Introduction to Design and Analysis of High Speed Pumps

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1.0 GENERAL INTRODUCTION

1.1 General Remarks on Main Flow Characteristics in Pumps

The physical mechanisms which govern the flow characteristics in a turbo-machine and specifically a pump are complex, numerous and partially explained. Flow conditions are generally three dimensional, viscous and non stationary and for the case of cavitating flow, one have to take into account two or multiphase conditions.

The non stationary character of the flow is obvious if one consider rotor-stator or rotor-volute interactions. In the case of non uniform inlet conditions or intrinsic heterogeneity created by off design conditions and/or two phase flow, the phenomena is more complicated to understand. Transient regimes also lead to non stationary phenomena.

The correct understanding and evaluation of these physical mechanisms become more and more important because of the high level of concurrence between pump manufacturers. They have to deal with high efficiencies, extended stabilised operating zones, more compact and reliable machines with severe geometrical constrains.

For example, the search for more compact and lighter pumps has led to an increased rotational speed. The consequence of this trend has inevitably increased the potential for fluid- structure interaction problems and the severity of those problems on the pump itself and its environment. Even in the absence of cavitation and its complications, these fluid structure interaction phenomena can lead to increased wear and, under the worst conditions, to structural failure. In addition, and very often simultaneously, cavitation also becomes the main issue.

Cavitation first consequence in a pump concerns steady state hydraulic performance damages. Secondly, cavitation is typically a non stationary phenomena and, because of its very complicated nature, may lead to a quite important variety of other problems like global flow oscillations, local flow oscillations and fluid induced radial and rotordynamic forces.

An attempt to clarify and classify these phenomena has been already done by Brennen [1].

Perform a good identification and a good analysis of all the mechanisms that may occur in the different pump elements is still a challenge. Investigations and researches are still both needed in theoretical and experimental fields in order to get a better improvement of the pump performances in the maximum

Bois, G. (2006) Introduction to Design and Analysis of High Speed Pumps. In *Design and Analysis of High Speed Pumps* (pp. 1-1 – 1-20). Educational Notes RTO-EN-AVT-143, Paper 1. Neuilly-sur-Seine, France: RTO. Available from: http://www.rto.nato.int/abstracts.asp.

operating range. A lot of work has been performed on both subjects in the last three decades because of the use of more efficient CFD methods and sophisticated non intrusive measurement techniques allowing to take into account the flow characteristics with less simplified hypothesis.

1.2 General Remarks on Pump Business Evolutions

(Comments appearing in this section are mainly extract from Cooper [3], 1996)

One has to keep in mind that the pump business context has completely changed during the last decades. As shown in figure (1), the required pump power and stage pressure increase quite a lot since 1950.

