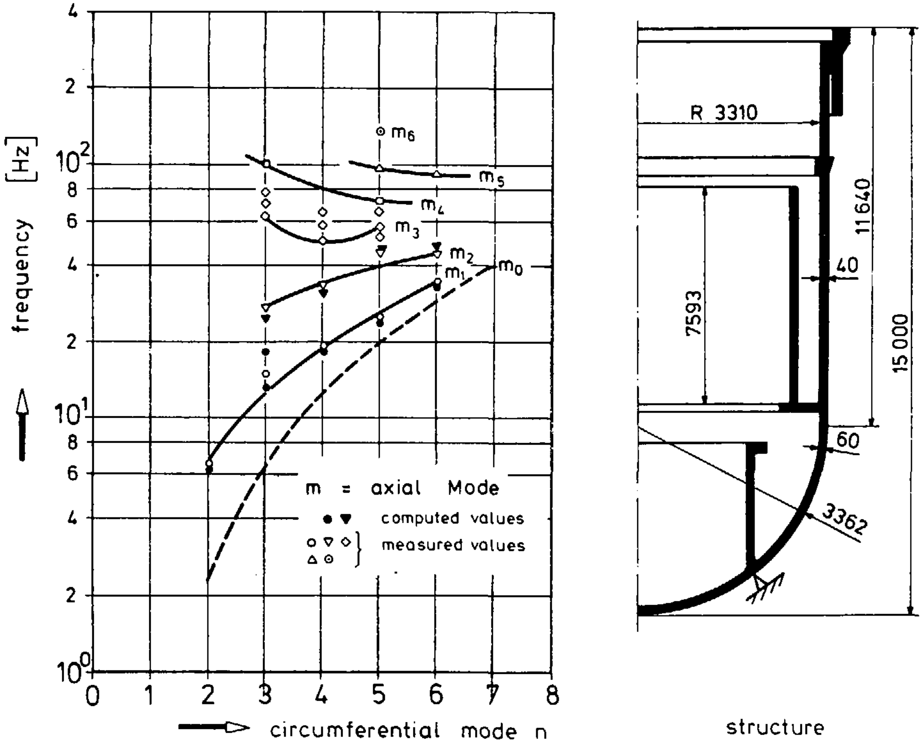

FIG. 9. Predicted and measured Eigenfrequencies of the SNR 300 reactor vessel

Substantiation of Vibration Strength of Nuclear Reactor and Steam Generator Internals; Main Problems by V. G. Fyodorov and V. F. Sinyavsky, USSR.

ABSTRACT

The report details the scope and priority of studies necessary for substantiation of vibration strength of steam generator tube bundles and reactor fuel assemblies, and design modifications helping to reduce flow-induced vibration of the internals specified.

Steam generator tube bundles are studied on the basis of a standard establishing vibration requirements at various stages of design, manufacture and operation of a steam generator at a nuclear power station. The main vibration characteristics of tubes obtained through model and full-scale tests are compared with calculation results.

Results are provided concerning test-stand vibration tests of fuel elements and fuel assemblies.

I. INTRODUCTION

Coolant flow in the NPS circulating circuit is the most dangerous source of appearance of disturbance forces which are able to induce and maintain vibration of the structural elements,primarily, the reactor and steam generator internals. Currently, the research study of the NPS equipment vibration assumes the fundamental nature when one should not be limited by the development of private solutions to prevent the separate structures from excessive vibration but conduct the comprehensive program of theoretical and experimental works. Recent tendency to the continuous increasing of power, reduction of weight-dimension characteristics of the reactor plants as well as the simultaneous increasing of requirements for their safe operation predetermine the most increasing importance of vibration problem and its actuality in the nuclear energy [1,2] .

In spite of internals variety the following structural elements for which flow-induced vibration is the most dangerous may be specified:

a) Cylindrical shells of large diameters and high rigidity. When coolant flows around these shells even low intensity of hydrodynamic forces causes significant total disturbance force.

b) Thin-walled tubes of small diameter and large length which are characterized by low natural frequencies of vibration.

c) Units interconnected with a small gap within which relative displacements during vibration may occur.

The simultaneous use of the main technique for reduction of internals vibration (by means of increasing their rigidity, decreasing the intensity of disturbing forces, increasing the vibration damping) together with vibration and vibroacoustic check is of the highest efficiency. As far as the fast reactor equipment is concerned the economical and ,frequently, the only possible way of investigating the vibration is the use of simulation and substitution of water for sodium.

2. VIBRATION OF REACTOR INTERNALS

Hydraulic circuit of the reactor with coolant supply through some of the radial nozzles into the annuli between the vessel, shield and core barrel predetermines the availability of disturbing hydrodynamic forces over the frequency range 2-500 Hz with pressure fluctuation amplitudes 0.01-1 kg/cm^2. Just these forces caused the vibration of thermal shields and then fatigue breakdown of securing pins as well as emergency situation with shield breaking [2,3] . These emergency cases have resulted in the numerous investigations carried out on the large-scale models(scale 1:4 or scale 1:5) to determine natural frequencies of the structural elements vibration and amplitude-frequency characteristics of the coolant disturbing forces as well as in the essential improvement of internals structures [4] . Circulation loop and pumps may significantly contribute into the acting hydrodynamic flow forces,that is why there is a tendency to limit the maximum pressure fluctuation at the pump delivery by the value of 0.2 kg/cm^2. Just a circulation loop configuration results in the selective amplification of the coolant pressure fluctuation over the specified frequency range . Results of investigations carried out on the test stand have shown that substitution of equalizing grid for the gate valve reduces the circulation loop pressure fluctuations by 5 times while removing of one operating blade in the pump almost does not change the coolant pressure fluctuations [2] .

To investigate the reactor vibrations the following semiempirical approach is used:

a) Construction of the calculation model using the experimental data obtained.

b) Measurement, in air and water, of natural frequencies, modes and decrements of elements vibration using the large-scale (scale 1:5) reactor and circulation loop model.

c) Determination of dynamic stresses, internals vibration and acceleration amplitudes as well as pressure fluctuations and acoustic spectrum at various water flow rates on the test stand.

d) Correction of the calculation methods and evaluation of the main vibration characteristics for the NPS structures.

e) Formation of the measurement program during the starting-and-adjustment works at the NPS.

f) Vibration, hydrodynamic and acoustic measurements at the NPS.

g) Correction of the simulation methods and vibration calculations for the internals.

