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SICODYN international benchmark on dynamic analysis of structure assemblies: variability and numerical-experimental correlation on an industrial pump

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Abstract – A benchmark is organised to quantify the variability relative to structure dynamics computations. The chosen demonstrator is a pump in service in thermal central units, which is an engineered system with not well-known parameters, considered in its work environment. The blind modal characterisation of the separate pump components shows a 5%–12% variability on eigenfrequency values and a less than 15% frequency error in comparison with experimental values. The numerical-experimental MAC numbers reach 0.7 at the maximum, even after updating. An example of modal results on the pump assembly fixed is presented, which shows a larger discrepancy with measurement values, essentially due to the modelling of the interfaces and boundary condition, and to the possible simplification of the main components F.E. models to reduce their size. Though a significant frequency error, the first overall modes are correctly identified. If this tendency can be confirmed from all the participants' results, the conclusion to be drawn is that, if the predictive capability of F.E. models to represent the dynamical behaviour of sub-structures is satisfactory, the one relative to structures that are built-up of several components does not allow their confident use. Additional information issued from measurements is needed to improve their accuracy.

Key words: Benchmark / structural dynamics / modal analysis / finite-elements / blind comparison / numerical-experimental correlation / numerical variability / uncertainty / model validation / error estimation

Résumé – Benchmark international SICODYN sur l'analyse dynamique des structures assemblées : variabilité et corrélation numérique-expérimentale sur une pompe industrielle. Un Benchmark est organisé afin de quantifier la variabilité relative à des simulations en dynamique des structures. Le démonstrateur choisi est une pompe en service dans des centrales thermiques, structure industrielle considérée dans son environnement et dont tous les paramètres ne sont pas parfaitement connus. La caractérisation modale en aveugle des composants séparés montre une variabilité des fréquences propres entre 5 % et 12 % et un écart fréquentiel avec les mesures inférieur à 15 %. Les nombres de MAC numérique-expérimental valent au mieux 0,7, même après recalage. Un exemple de résultats modaux sur la pompe assemblée enchâssée dans le béton est présenté, montrant un écart plus marqué avec la mesure, due essentiellement à la modélisation des interfaces et de la condition à la limite, ainsi qu'à la simplification éventuelle des modèles des principaux composants en vue de réduire leur taille. Malgré un écart fréquentiel non négligeable, les premiers modes d'ensemble sont correctement identifiés. Si cette tendance est confirmée avec les résultats complets des participants, l'enseignement à tirer est que, si la prédictibilité du comportement dynamique par des modèles éléments-finis se montre relativement satisfaisante pour des structures d'un seul tenant, celle concernant les structures constituées d'un assemblage de pièces distinctes se révèle insuffisante et nécessite la prise en compte d'informations supplémentaires issues de la mesure.

Mots clés : Benchmark / dynamique des structures / analyse modale / éléments-finis / comparaison en aveugle / corrélation numérique-expérimentale / variabilité numérique / incertitude / validation de modèle / estimation d'erreur

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1 Introduction

1.1 Context

A main objective of industrial companies is to quantify the confidence they have in numerical models used either in design purpose or in expertise purpose. The systems of interest include proposed or existing systems that operate at design conditions, at off-design conditions and at failure-mode conditions that apply in accident scenarios. In particular, the dynamical behaviour of engineered systems that equip power plants must be confidently predicted. The numerical models built to do so in a design purpose must be able to represent the characteristics of the structure itself, its coupling with its environment, the usually unknown excitations and the corresponding error sources and uncertainties; in an expertise purpose, where measurements can be carried out on the existing structure, the numerical models are generally generic and must be able to reproduce the behaviour of the whole family of nominally-identical structures: their accuracy can be improved using measurement results so that they can be used for instance to predict the effects of a structural modification.

1.2 SICODYN benchmark purpose

Within this objective and according with the fact that "our confidence in the process of *accurate* prediction does require benchmarks" [1], it was decided to organise a dedicated international benchmark in order to quantify the a posteriori variability of a dynamical simulation, by comparing simulation results obtained by different participants in blind conditions and by comparing simulation and experimental results. A benchmark is usually based on "a choice of information that is believed to be accurate or true for use in verification, validation or calibration" [1]. It is well known that *Verification* (ASC) is the process of confirming that a computer code correctly implements the algorithms that were intended, *Validation* (ASC) is the process of confirming that the predictions of a code adequately represent measured physical phenomena [1] and *Calibration* denotes activities to optimize model parameters when code results are compared with experimental measurements [2]. Though verification, validation and calibration are parts of the proposed benchmark, the intended purpose cannot strictly be defined in these terms and so be classified using the often used benchmark formalism: SICODYN benchmark purpose is clearly to quantify, from an *industrial* point of view, that is in the real conditions of an industrial study, the variability of computational blind results, the discrepancy between numerical and measured results, and the improvement of this discrepancy using measurement information.

1.3 SICODYN benchmark definition

Considering this particular characteristic of the proposed benchmark, the constitutive parts of it have been so defined:

(1) definition and search of a target engineered dynamical system;
(2) choice of information (input and output data) to be accurate and representative of industrial data;
(3) definition of methods of comparing computational results and computational and measured results;
(4) logical procedures for drawing conclusions from these comparisons.

2 Definition of the target dynamical structure

2.1 What type of demonstrator?

Most of benchmarks are based on academic tests, as those used in The Validation Challenge Workshop, 22–23 May 2006, Albuquerque, New Mexico, organised by Sandia National Laboratories [3–6] or the "Benchmark study on reliability estimation in higher dimensions of structural systems" [7]. No experimental measurements are available, but the validation environment is represented by the variation of input parameters. It seemed that this type of benchmark, although useful in a verification purpose, was not able to answer our industrial aim.

Another possible type can be based on a laboratory device, like GARTEUR [8] or EDF-CEA SMART benchmark [9, 10], that considers a simplification of an industrial structure regarding the geometry, the physical and the environmental complexity and builds a dedicated structure with well-known parameters. After studying the conclusions that can be obtained using a test device instead of a complex structure, it seemed that this type of controlled structure is not able to represent the industrial problematic. This fact is well demonstrated in a previous study carried out by Sulzer [11], where experimental and numerical modal analyses on a pump are compared. Before considering the full barrel pump, methods were tested on a test device similar to the drive-end of the barrel pump. It could be shown that an accurate representation of the modal behaviour could be achieved using a finite element model on the test device with known boundary conditions. On the contrary, difficulties still remain to obtain a confident model of the full structure. In particular the variability relative to the boundary conditions, as a result of a numerical sensitivity analysis, can be large. Furthermore problems due to the large size of the whole finite element model and to the too low number of measurement points have been pointed out, so that the correlation process could not be optimised.