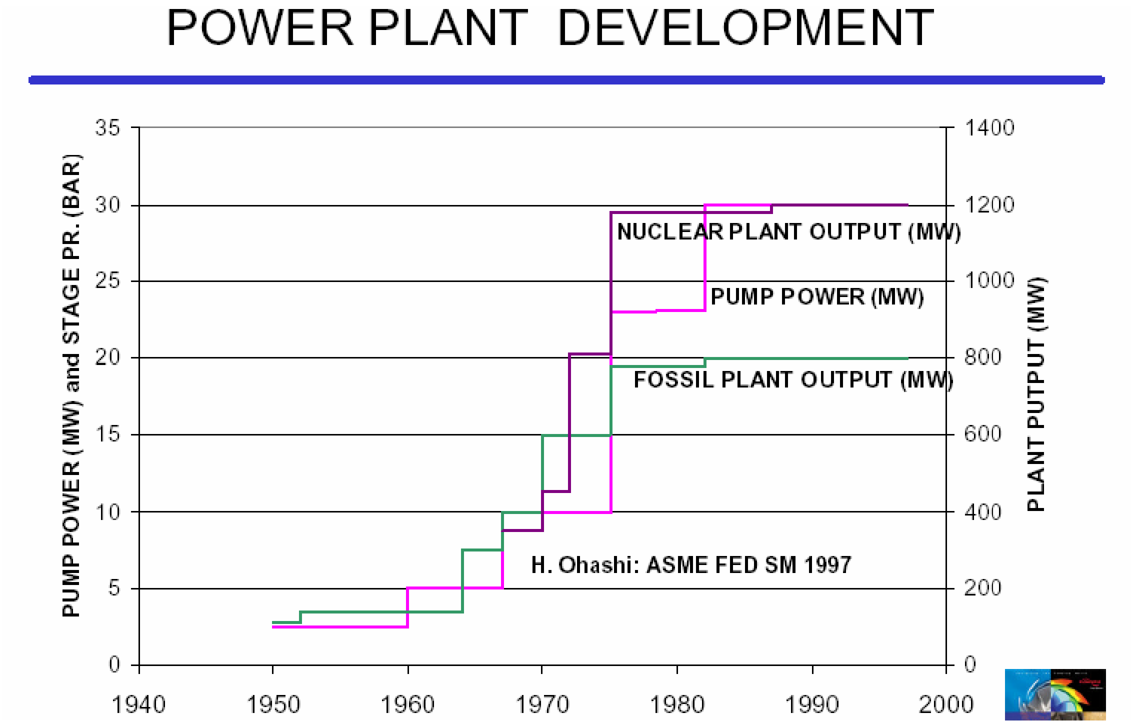


Figure 1: Power Plant Development from 1950 to 2000.

This has seriously changed the research approaches and targets. As observed by Ohashi and Tsujimoto [2], and reported by Cooper [3], one can determine three phases in the industry, known as the Expansion Phase (Past), the Environment Phase (Present) and the Globalization Phase (Future). These phases are summarised in the paper of T. Dahl [4]) as following:

- The Expansion Phase occurred during the post WW II re-building of major economic centres. During this period, the majority of new pump families were conceptualised and developed. The underlying technologies that led to the fundamental physical understanding of the behaviour of pumps also evolved during this period. This included research on rotor-dynamics, hydraulic instabilities, cavitation and engineering materials. This phase was subsidised by the economic boom associated with the post WW II expansion, dedicating substantial flow of funds and resources to new pump technology research and development.
- The Environmental Phase, which began in the 1980's, resulted in a convergence of the performance-oriented design and product design technologies of the earliest phase with the environmental and emission control issues of the period. Sophisticated hydraulic analysis and design software were developed to optimise the basic design embodiments developed earlier.

Higher efficiency machines and sealing technology with lower emissions were profiled during this period. In addition, pump installations were evaluated for their total energy costs and retrofitted with equipment that was better suited for the current or changing conditions of services. The main customer requirements are the shown in figure (2). One can observed that reliability and efficiency remain the most important features. Control range and suction ability are also pointed out. All these items are mainly dealing with fluid mechanism problems in turbo-machines.

