with increased inlet turbulence

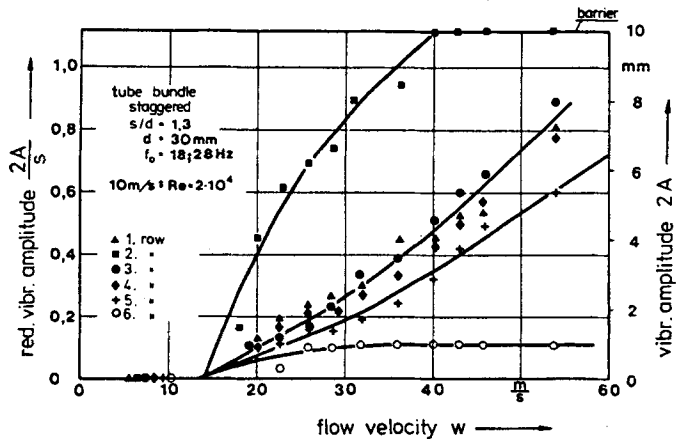


Fig. 16: Vibration amplitude with misstuned resonance frequency

be compared with that of fig. 5. The resonance frequency of the rods in fig. 5 was 28 Hz. The frequency of the bundle was then misstuned in that way that each second rod in a row was adjusted to a resonance frequency of 18 Hz simply by using a higher mass, i.e. a thicker wall of the tube. A remarkably reduced amplitude mainly in the 1st and 3rd row was the consequence. The situation can be still improved if instead of two different resonance

frequencies three are applied.

In a staggered arrangement the vibration is mainly induced by aeroelastic coupling - as we can conclude from the experimental results discussed before. This aeroelastic coupling is introduced by the beginning of oscillations in the critical row. From the experiments one also can deduce a simple empirical correlation for the critical velocity /19/.

$$W_{krit} = - \frac{2 \cdot d^*}{\rho \cdot d \cdot l \cdot K} \quad (13)$$

at which the vibrations start in a flow of normal turbulence. This velocity - as already pointed out in chapter 2 - is mainly influenced by the damping factor d^* , a constant K representing the drag force, the density of the fluid ρ , the diameter of the tube d and the length of the tube between two baffles.

Exciting by periodic vortices or by resonant buffeting can only expected in the first row at the very beginning of the vibration. The amplitudes in this region, however, are low and no rod damage should be expected.

There is certainly a wide and interesting field of further research work for better understanding vibration phenomena and improving the design of heat exchanger tube banks.

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