Considering the fact that clearly the choice of a demonstrator must be well adapted to the intended purpose of the benchmark, we decided to use a real industrial engineered dynamical system. The requirements for the demonstrator are:

- industrial (or quasi) complex structure;
- assembly of several sub-structures (bolted, with rivets...);
- considered in its environment;
- excitation possibly unknown;

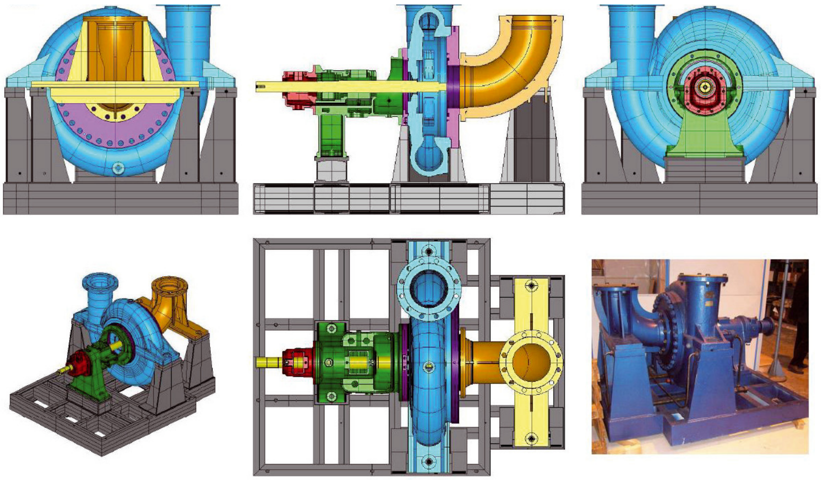

Fig. 1. The booster SULZER mono-stage horizontal pump.

- availability of plans with marked dimensions or computer-aided design (CAD), authorisation to disclose them to participants;
- possibility to measure in stationary or in operating conditions;
- possibility to partially dismount and reassembly the structure;
- limited time to elaborate the finite element model.

The chosen equipment is a pump used in EDF thermal units (Fig. 1). It is a one-stage booster pump, composed of a diffuser and a volute, with axial suction and vertical delivery (body with volute called "snail"), mounted on a metallic frame. It has been designed forty years ago by Sulzer Pumps.

The principal characteristics of the pump are:

- the dimensions of the pump without support are approximately 1 m × 1 m;
- the mass of pump and support is approximately 2800 kg;
- the material of the pump is cast iron and steel;
- the nominal rotational speed is 1500 rpm;
- the shaft is supported in radial direction by two plain bearings (unknown stiffness and damping linearised characteristics) and in axial direction by a roller.
- the impellor and shaft are connected by keys.

2.2 The industrial context of considering a pump

For EDF, and equivalently for oil or water industries, demands relative to this type of equipment are more and more precise, as far as technical requirements and operating reliability are concerned.

Concerning EPR design, EDF introduced additional technical specifications for pumps. Until now, only a minimal gap between the nominal rotating speed and the critical speeds was demanded. Now it is in addition required to assure, for certain types of pumps, a minimal gap between the nominal rotating speed and the resonances of the statoric parts of the machine. If the theoretical determination of the shaft critical speeds is well-known for pump manufacturers, the predictive determination of the resonances of the statoric parts is not part of the usual know-how. As the manufacturers will probably justify the corresponding values using simulations, EDF must have a precise idea of the reliability of such calculations.

3 Definition of accurate information

3.1 A step by step approach

3.1.1 Hierarchical process related to geometrical and environmental complexity

Though validation is not our main purpose, we adopt a fundamental, as well as practical, aspect of it in a real engineering environment, which is the construct of a validation hierarchy (AIAA 1998, ASME 2006) [4]. This approach divides the complex engineering system of interest into progressively simpler levels of complexity (tiers): subsystem cases, benchmark cases and unit problems. The strategy in the tiered approach is to assess how accurately the computational responses compare themselves and with the experimental responses at multiple levels of physics coupling and geometric complexity. Importantly, each comparison of computational responses and experimental responses in a validation hierarchy allows an inference of model accuracy to be made relative to the tiers that are immediately above and below the tiers where the comparison is made [2].

Considering that the Booster pump, in operating conditions as it is used in a production energy unit, is an assembly of different components in a complex environment (frame fixed in concrete, pump connected to pipes and another pump), the following division into increasing levels of complexity has been adopted (Fig. 2):

- increasing the geometrical and physical complexity level, 3 possible steps:

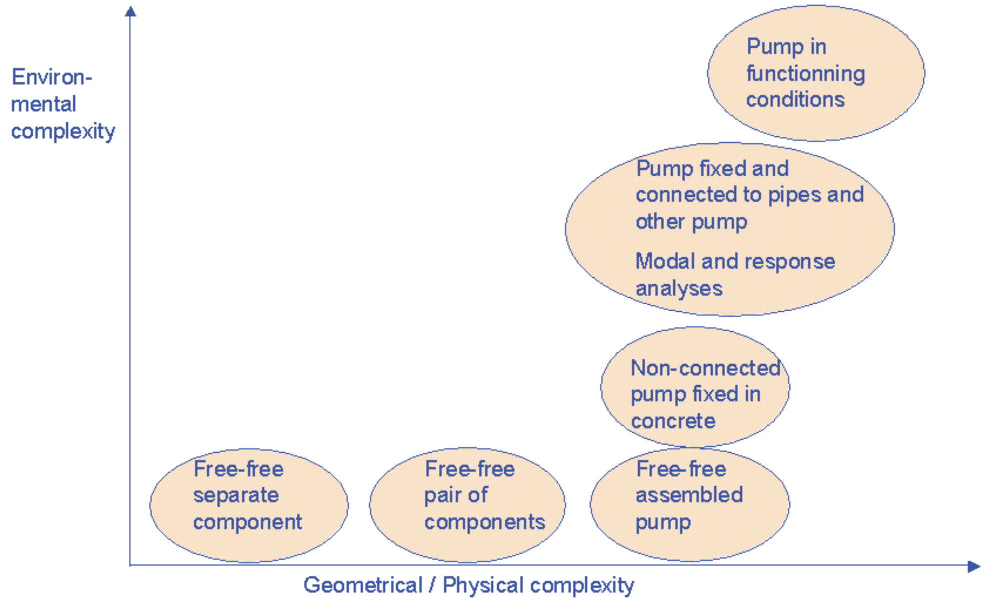

Fig. 2. The hierarchical process.