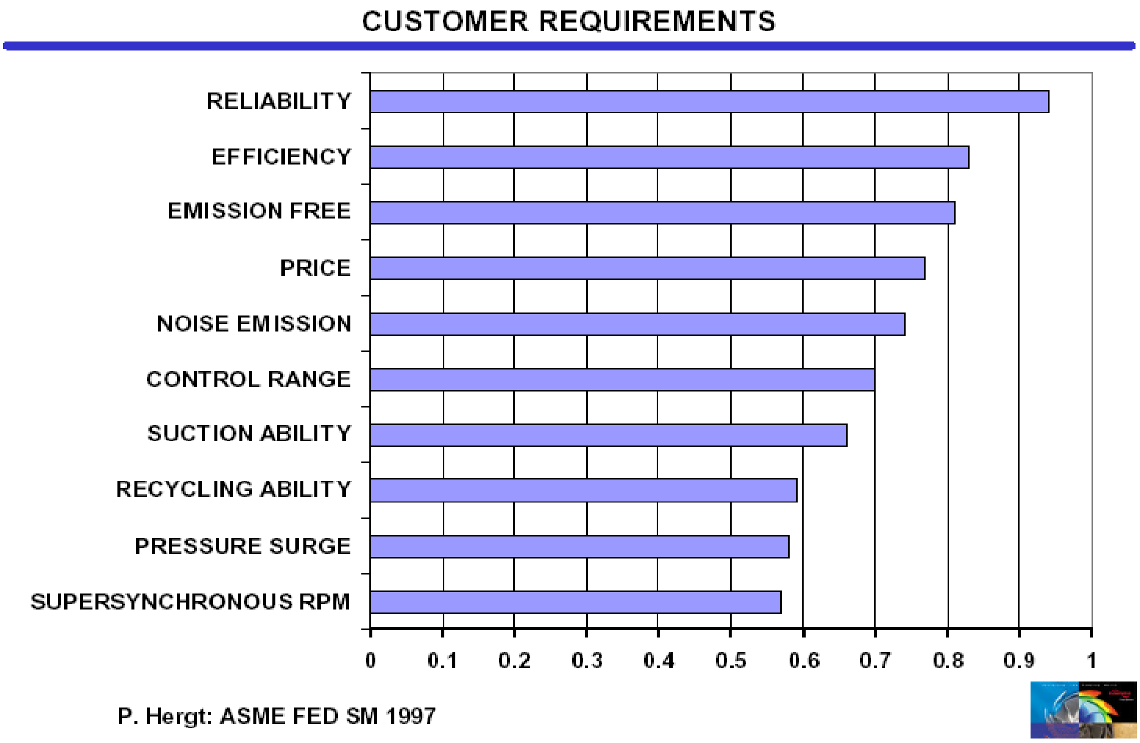


Figure 2: List of Customer Requirements (from P. Hergt).

- Now, the pumping industry is evolving into a Globalisation Phase. This phase is driven by the substantial influence of information technology on the economics of the traditional manufacturing firm and its role in the pumping industry's value-chain. In this latest situation, the fluids engineering and other technical challenges are assumed to be solvable by the multinational companies who know emphasise quality, productivity and faster delivery as they battle for global market share.

2.0 SOME TECHNICAL ASPECTS CONCERNING RELIABILITY AND EFFICIENCY

2.1 Reliability

Concerning the economics of reliability, one important fact is that the cost of maintaining a pump significantly exceed the first cost. The chart presented in figure (3) show that the repair cost may rise to 27 % of the total cost of an operative pump.