Investigation experience shows that the available calculation methods with regard for hydroelastic nature of internals-with-coolant disturbing forces interaction are the most difficult to develop as there is a great uncertainty in vibration damping and correlation of hydrodynamic forces, of different values, variable in time and space which essentially depend upon the vibration amplitude sought for. Vibration hydroelastic nature shall be taken into account during the calculation model construction when vibration of large amplitudes not allowable during long-term equipment operation occurs. Therefore, determination of conditions for appearing the intensive hydroelastic vibrations of the internals and their elimination under all the NPS operating conditions assumes the great importance.

The most important and complicated processes requiring thorough investigations during design stage are vibration wear and fretting-corrosion of the core elements.

To investigate the operating conditions of compensating assemblies of reactors WWR-440, hydrodynamic and vibrational tests of assemblies with casing thickness δ = 1.5mm and δ = 2.1mm have been carried out in supposition that vibration of a hexanedral assembly precipitates the fretting-corrosion process of its zirconium casing in the areas of contact with steel spacing grids of the fuel element bundle [5]. Investigation of reasons resulting in vibration and fretting in the fuel elements of CANDU reactor [6] has shown that wear of the fuel element cans is increasing with growth of coolant velocity in the horizontal channels.

Assemblies tested which differ, mainly, only in the casing thickness (1.5 and 2.1mm) have fuel follower 1 consisting of 126 fuel elements with 12 spacing grids and absorber 2 which are connected to control rod drive on mast 3, Fig.1. Fuel elements are arranged in triangular array with pitch 12.2mm. Fuel element outer diameter is equal to 9.1mm, wall thickness - 0.65mm. Experimental channel 4 of the hot run stand for assemblies testing instrumented with outer accelerometers 5 for vibroacoustic measurements (see Fig.1) is a closed loop with forced water circulation. Inner tube cavity simulates full-scale channels for fuel assemblies and compensating assembly, Fig.1. Water circulation on the stand is caused by the pump; from the discharge nozzle of the pump water enters the column with the compensating assembly and then, having passed through the mechanical filter, returns to the pump suction. Loop water is heated up by the electrical heaters and the temperature conditions required are maintained by the systems of electric heater power control and cooling.

Tests have been carried out with various arrangements of compensating assemblies within the cannel and different

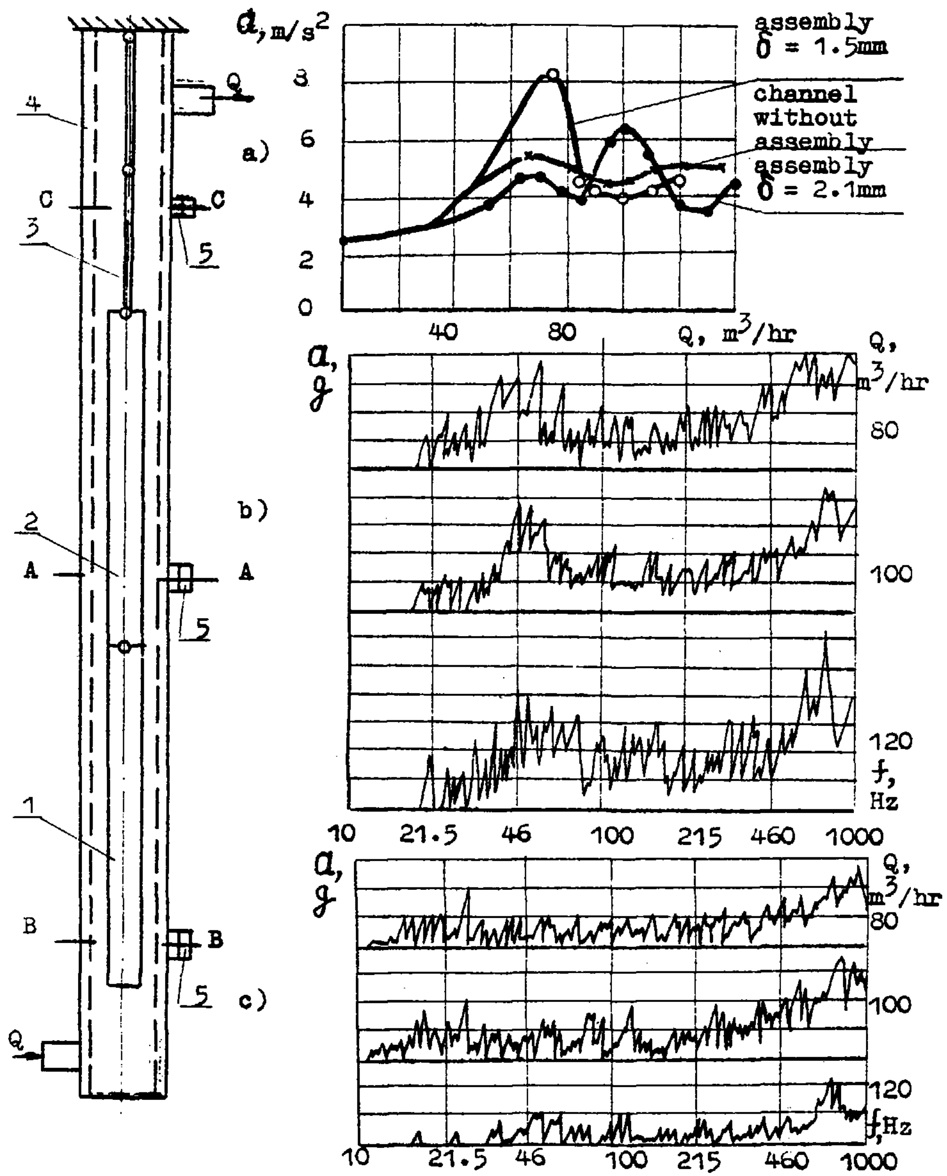

Fig.1. Results of vibroacoustic check of assembly in the test channel:
a) root-mean-square value of acceleration a;
b) frequency spectrum for assembly δ = 1.5 mm;
c) frequency spectrum for assembly δ = 2.1 mm.

water flow rate Q through the assembly.