   ◦ modal characterisation of free-free separate component;
   ◦ modal characterisation of free-free pair of components (to validate the assembly representation, particularly, the interface modelling);
   ◦ modal characterisation of free-free assembled pump;
- increasing the environmental complexity, 3 possible steps:
   ◦ modal characterisation of the non-connected assembled pump fixed in concrete;
   ◦ modal characterisation and response analysis to a known excitation of the non-rotating pump fixed in concrete and connected to pipes and to the other pump;
   ◦ response analysis of the pump in operating conditions.

In this paper, will only be considered the two steps that are the modal analysis of the free-free separate components and of the non-connected assembled pump fixed in concrete.

3.1.2 Blind comparisons and updated procedures

At each step of that hierarchical process, blind comparisons between results obtained by the participants and blind correlations with experimental measurements that are not known beforehand are performed. Then a parametrical or sensitivity study and a possible updating procedure using measurement results can be carried out to improve the correlation of the numerical results with the reference data.

3.2 Input, output and reference benchmark data

3.2.1 Input data

Input benchmark data provided by EDF are paper plans of the assembled pump and its parts and CAD models of the 8 main pump components (Fig. 3). CAD models were elaborated using Salome_Meca free software [12]. Experimental modal analyses on three components (shaft and impeller, pump casing and bearing support) and on the assembled pump fixed in concrete have been performed by EDF R&D and are made available for the participants when they have given corresponding blind modal results.

Material characteristic values (density, Young's modulus and Poisson's ratio) are not part of input data, as it usually is in an industrial dynamics study: participants choose themselves the most appropriate values, taking into account the information available from the manufacturer on material type. This original approach, which is evidently unusual in validation benchmarks, aims to reproduce the real conditions of a dynamics simulation in an engineering context.

3.2.2 Output data

Output data essentially are eigenfrequency values, mode shapes and modal displacements at measurement points. The analysis is performed under linear assumptions.

3.2.3 Reference data

Experimental modal analyses have been performed by EDF R&D on the shaft and impeller system, the bearing

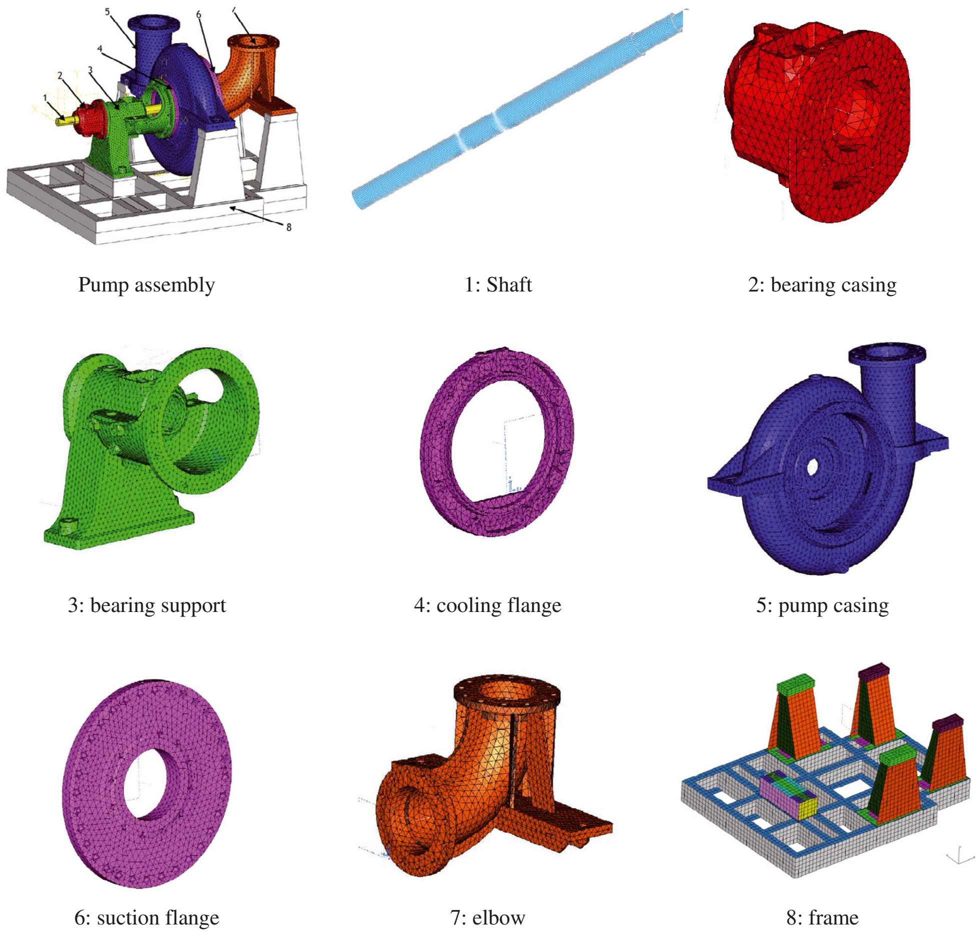

Fig. 3. Examples of component meshes based on 3D CAD models.

support and the pump casing in free-free conditions, and on the assembled pump fixed in concrete.

The experimental mesh relative to the assembled pump contains 55 measurements points (Fig. 4a). The pump was excited using an impact hammer in 3 distinct points: a point located at the bearing casing, orthoradial horizontal direction, a point at the bearing support in vertical direction and a point on one of the poles which support the pump casing. An example of frequency response is given in Figure 4b. 5 mean operations have been carried out for each frequency response. The sampling frequency is 4096 Hz.

4 Methodologies, computer codes and metrics

4.1 Methodologies applied by participants

If we gather the methodologies used by participants, we can distinguish the following steps, which are largely inspired of V&V (Verification and Validation) process defined by ASME [2]:

- elaborating detailed meshes of most of the main components, so that the geometrical complexity of the component can be taken into account;
- convergence analysis relative to meshes;
- parametrical or sensitivity analysis on the material characteristics on components for which measurement data are available;
- optimisation of the measurement process from a non validated component numerical model;
- updating the material characteristics for those components to improve the numerical-experimental eigenfrequency and modeshape agreement;
- reduction of the model size of the components in view of limiting the number of d.o.f. of the whole structure model;
- defining adequate interfaces between components and assembling the pump models, considering two different boundary conditions: free-free and fixed in concrete;
- parametrical study on interface and boundary condition representations;
- updating the interface and boundary conditions using measurement data.