COST OF RELIABILITY

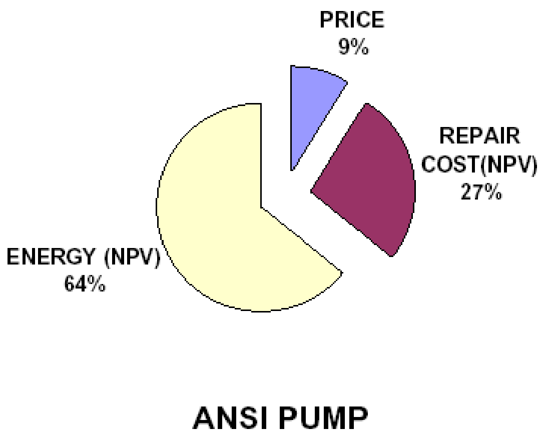
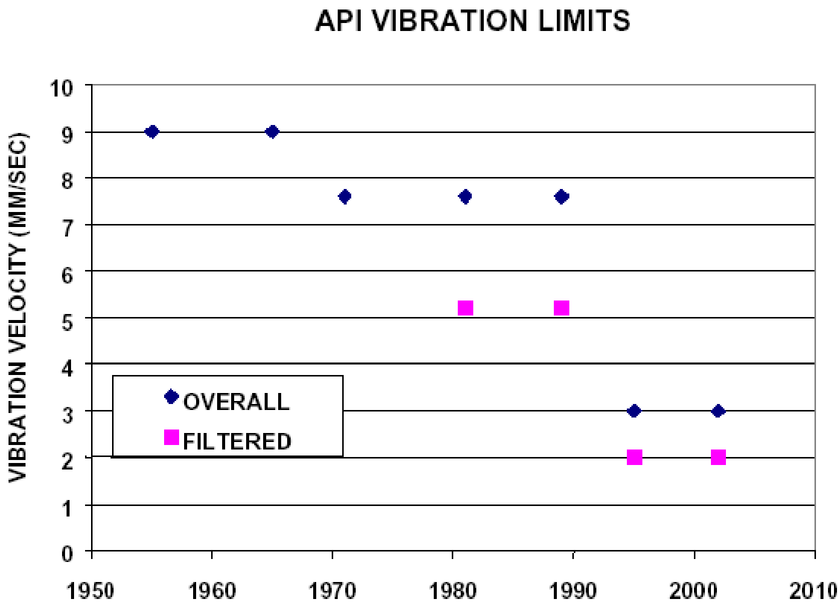


Figure 3: Repartition of Cost.

Manufacturers has to increase the MTBR (Mean Time Before Repair) due to failure on several parts of a pump.

In order to improve the reliability, more research has to be made to minimise vibrations; this means mainly to try to reduce the forces and eliminate the sources of resonances. Improvement on cavitation resistance is also an important feature. Figure (4) show how vibration limits decrease since 1950 up to now. It has been possible by some technological achievements by increasing foot thickness, decreasing bearings span and by mounting robust bearing adaptors. An example is given, figure (5), from the fluid engineering point of view, to reduce vibration by an increase of the gap between impeller output and the volute . The consequence of such modifications reduce the maximum bearing vibrations with flow rate as shown in figure (6).

RELIABILITY IMPROVEMENT



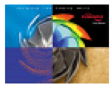


Figure 4: Reduction of the Maximum Bearing Vibrations with Flow Rate.

INCREASE OF GAP “B”

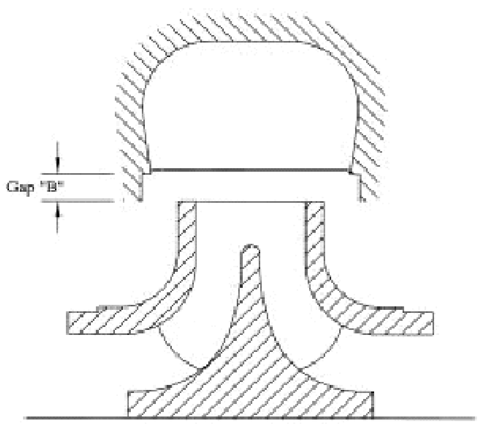


Figure 5: A Way to Reduce Vibration by an Increase of the Gap “B” between Impeller Output and Volute.