Under the action of hydrodynamic forces of the coolant longitudinal flow the fuel elements of the compensating assemblies may travel in the transverse direction within the stand testing channels simulating the standard channels for fuel assemblies and compensating ones. To measure these dynamic displacements in sections A-A and B-B (see Fig.1) of the test stand the holes have been drilled so that inductive displacement pick-ups rods with the appropriate sealing within the stand column may be installed in the middle of each edge of the assembly tested. Rod of each pick-up has been arranged in such a way that it was possible to measure the transverse displacements of assembly in all directions by the value over 3mm.

Prior to displacement measuring and at the end of the tests one should calibrate the test channels. Working range of the displacement pick-ups equals f =0-10 Hz.

To carry out vibroacoustic measurements three accelerometers, 120° apart, in each cross-section were installed on the outer surface of column in the sections A-A, B-B and C-C.

Besides, dynamic stresses of some fuel elements have been measured with the help of thermoresistors in the region of the fuel element end-piece and in the middle of spans between the spacing grids. The total number of thermoresistors installed on the fuel elements cans were 12.

Vibroacoustic characteristics of column-assembly system were measured over the range of water flow rates Q=20-140m^3/hr through the assembly tested with its following positions: lower position, assembly is in suspension from below, assembly is at $\frac{1}{4}$ H , $\frac{1}{2}$ H and $\frac{3}{4}$ H (where H- assembly height); and upper assembly position. Vibroacoustic noise spectra and root-mean-square level of vibroaccelerations versus water flow rate through the assembly have been obtained for these conditions.

Root-mean-square values of accelerations versus water flow rate Q for the compensating assembly with casing thickness δ = 2.1mm and δ =1.5 mm is presented in Fig.1. The main difference of the vibroacoustic-noise frequency spectra was that the compensating assembly with casing thickness δ = 1.5mm is characterized by considerable low-frequency spectrum components over the range f =30-100 Hz which are essentially less for the modificated compensating assembly with casing thickness δ = 2.1mm.

Dynamic displacements of the both assemblies within the test-stand channel have been measured with various flow rates Q through each assembly for its two positions: middle and upper. Typical oscillogram of the compensating assembly dynamic displacements(δ = 1.5mm) is shown in Fig.2a for the pick-ups located in section A-A. The largest displacements of both compensating assemblies(δ =1.5mm and δ = 2.1mm) were observed for their middle position within the test channel near the displacement pick-ups mounted in section B-B. The corresponding displacements of the both assemblies in the upper position within the test channel were less by about 20%.

Maximum dynamic displacements of the compensating assemblies(δ =1.5 and δ =2.1mm) versus various water flow rates Q are given in Fig.2b. For both assemblies the tranverse dynamic displacements within the test channel increase with growth of water flow rate Q but in this case the maximum displacements over the range of Q=20-140 m^3/hr do not exceed 1mm for the modificated compensating assembly (δ =2.1mm) and are equal to 3mm for the compensating assembly (δ = 1.5mm) that equals the value of nominal clearance between the outer wall of assembly casing and inner wall of the test channel tube.

Maximum dynamic stresses in the fuel element can are equal to σ_{max}= 9 kg/cm^2 at basic frequency f = 40$\pm$ 5Hz and do not depend upon water flow through the assembly. They may occur due to mechanical disturbance of the stand body during its operation.

To determine the hydromechanical resistance of the modificated compensating assembly with casing thickness δ = 2.1mm four full-scale assemblies(two fuel ones and two modificated compensating ones) have been tested for operational life in coolant flow having parameters close to the ones of NPS conditions. Test duration of the modificated compensating assembly was 6790 and 9090 hours; for this period

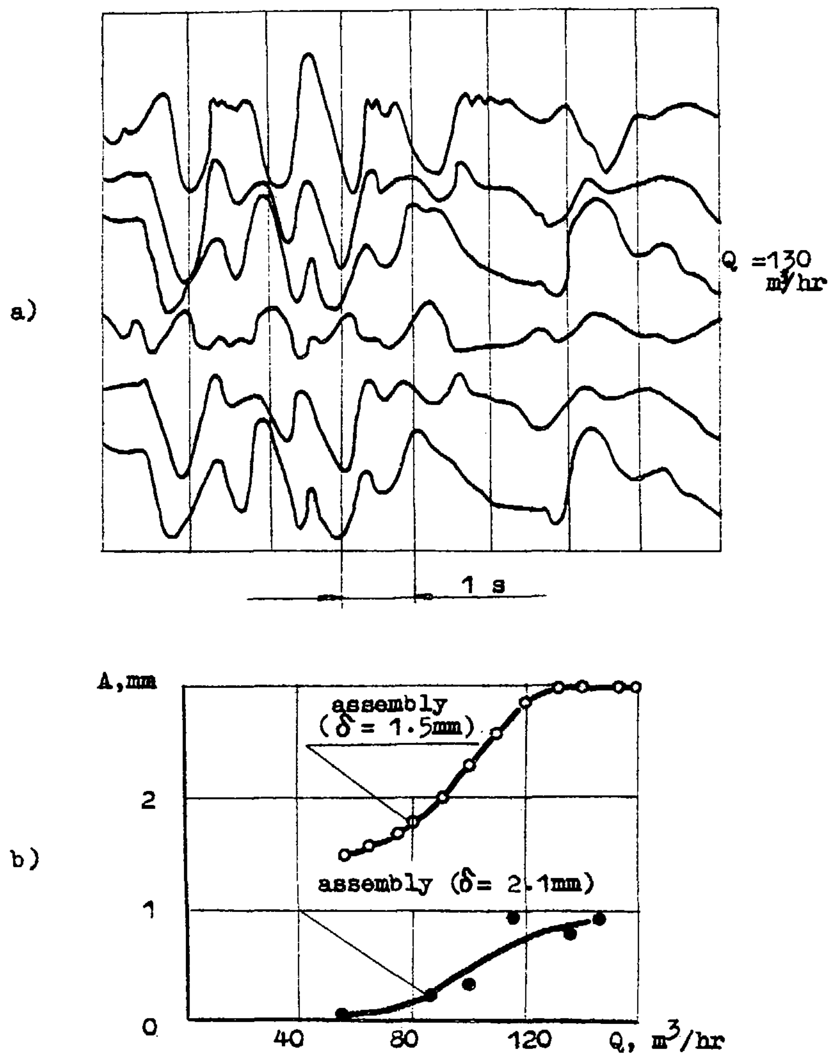

Fig. 2. Typical oscillograms (a) and maximum dynamic displacements (b) of assembly faces in the test channel.

there were 5220 and 3487 double travels as well as 246 and 681 rejections, respectively. The total test duration of fuel assemblies was 9718 and 6619 hours . After operational-life proof all assemblies remained leak-tight and operable.