It can be noticed that no participant tried to identify which of the various components that make up the complete structure have most influence on the whole-structure dynamics. No participant used the component mode synthesis method for the computation of the pump assembly.

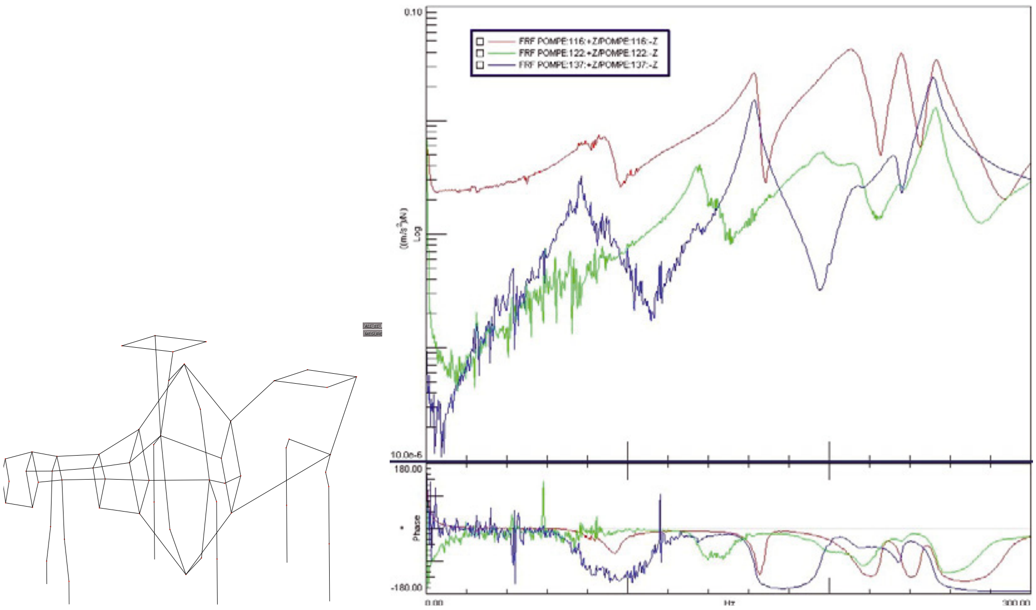

Fig. 4. Measured data on the pump assembly fixed in concrete, (a) experimental mesh, (b) frequency response.

Table 1. Mean eigenfrequency and mean gap values for each pump component.

	1 – shaft		2 – bearing casing		3 – bearing support		4 – cooling flange		5 – pump casing		6 – suction flange		7 – elbow		8 – frame	
Material	steel		cast iron		cast iron		cast iron		steel		steel		steel		steel	
Mode	Mean freq. (Hz)	Mean gap (%)	Mean freq. (Hz)	Mean gap (%)	Mean freq. (Hz)	Mean gap (%)	Mean freq. (Hz)	Mean gap (%)	Mean freq. (Hz)	Mean gap (%)	Mean freq. (Hz)	Mean gap (%)	Mean freq. (Hz)	Mean gap (%)	Mean freq. (Hz)	Mean gap (%)
1	244	1.0	1092	10.7	251	6.8	305	12.6	434	3.7	566	1.8	296	4.6	45	7.4
2	244	1.0	1522	10.0	312	7.2	310	12.6	487	3.5	566	1.8	477	2.2	106	2.5
3	627	1.3	2008	8.6	457	7.1	752	10.0	620	4.8	1029	1.9	499	3.1	114	1.7
4	628	1.2	2201	9.2	465	7.4	774	10.0	768	2.9	1356	2.0	557	2.1	122	6.4
5	1172	2.0	2437	8.5	486	5.9	860	12.0	890	3.1	1358	2.0	658	2.9	151	2.6
6	1180	1.7	2463	9.0	626	5.9	876	11.9	927	3.2	1717	1.5	690	3.0	158	2.5
7	1464	0.7	2570	8.1	643	6.1	1106	10.8	1193	3.0	1717	1.7	702	2.9	158	3.3
8	1847	2.7	2657	9.4	751	6.0	1369	12.6	1235	3.1	1811	2.3	776	2.8	163	2.8
9	1875	3.2	2884	8.0	773	7.2	1503	11.7	1255	3.4	1831	2.4	830	1.5	167	0.7
10	2211	1.2	2953	8.2	906	8.0	1671	10.7	1279	3.6	2247	3.0	879	4.8	198	2.8

4.2 Computer codes used by participants

The computer platforms and codes used by participants are Catia, Ansys, Patran, Nastran, Abaqus, Ideas, Samcef, *Code_Aster*, SALOME_Méca.

4.3 Metrics

Classical metrics (measures of comparison) are used to compare modal results. The frequency error is used to compare participants' numerical results and to compare numerical to experimental results. In addition, the Modal Assurance Criterion [13] is used in view of correlating experimental and computed modeshapes.

More complex metrics could be used in a second phase when uncertainties will be considered both in computed results and experimental data [14, 15].

5 Comparative blind modal results on separate pump components

5.1 Numerical variability of separate component modal analyses

Table 1 shows the mean frequency and mean gap values relative to blind results obtained on the eight main pump components.

The comparison of numerical modal analyses of free-free separate components shows discrepancy values that are not so negligible if we consider the fact that there is no boundary condition variability. The minimal discrepancies on eigenfrequency values concern the elbow and the suction flange. As expected, the variability associated to eigenfrequency values can be in a large part related to the variability associated with the material properties considered by the participants: the steel pump

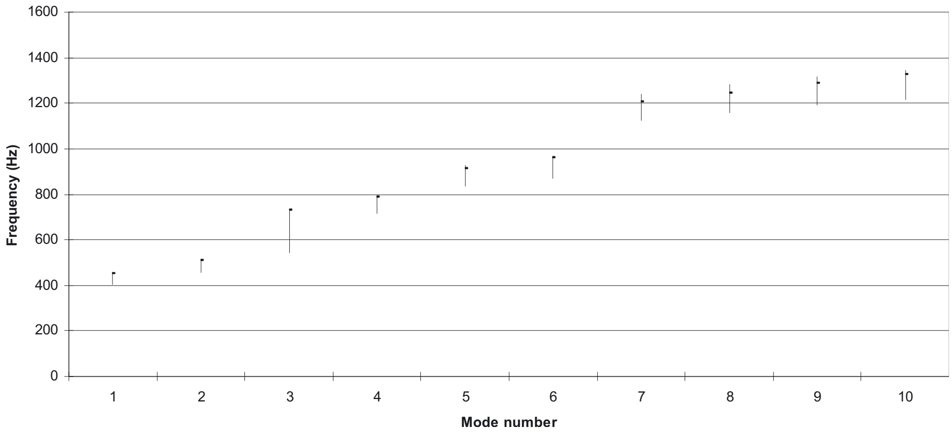

Fig. 5. Pump casing: blind comparison min-max numerical and measured eigenfrequency values (modes 1–10).