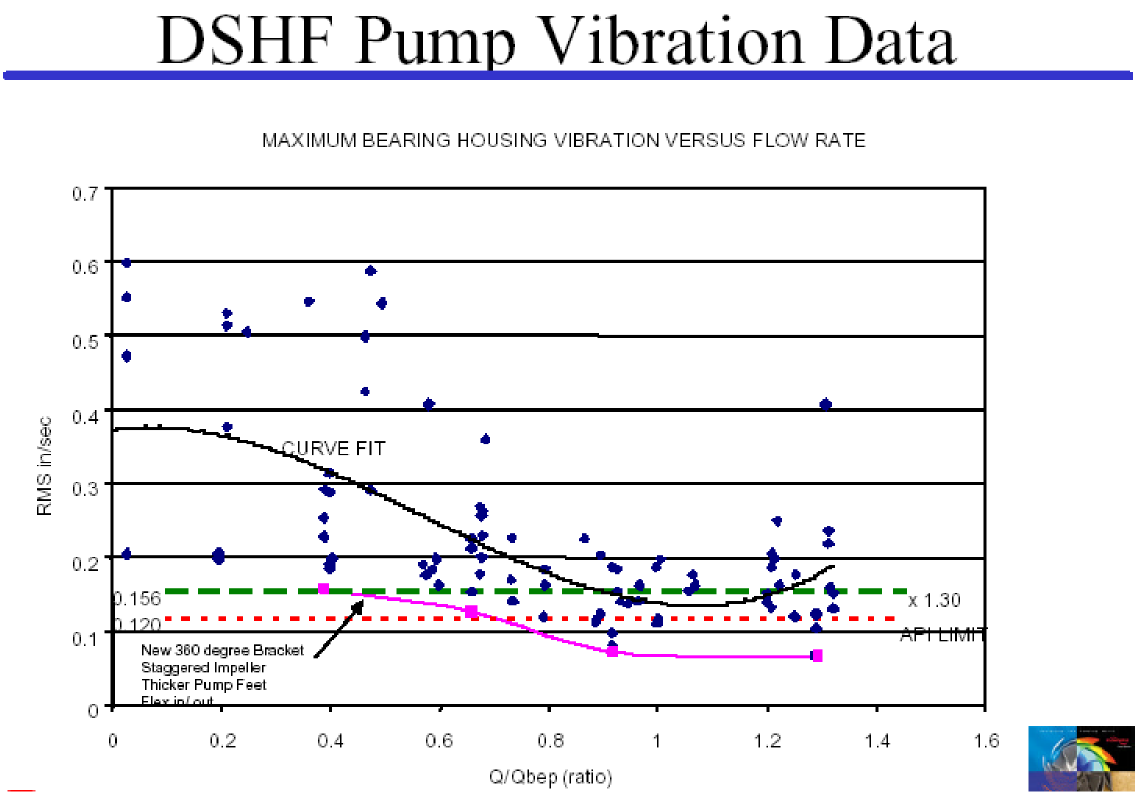


Figure 6: Evolution of Maximum Housing Vibration versus Flow Rate.

2.2 Efficiency

A project funded by the European Commission (SAVE) has concluded:

- Pump efficiencies can be improved with present technology by 3 points.
- If all EU pump are upgraded, a total of 1.1 TWhr of energy can be saved. At 5c/kWhr, this amounts to about more than 40 million Euros saving per year.
- Basic infrastructure issues are the impediment to this upgrade.

Figure (7) presents the theoretical attainable maximum efficiency for a volute casing single stage pumps. For more details and requested conditions to apply, see references [5], [6], and [7].

Even if this conclusion is applicable for specific centrifugal pumps with rather low specific speed, one may supposed that a better understanding of flow structures inside pumps can provide better predicting models in order to gain in efficiency not only at best design point but also all over the operating range.