Conclusion on the long-term operational ability of the modificated compensating assemblies with hexahedral casing of thickness δ = 2.1mm for serial reactors WWR-440 with nominal water flow through the assembly Q= 125 m^3/hr has been made on the basis of the operational-life proof and vibrational tests carried out.

Currently, the program of hydrodynamic and vibration measurements as well as fretting test of fuel elements is still in progress.

Fig.3 shows instrumentation of the fuel elements having the hexahedral zirconium casing with measuring rods having two-component accelerometers inside, and typical oscillograms of fuel element vibrations induced by the longitudinal coolant flow.

On determining the main vibration characteristics (natural vibration frequencies and modes, damping,dependence of dynamic stresses , displacements and amplitudes of vibrations on coolant flow rate) the model fretting tests as per scheme given in Fig.4 are supposed to be carried out. During tests the following values are to be measured: amplitude of casing vibration with respect to spacing grid, frequencies of vibration and acceleration of casing and grid; time of contact between casing and grid faces: temperature, pressure and flow rate of coolant. The model developed gives the possibility to induce vibration over the wide amplitude-frequency range with the help of electromagnetic vibrators or pressure pulser. Approach to the problem of core elements fretting is much the same as the approach described in papers [6-8] . As for the full-scale structures the aspects of influence of coolant transverse leaks and some design solutions on vibration and fretting degree are of great importance. For instance, as a result of long-term operational-life proof of different design versions for fuel elements spacing within the bundle, the maximum allowable length of the span between spacing grids which has been chosen to be 250mm for WWR-440 reactors is the most important factor of vibration resistance[2]. As it was described in [9] , at deviation of flow inlet conditions from the symmetry the resistant eddy cord resulting in the intensive pressure fluctuations, rod wear and outer tube

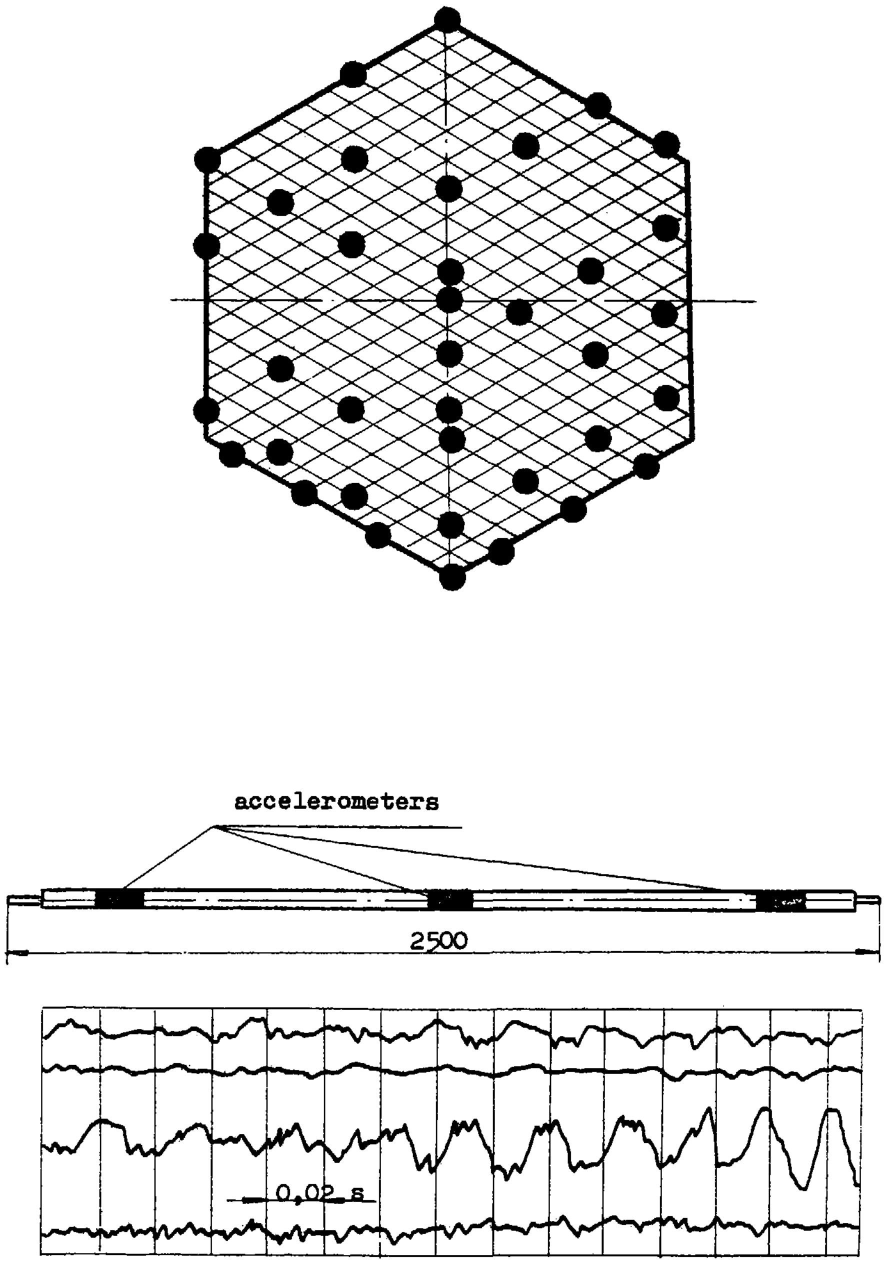

Fig.3. Lay-out of accelerometers within the measuring rod and fuel assembly and typical vibration oscillograms.