components present a mean gap around 5% (to be related with 2% variability on Young's modulus), whereas the cast iron components show a higher mean gap around 11% (to be related with 20% variability on Young's modulus). This high variability on cast iron material property values correspond to different possible shade cast. The frame is the pump component with the lowest eigenfrequencies, even if its lower part is in a concrete slab: it will certainly be more influent on the dynamic behaviour of the assembled structure than the other components. The bearing casing is, on the opposite, the stiffest component, for which an experimental modal analysis will not be necessary.

5.2 Experimental-numerical comparison of separate component modal analyses

In the following, we present the experimental-numerical comparison relatively to one pump component which is the pump casing. The shaft and impeller system modal results are not presented because, due to the lack of impeller CAD model, most of participants used experimental data to identify equivalent mass and stiffness representation of the impeller, which leads to no purely blind results. The pump casing is preferred to the bearing support because it is a main part of the pump and because the pairing of numerical and experimental modes is obvious.

Figure 5 shows the blind comparison between the set of numerical modal results and the experimental modal data relative to the ten first eigenmodes.

Comparing the interval constituted by the minimum and maximal eigenfrequency values obtained by eight participants to the measured corresponding values (Fig. 5), we can notice that every measured frequency is contained within the interval. The prediction accuracy on eigenfrequency values, illustrated in Figure 6, is less than 15% for the majority of modes considered; the numerical values are generally lower than experimental ones, that corresponds to a less stiff behaviour of the substructure. However, examining the MAC numbers on the four first modes among the available results, we can see that the agreement between the modeshapes is not so clear (Fig. 7). The modes 1, 2 and 4 are differently correlated regarding the participant; mode 3 shows the best agreement ($0.6 < \mathrm{MAC} < 0.7$).

Some participants tuned the material property values in order to get the frequency error lower (to correct possible differences between pump plans and realisations), but the modeshapes correlation could not be improved. A participant carried out a specific study which showed that the sensor location was not optimised in view of model updating and could be improved to do so.

A conclusion that can be drawn from this is the ability of numerical models to accurately predict the global modal behaviour of a structure with complex geometry but that can be considered as a structure unit (not composed of different assembled sub-structures), in terms of eigenfrequencies; it seems that the modeshapes prediction presents more difficulties. Further investigations to explain this poor result on MAC number presented by several participants must be conducted; in particular, the quality of the projection of the 3D numerical results on the 20-point experimental mesh must be checked, as it is mentioned in [13] about the abuses of the Modal Assurance Criterion.

6 Assembling the pump components

6.1 Reducing the model size

Two participants performed a specific study in order to reduce the model size of the pump assembly and to avoid possible computing problems or simply to deal with less time-consuming simulations without degrading the results (model reduction or defeaturing process). To do so, fillets, chamfered edges, small holes and ribs have been removed. Figure 8 shows the simplified geometries that can be obtained by defeaturing (simplification) for the suction flange, the cooling flange and the frame.

With reduction of the mesh size that can be important (divided by 10 for the suction flange), it can be seen (Tab. 2) that the frequency error is low, so that simplified

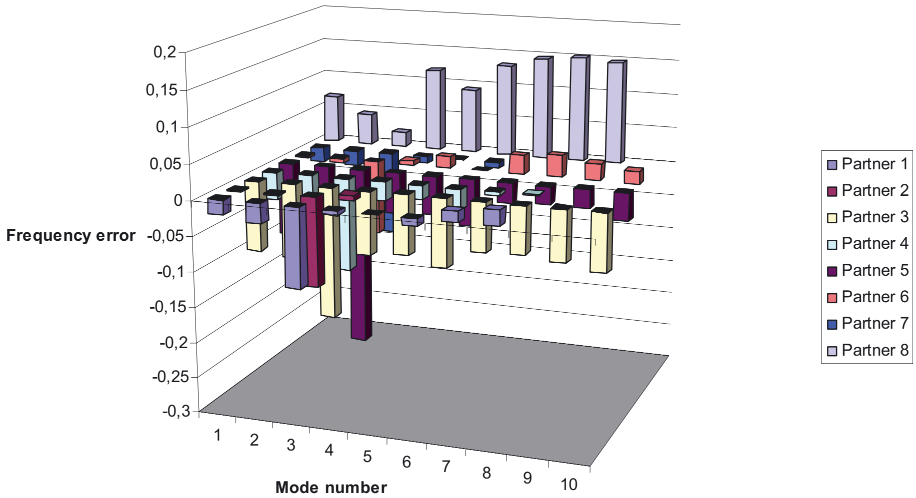

Fig. 6. Pump casing: frequency error (modes 1–10).

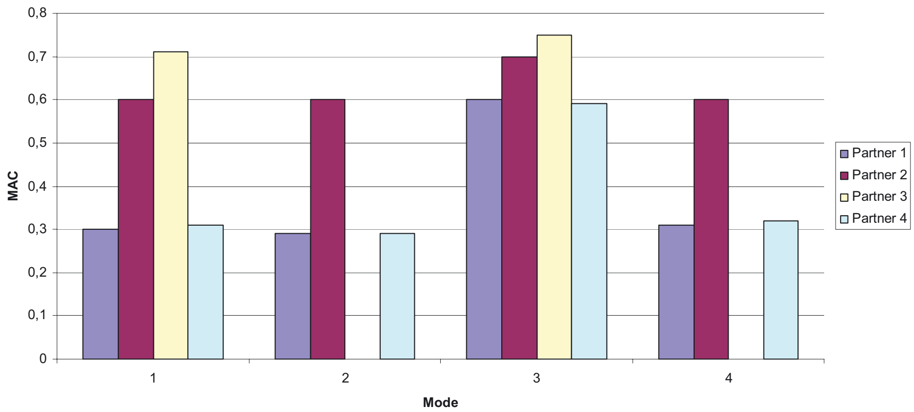

Fig. 7. Pump casing: MAC numbers (modes 1–4).