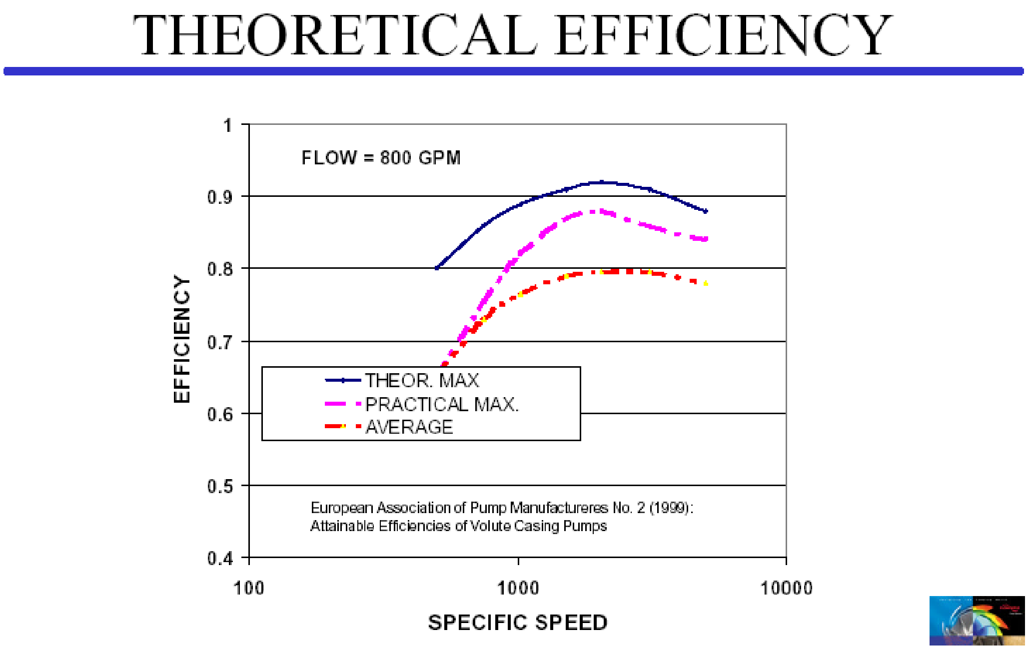


Figure 7: Theoretical Attainable Maximum Efficiency for a Volute Casing Single Stage Pumps.

3.0 MAIN FLOW STRUCTURES IN A PUMP

3.1 Flow Structures without Cavitation

All rotodynamic pumps (axial, mixed and radial flow pumps), of all sizes and power, for any kind of application will operate at some time away from their design point in spite of the more or less large energy dissipations that are encountered in these off-design conditions. However, the starting point of the design or the study of a specific pump starts from design configurations depending on the conditions the pump is going to be used. In order to reach good efficiency, it is necessary to predict correctly the flow, not only by one dimensional considerations and approaches but with reliable 3D flow calculations capable to reproduce flow structures in all the different parts of the pump and finally fit the overall pump performances in the maximum range of operation.

When specifications are given, a preliminary design is performed using different tools going to the so called design process. Then, component designs are performed, taking into account performance, weight, cost, life, reliability, structural strength, maintainability, envelope, etc.

Depending on the level of in house design data base and/or CFD reliability and confidence, component design, test and analysis are performed before final design and acceptance tests.

For some particular aspects, 1D approaches can give rather good global evaluations on pump performances but they are often used for preliminary approaches in rather classical cases.

Figure (8) give a comparison between predicted (1D calibrated approach) and experimental overall performances of the whole stage of a mixed flow pump, from reference [8].