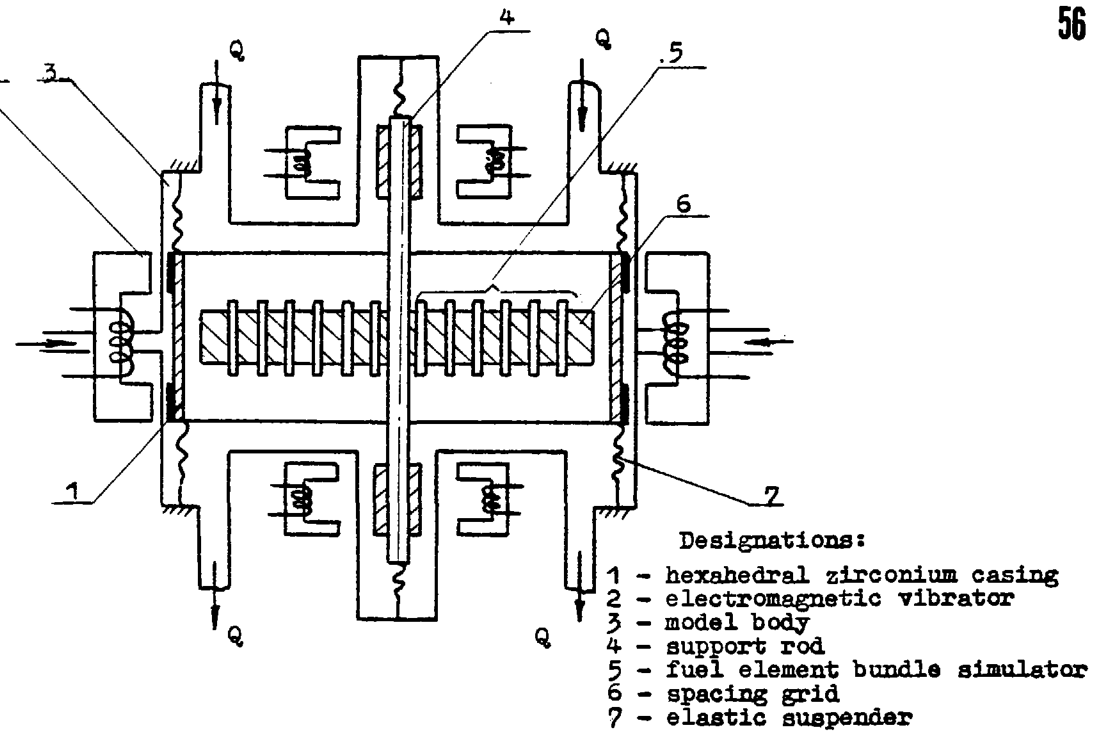

Fig.4. Scheme of model for fretting tests.

wall deterioration is formed instead of the developed turbulent flow. To stabilize flow during its non-symmetrical inlet and to eliminate the excessive rod vibration and fracture it is enough to install the small plate at the point of rod fixing (at the nucleus of eddy cord).

It is necessary to note that phenomena of vibration wear and fretting are very complicated and inadequately studied, and structures in which these processes become apparent are of great variety and therefore there is no possibility at present time to obtain the general solution of the problem and it is reasonable to carry out the wide programs of investigations directed not only to solving the problems for separate structures but to developing the experimental and design methods of more general nature as well.

3. VIBRATION OF TUBE BUNDLES

Substantiation of vibration strength of steam generator tubes is carried out according to norms for investigations and vibration tests of models and full-scale tube bundles of the experimental, first and serial heat-exchanging apparatus. Norms set up the main parameters: geometric, hydrodynamic and mechanical ones which determine the tube bundle vibration. Experimental and calculation methods, requirements for simulation and for programs of investigation at different stages of design, fabrication and operation of steam generator at NPS as well as vibration strength and acceptance criteria are included into these norms. The norms have been worked out with consideration of test results, some of which are given in papers [10,11] and shall be developed upon completion of the investigation programs available. The availability of such norms allows the program to be reasonably limited and investigation methods unified that results in accumulation and systematization of the necessary information. Scheme of investigating the steam generator tube vibration strength is shown in Fig.5. When conducting vibration investigations one should determine dynamic stresses σ_g , amplitude of vibrations A , accelerations a and frequency spectrum Δf of tube versus changes of coolant flow velocity, geometry of tubes arrangement within the tube bundle, the number and construction of spacing grids, inlet and outlet sections configuration etc. Vibration characteristics of tubes are determined by experiment in the only case when it is impossible to calculate them with error less $\pm$ 20%. For the tube bundles of the first steam generators one should obligatory check all the investigated vibration characteristics on the full-scale tube bundle during starting-and-adjustment works and compare them with analogous characteristics obtained before. Research programs have been developed for each steam generator type: experimental, first, having prototype or serial one. For the first steam generator tube bundles one should obligatory carry out periodical check of the main vibration characteristics during operation. For the tube bundle having prototype such check is not obligatory, and when measuring at the stage of starting-and-adjustment works it is enough to check the main vibration characteristics and their correspondence with the results obtained before on the tube bundles of the first heat-exchanging apparatus.

If at the stage of final design of the tube bundle having prototype all vibration characteristics have been measured on the full-scale tube bundle and their correspondence with analogous vibration characteristics of the first steam generator tube bundle has been obtained ,then measurements of vibration parameters for this tube bundle during NPS starting-and-adjustment works are not obligatory.

Vibration test program is considered to be fulfilled after developing the vibration characteristics of model or the full-scale tube bundle to design values.

Vibration investigation program is considered to be fulfilled after the first steam generator has accrued the preset operational life without accidents caused by excessive vibration of tube bundles and the allowable norms of vibration for the tube bundle type under consideration have been set up.

Vibration investigation carried out makes the use of tube bundle structure having passed the test obligatory for the analogous heat-exchanging apparatus and does not allow transition to new structures of tube bundles without substantiation at the stage of technical design.

It is reasonable to consider in details some aspects of interest when substantiating vibration strength of the steam generator tubes.