Table 2. Influence of the defeaturing on the eigenfrequency values.

	Suction flange	Cooling flange	Frame
Mode number	Frequency error (%)	Frequency error (%)	Frequency error (%)
1	–0.7	–1.8	3.3
2	–0.7	–1.5	1.8
3	–1.1	1.7	1.2
4	–0.1	4.2	2.3
5	–0.6	–0.5	2.4

models can be used in assembling the pump components instead of the detailed models.

6.2 Modeling the interfaces and the boundary conditions

6.2.1 Generalities

As it is well-known, the complex behaviour of connecting elements plays an important role in the overall dynamic characteristics, such as natural frequencies, modeshapes, and non-linear response characteristics to external excitations.

Bolted joints and fasteners have a significant effect on the damping and stiffness of the joint. The damping is created by friction on the screw thread, gas pumping, or impact-induced damping in local microgaps between joint surfaces, material damping in the asperities of contact surfaces, and plastic deformation. The stiffness of the joint is affected by the hardness and roughness of contact surfaces. In most cases, these parameters cannot be accurately modelled due to uncertainties in the production, variability in the material properties, geometry parameters, and the relaxation process [16].

In a predictive purpose, conventional design and analysis of structural systems are based on the well-known two extreme idealizations of joints: perfectly rigid (or fully

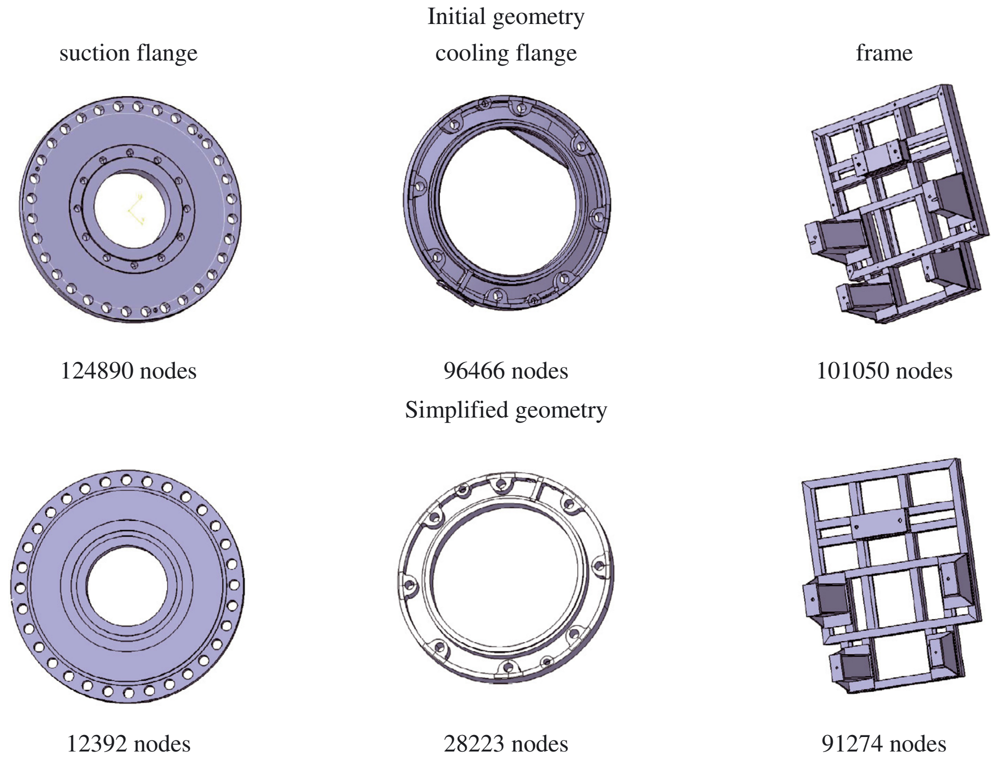

Fig. 8. Initial and simplified geometries for 3 pump components.

restrained) and ideally pinned. The fact is that they do not represent the real structural behaviour as reflected experimentally. More precise is deducing equivalent flexibility and damping properties of joints from detailed FE models [17–20] and introducing additional zero dimension and zero mass components to the structural assembly having these properties. But the load conditions are usually unknown and the elaboration of additional FEM models are not easily compatible with the duration of the industrial study.

6.2.2 Participants' approaches

The interfaces have been represented using either kinematics relationships or sticking the surfaces concerned. One way used by the participants to represent the screw link between statoric parts (pump casing, flanges, bearing casing and support) and the 6 screw link between the pump and the frame is to relate the screw head to the thread by a rigid element, as illustrated in Figure 9.

The non-rotating shaft, supported by two ball bearings and two hydrostatic bearings, has been considered either ideally fixed or with stiffnesses whose values have been previously determined using analytical computations (radial and axial stiffnesses for ball bearings) or expert analysis (hydrodynamical bearings). The stuffing box radial and vertical stiffnesses have been also considered by a participant.

The boundary condition relative to the frame completely fixed in concrete has been taken into account by clamping all or parts of the nodes that belong to the frame.

6.3 Parametrical analysis on the clamped boundary condition

A parametrical analysis has been performed by one participant. Two configurations have been taken in account: (1) the frame is completely fixed in concrete (all the nodes from the ground to the top of the frame are constrained) and (2) the frame is partially fixed in concrete (not on the whole height, the ultimate line of finite elements is not constrained). It can be seen in Figure 10 that the modes 3, 4, 9 and 10, which all are overall modes, are strongly influenced by the boundary condition representation, with frequency values which can be 20% to 30% lower in the partially fixed configuration.

7 Example of blind modal results on the pump assembly

From partial results available, attempt is made to correlate the first overall numerical eigenmodes of the pump assembly, with respect to the experimentally identified eigenmodes. Figure 11 illustrates the correlation that can be performed using one participant's results: the flexion of the pump along the longitudinal axis, the flexion along the radial axis and a torsion mode can be so highlighted, though the corresponding frequency values present a great discrepancy. A numerical mode has been simulated, which has no correspondent among experimental modes identified.