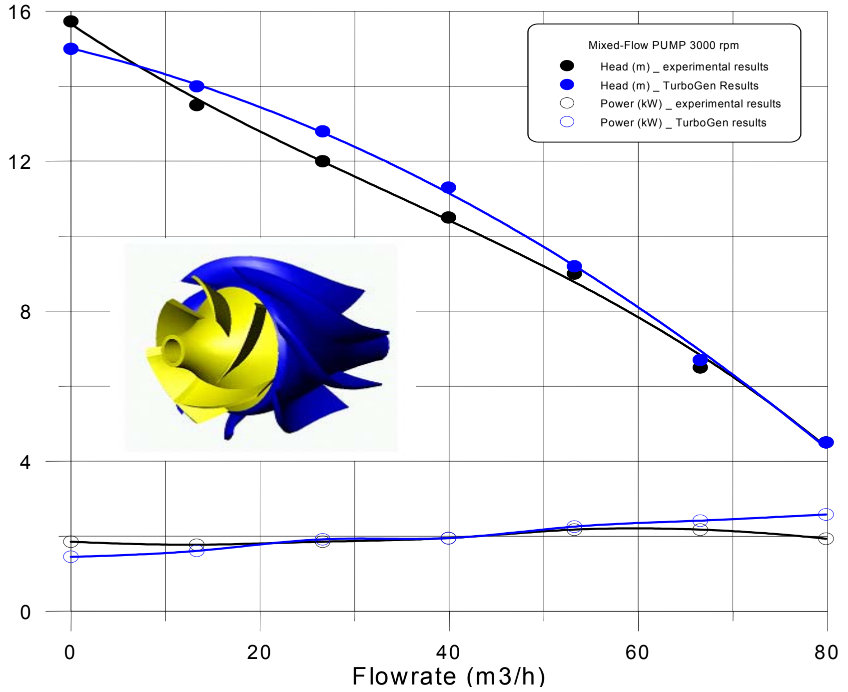


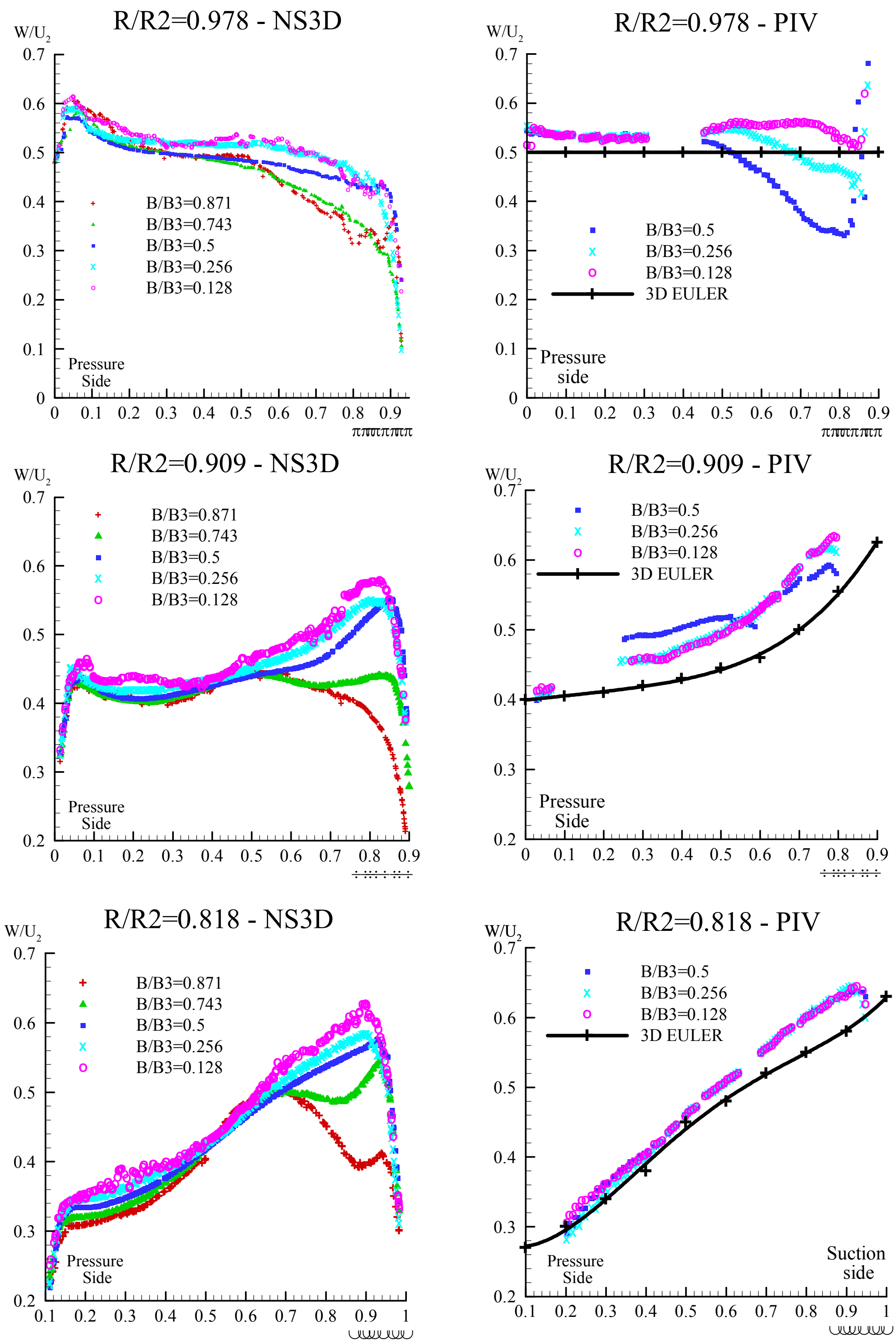
Figure 8: Mixed Flow Pump.