3.1. Estimation of probability of the tube excessive vibration

It is recommended [12-14] that to eliminate vibration problem in the steam generator structure one should perform the appropriate tube vibration basic frequency shift from frequencies of disturbing forces caused by:

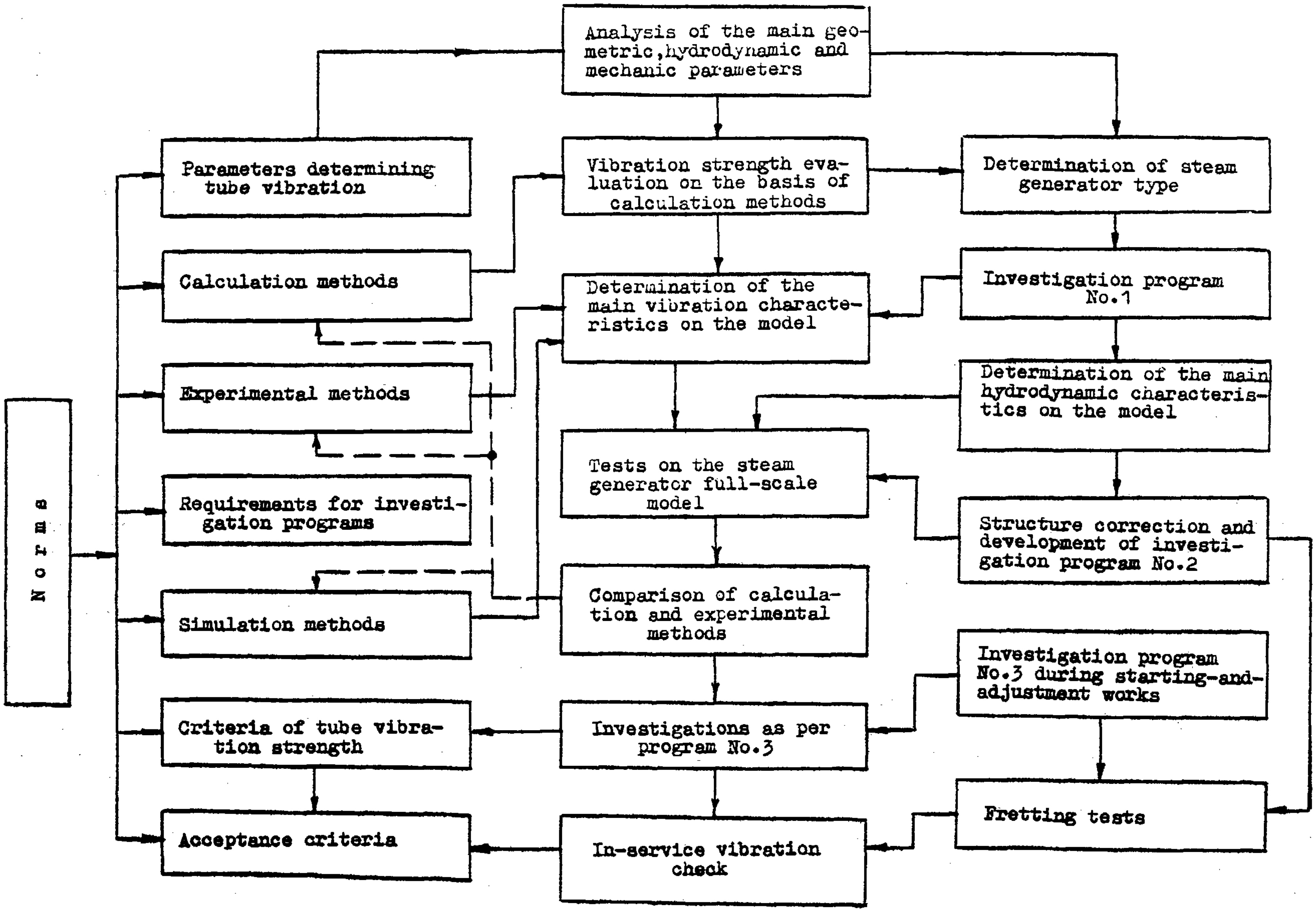

Fig.5. Scheme of investigation of steam generator tube vibration strength

a) stall of Karman vortexes;

b) hydroelastic forces;

c) flow turbulence;

d) acoustic coolant vibration in the circulation loop.

As for real steam generator structures with sodium surrounding tubes only the first two factors are of essential danger then to eliminate excessive tube vibration the following conditions shall be satisfied:

$$f_1 \geq K_1 \frac{Sh \cdot V_{max}}{d} \quad \ldots (1),$$

$$f_1 \geq K_2 \left(\frac{\rho}{m \cdot \delta_K} \right)^{\frac{1}{2}} \quad \ldots (2),$$

where

f_1 - basic frequency of the tube natural vibration with regard for the associated coolant mass, Hz ;

K_1, K_2 - frequency shift factors;

Sh - Strukhal number;

V_{max} - sodium rate in the narrowest clear opening of the bundle, cm/s;

d - tube diameter, cm;

ρ - sodium density , $\frac{g \cdot s^2}{cm^4}$;

m - tube mass and associated sodium mass per unit of tube length, $\frac{g \cdot s^2}{cm^2}$;

δ_K - logarithmic coefficient of vibration damping.

Results obtained by the authors and other investigators [10 -15] permit to propose a generalized diagram of amplitudes of displacements A and accelerations of the steam generator tubes versus mean coolant velocity V at the cross-flow sections (Fig.6) to evaluate the probability of excessive vibration. Here point O_1 corresponds to such flow rate when tube vibration is on the sensitivity threshold of the available vibration measuring means. At point A tube vibration level begins to rize non-linearly(with regard for the vibration measuring error) with increasing the coolant flow rate, and at point B vibration amplitude is 30% higher than at point B_1 lying on the continuation of the straight section O_1A. In the most investigations subsequent flow rate V increase results in a continuous growth of vibration in the direction of points C and D, but in some tube bundle structures and flow patterns [13,14] that on the characteristic curve A, $a = \varphi(V)$ there are pronounced maximum at point C and minimum at point E with sharp increase of vibration amplitude towards point F. In the opinion of the investigators, the vibration amplitude peak at point C corresponds to resonance vibration caused by stall of Karman vortexes, see formula (1), while vibration increase at point F is accounted for by hydroelastic forces, formula (2).

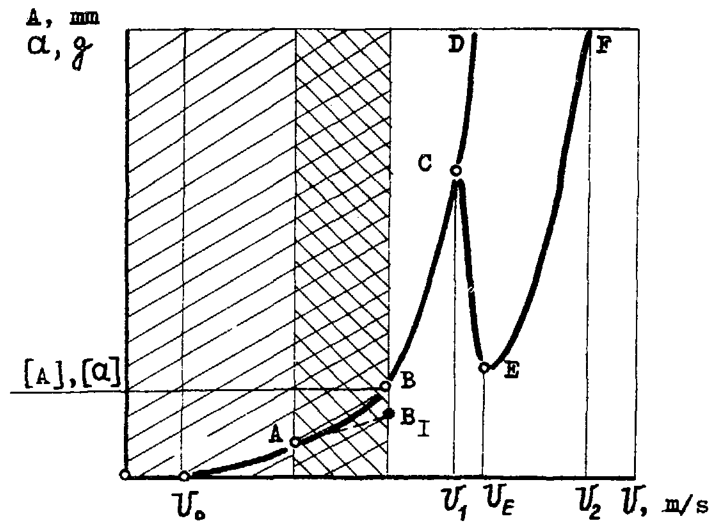

Fig.6. Generalized diagram of amplitudes of displacements A and accelerations a of the steam generator tubes versus mean coolant rate V.