The Figure 12 illustrates an example of corresponding numerical-experimental MAC matrix: we observe that

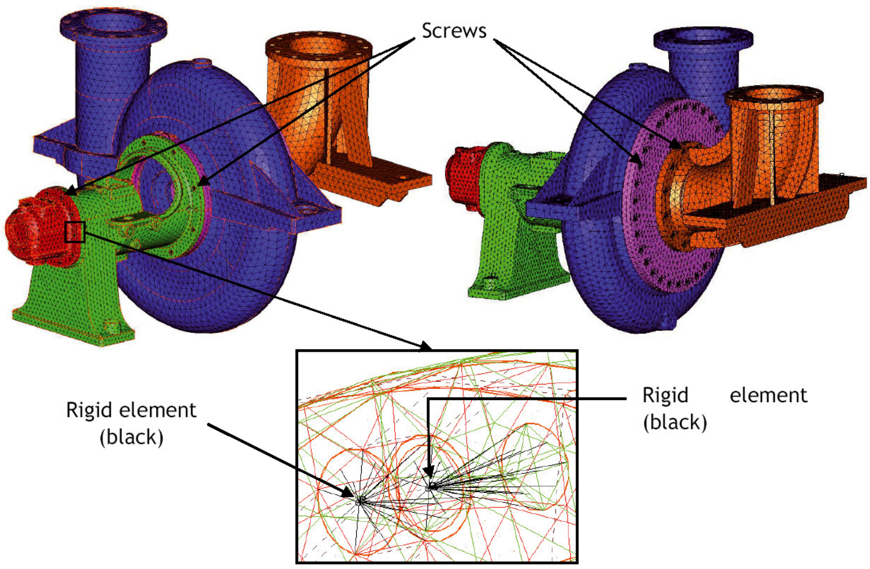

Fig. 9. Connections between pump components.

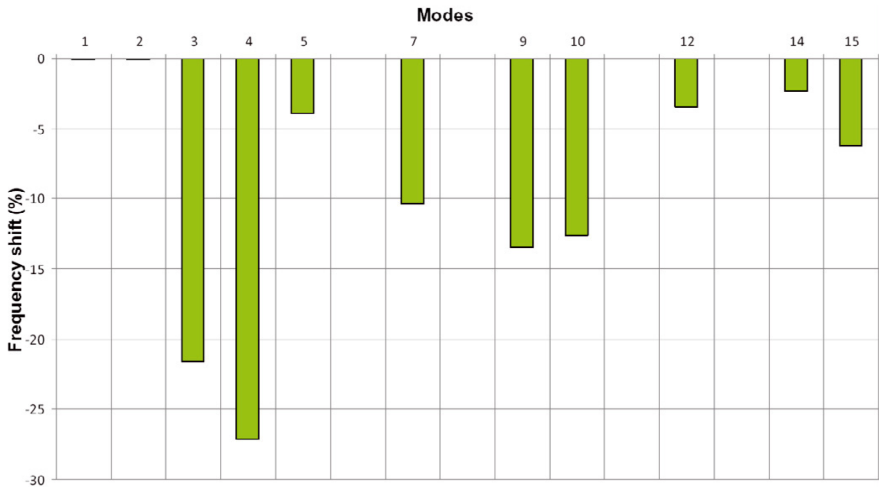

Fig. 10. Frequency shift with respect to the boundary condition configuration (2) compared with configuration (1).

the diagonal line is shifted due to the first numerical mode, which cannot be correlated with an experimental one. Nevertheless the correlation is higher than 0.9 for two modes among the three first ones, which surprisingly shows a better agreement than for the separate pump components.

Updating the interface and the boundary conditions modelling is a work in progress, in view of improving the agreement between numerical and experimental modal results.

8 Discussion

8.1 From modal to frequency response characterisation

If the modal characterisation of a dynamical system modelled linearly in the adequate frequency range can theoretically be sufficient to ensure a satisfactory prediction of its response to a known excitation, we are aware that numerical-experimental confrontation is necessary to confidently use the numerical model within that objective, especially when damping is not well known. It is the reason why modal characterisation is a step in the benchmark process and will be followed by a response analysis.

8.2 On the reference data

Reference experimental modal data used in the first part of this benchmark have been considered without the associated uncertainties. Therefore, for a given structural system, at a given time, experimental eigenmodes present uncertainties both with respect to eigen frequencies and modeshapes. Modeshape errors may first arise due to imperfections in modal parameter extraction sensor calibration errors, measurement noise, accelerometer cross-axis

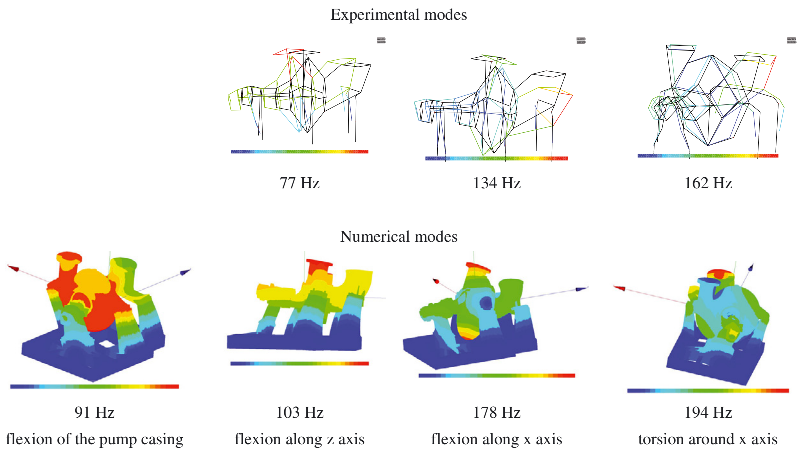

Fig. 11. Compared pump assembly eigenmodes.

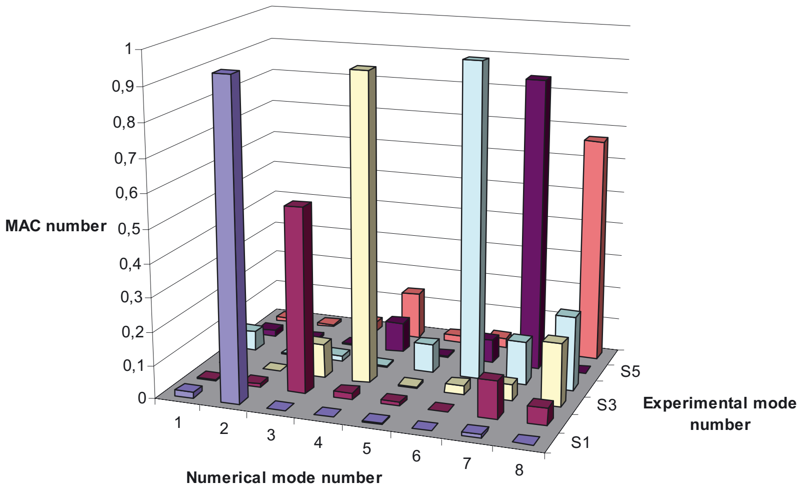

Fig. 12. Example of numerical-experimental MAC matrix relative to the pump assembly.

sensitivity [21]... In addition, variability can be related to time variable interface and boundary conditions: the clamped condition of the frame in concrete, for instance, can become less and less ideal when concrete becomes old. Furthermore, variability can be observed in a series of identical dynamical systems. If attempts are made to rigorously characterize the final measurement error using probabilistic investigation [21], we think that additional measurements within the present benchmark are necessary to obtain an order of magnitude of global experimental data uncertainties.