Results analysis of investigating the steam generator tubes vibration and its operating experience permits to make the following conclusions:

a) If one assumes $K_1 \approx K_2$, then for the majority of real structures $\frac{V_2}{V_1}$= 3-5.

b) Section BCEF of the characteristic curve $A, a=\varphi(V)$ is the most interesting and difficult one for the investigators, but at the rates of $V_1 > [V]$ operation of the NPS steam generators is not allowed.

c) Section OA corresponds to the steam generator unlimited operational life from the standpoint of tube vibration strength.

d) It is reasonable to establish the allowable norms of vibration $[A]$ and $[a]$ according to the location of point B on the characteristic curve $A, a=\varphi(V)$.

3.2. Vibration and vibroacoustic check of tube vibrations.

With growth of the number of operating NPS and heat exchangers having accrued 100.000 hours under the normal operating conditions , questions of vibration and vibroacoustic check become a matter of great urgency. Tubes vibration measuring methods suitable for vibration check at a NPS are described in detail in [16] ,and the results of vibration check of BN-350 NPS steam superheater tubes in operation during 1973-1976 - in [10] .

Combination of the vibration and vibroacoustic check is the most effective . For example, during operational-life proof of the seven-tube model of BN-350 steam superheaters and maintaining similarity of the main parameters(vibration acceleration amplitude and vibration frequency of tubes, tubes and spacing grids geometry, materials, sodium temperature) with the NPS parameters, periodical vibration and vibroacoustic measurements followed by cutting of the models and metallographic examinations [17] have been carried out. The tests have revealed that just those tubes in which normalized autocorrelation functions $R(a)$ and spectral densities $S(a)$ of accelerations were considerably changing during operational life , Fig. 7 , are worn out at points of tube contact with spacing grid. During these tests amplitude-frequency spectrum was changing according to readings of the accelerometers mounted on the body. Influence of the tube vibration level upon the amplitude-frequency spectrum of the accelerometers mounted on the heat exchanger model body is mostly evident from Fig. 8 representing characteristic oscillograms (a) and spectrograms (b) of tube vibration accelerations according to the vibration measuring results, and spectrogram (c) of readings of the outside accelerometers for vibration acoustic check. When Q= 30 m^3/hr minor vibration of model tubes having basic frequency f =50±5 Hz(see

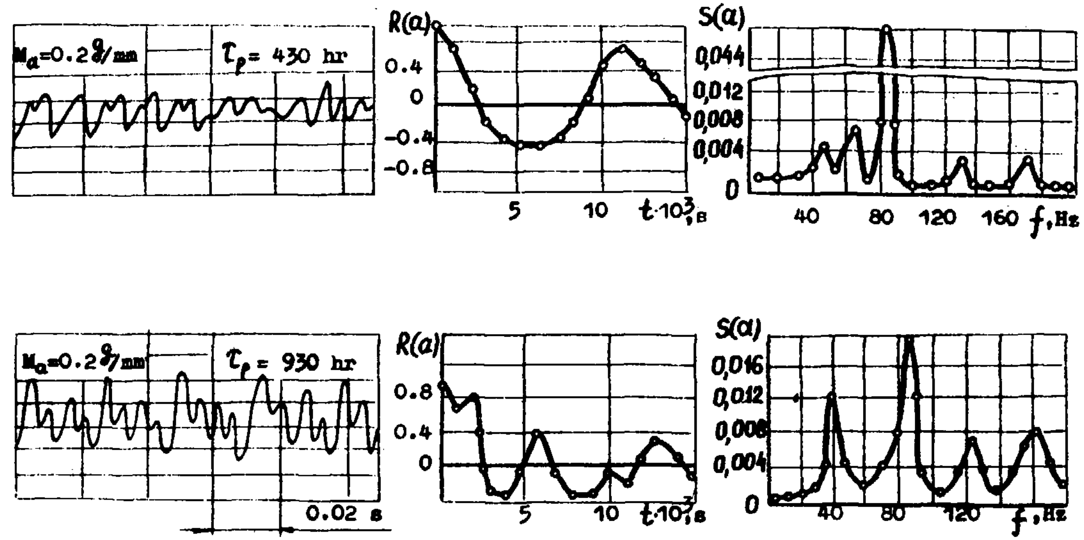

Fig.7. Typical oscillograms, diagrams of normalized autocorrelation functions $R(a)$ and spectral density $S(a)$ of tube acceleration amplitudes at the level of the 1-st spacing grid in BN-350 steam generator model at Q = 140m^3/hr for various times of operational life.

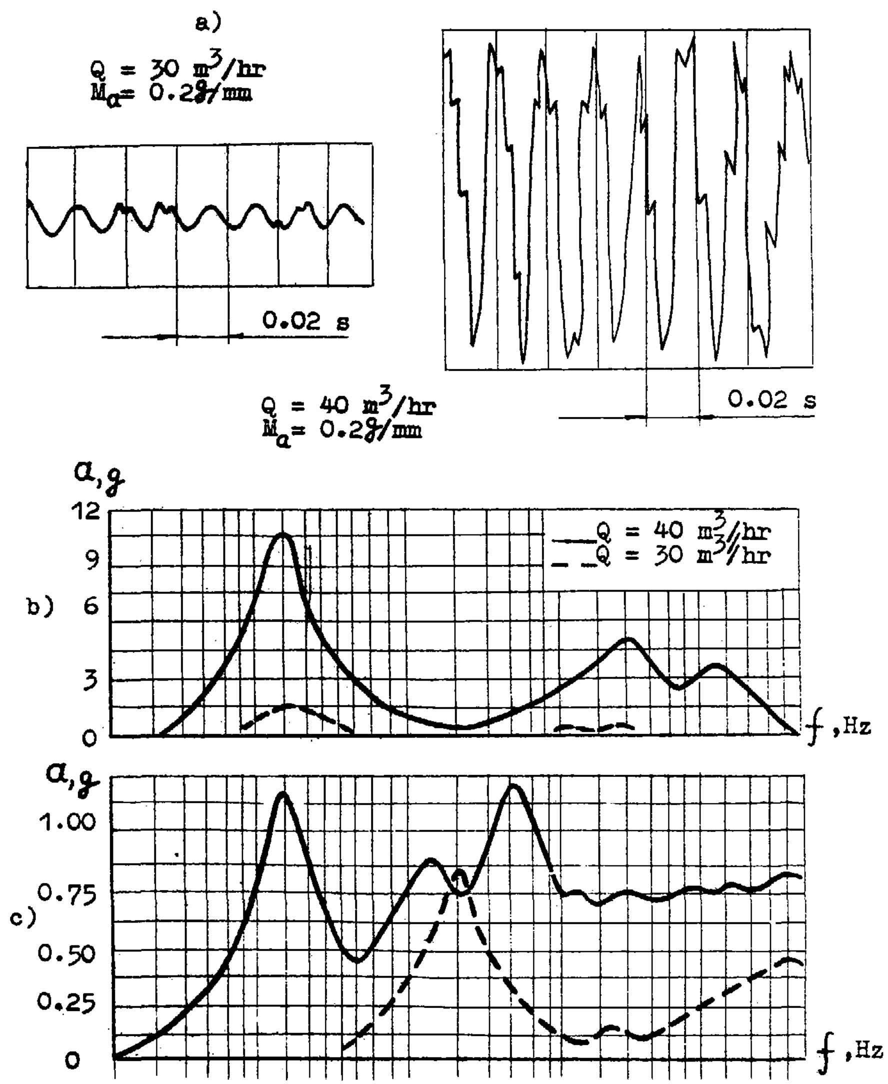

Fig.8. Oscillograms (a) and spectrograms (b) of tube acceleration and spectrogram (c) of vibroacoustic noises at flow rate Q.

Fig.8b) is not revealed in the spectrograms (see Fig.8c) of the outside accelerometers, but as flow rate runs up to Q=40m^3/hr and tube vibration considerably increases there is a sharp peak in the spectrogram (Fig.8c) over the narrow range around the basic natural frequency of model tube vibrations.

Results obtained are used to develop systems for vibration and vibroacoustic check of tubes for full-scale steam generators.

3.3. Some measures reducing tube vibration.

As is evident from formulae (1) and (2) one of the tubes main vibrocharacteristics upon which the steam generator operational life depends is natural frequency f of tubes vibration. Value f may be increased through installation of an additional spacing grid or decreasing of the gap between the tube and the spacing grid cell. Effectiveness of these structural measures has been checked by experiment on a model and an experimental full-scale heat exchanger. As the result of installation of an additional spacing grid in the middle of span in the multi-tube model at cross-flow section the vibration maximum amplitude has sharply reduced, Fig.9. Detailed description of the model and test conditions is given in [18] . Considerable reduction of the vibroaccelerations maximum amplitude also takes place if the tubes are expanded over the inner surface at the level of the spacing grids, Fig.10 . More detailed information is provided in [11].

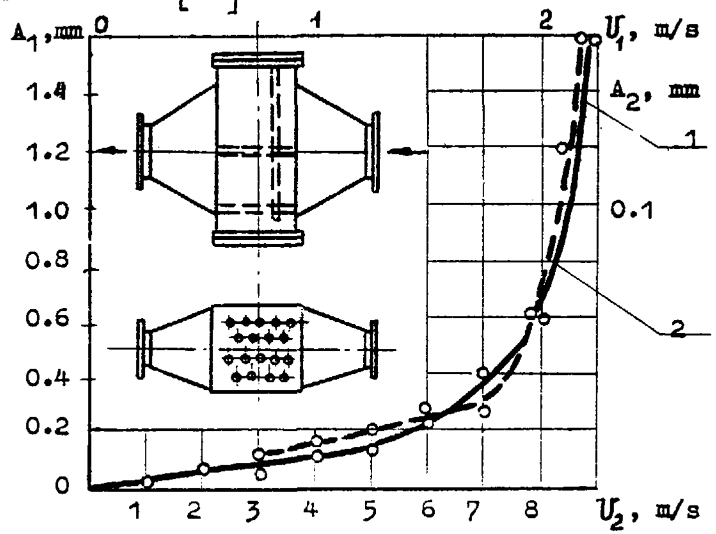

Fig.9. Maximum amplitude of vibration A versus water cross-flow rate:

1 - with one spacing grid;
2 - with additional grid in the middle of span.

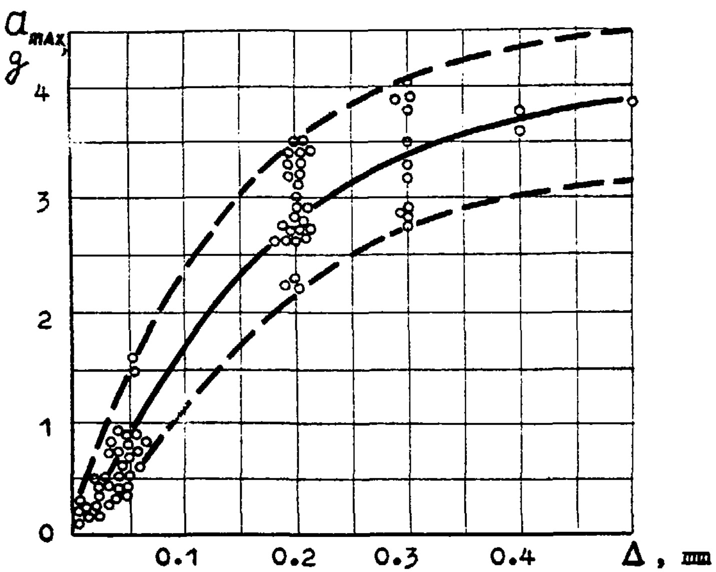

Fig.10. Maximum amplitude of the heat exchanger tube accelerations versus the value of radial gaps in the spacing grid.

4. CONCLUSION

To substantiate vibration strength of the reactor and steam generator internals it is necessary to further develop purposeful programs of theoretical and experimental investigations that would help improve available design methods and allowable vibration norms and develop new ones.

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