8.3 Parametrical analyses and sub-system updating

Considering the quantitative result of the parametrical study on the boundary condition, it is obvious that tuning material property values in view of increasing the numerical-experimental agreement on the pump components is not essential to improve the accuracy of the model assembly. Similar parametrical studies on an injection pump already showed the strong influence of interface and boundary conditions on eigenfrequency values [11]. The influence of the interface representation has to be quantitatively examined too on the booster pump to complete the analysis.

8.4 On the predictive capability of assembled structure models

The first rough results presented relative to the pump assembly, in comparison with those relative to separate components, clearly show that the quality of theoretical built-up structure models lags far behind the quality achieved in conventional structure components, due to the lack of appropriate models of the interfaces and joints, easily usable in case of real industrial degree of complexity. On the booster pump example, the fully constrained representation of screws can be justified in case of a great number of them (flange interfaces), but is certainly not adapted in case of only a few screws (ex: 2-screw interface between the frame and the bearing support). Improving the modelling of such interfaces would require either the elaboration of dedicated F.E. models of the connected components, as largely proposed in the literature [18, 19] or the use of measurement data in order to tune equivalent stiffness and damping properties. The first solution is time-consuming but necessary in a purely design purpose; the second solution, combined use of experimental and theoretical techniques, is commonly applied and recommended [17]. Within the present benchmark, additional measurement data will be obtained on chosen pairs of pump components in view of deducing stiffness and damping representative terms for the connected elements. The question of the damping, not taken into account in the participants' simulations, is effectively a crucial complex problem, which must be nevertheless considered to obtain confident numerical models.

9 Concluding remarks

A main objective of the benchmark is to quantify the confidence in numerical models used either in design purpose or in expertise purpose and finally to ensure robust predictions, that is with a difference between prediction and actual response which is within an acceptable range including all uncertainties [21]. In order to measure the effective variability on structural dynamics computations we can observe among different operators and in comparison with reference experimentally measured data, an international benchmark is organised, based on an industrial demonstrator well-representative of modelling complexity and with not well-known parameters. That built-up dynamical system is a pump actually in service in power plants, considered in its work environment. A step-by-step hierarchical process is followed, from modal characterisation of the free-free separate pump component to the response determination of the pump assembly in industrial environment. First results presented concern the modal characterisation of the main free-free pump sub-structures and the pump assembly fixed in concrete. The blind modal characterisation of the separate pump components shows a 5%–12% variability on eigenfrequency values and a less than 15% frequency error in comparison with experimental values; the numerical-experimental MAC numbers reach 0.7 at the maximum, even after updating. After possible simplifying process in order to reduce the model final size, assembling is performed by participants at a macro level, applying kinematics relationships or considering fully restrained connections. Some parametrical studies show significant 20%–30% influence of the boundary condition representation on the eigenfrequency values. An example of modal blind results on the pump assembly fixed in concrete presents a larger numerical-experimental discrepancy than for the separate parts, essentially due to the modelling of the interfaces and boundary condition. Though a significant frequency error, the first overall modes are correctly identified; the corresponding MAC matrix shows agreement higher than 0.9 for two modes. Further work within the SICODYN project will interest in the improvement of the model accuracy using experimental test data on main sub-structures, pairs of components and pump assembly with different boundary conditions.

If this tendency can be confirmed from all the participants' results, the lesson to draw is that the quality of theoretical built-up structure models lags far behind the quality achieved in conventional structure components, due to the lack of appropriate models of the interfaces and joints, easily usable in case of real industrial degree of complexity. Among the sources of model errors (mathematical representation, elasticity constant uncertainties, geometry simplifications, discretisation error, ignoring non-linear interactions and flexibilities at joints, inaccurate modelling of the boundary conditions...) which are particularly highlighted in built-up complex systems, some can only be corrected through testing and model calibration [22]; so the confidence in complex models within a purely predictive (blind) purpose seems difficult to be reached. Furthermore, the variability concept must be enlarged in view of the use of numerical models for a whole family of nominally-identical structures and considering both numerical and experimental uncertainties. The stochastic or interval framework proposed in literature, possibly coupled with component mode synthesis method, can be an interesting way to properly quantify variability and uncertainty relative to numerical models [23–29].

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Appendix 1: participating groups

The 17 partners currently involved are:

– Institut FEMTO-ST UMR CNRS 6174: E. Foltête, M. Ouisse
– LAMCOS UMR CNRS 5259 INSA Lyon: S. Baguet, R. Dufour, C. Duran
– VIBRATEC: C. Clerc, A. Coulon, C. Devemy
– CETIM: Y. Goth, J. Peigney
– PHIMECA Engineering: E. Noret, T. Yalamas
– SAMTECH: P. Pasquet
– (SULZER Pompes France)[1]: R. Petit, P. Courcot
– EDF R&D: C. Bodel, P. Cadou, A. Mikchevitch, I. Lakhssassi
– ILM Technology
– Delft University of Technology (TU Delft, The Netherlands): D. Rixen, E. Hooijkamp
– Bristol University (England): D. Ewins
– Politecnico di Milano (Italy): N. Bachschmid
– Ecole Polytechnique de Lausanne : M. Calmon
– Gologanu (Roumania): M. Gologanu

[1] SULZER Pompes France is not involved in performing simulations, but in facilitating any work in relation with the structure itself (pump characterisation, experimental aspects, informations needed to simulate its dynamical behaviour).

– MSO Industrial (Colombia): N. Rueda
– PIKITAN (Spain): C. Colino
– CAEnable (USA): K. Sivagnanam.

Only part of these official participants' simulations were available for the partial synthesis of results presented within this article.

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