For main cases, different approaches with different level of approximations are generally needed in order to obtain better results taking into account maximum of the required constrains. Among these method, inverse and optimisation techniques give quite good results. This aspect will be treated by Prof. R. Van den Braembussche.

3.1.1 Design Analysis

3D Euler and RANS solvers are now available and are able to give local details on fluid structure for a better design of different parts of a pump. One has to keep in mind that interpretation of CFD results is one of the most important part of the design procedure (see section of the present document) and that it is necessary to compare experimental results with calculations in order to perform good design process.

An example of calculated and experimental results on blade to blade relative velocity distribution inside a shrouded radial impeller are given in the following figures (9)a. to (9) f. for different radii near the trailing edge region (reference [9]).



Figures 9.a. to 9.f: Calculated and Experimental Results on Blade to Blade Relative Velocity Distribution inside a Shrouded Radial Impeller for Different Relative Radial Positions and for Different Hub to Shroud Sections (low values of B corresponds to the hub region).

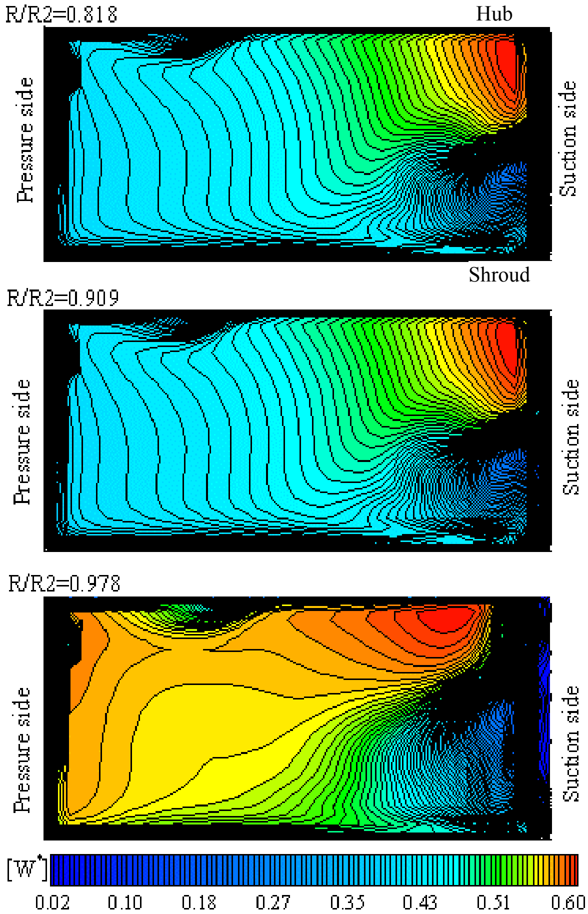


Figure 10: RANS Calculated Relative Non Dimensional Velocity in the Same Plane given in Figure 9.

3.1.2 Off-Design Analysis and Problems

Off design conditions strongly modify internal flows and affect pump performances. They are characterised first by the mismatch between inlet blade angles and local fluid angles. These mismatches induce additional losses that are referred to incidence losses which affect the hydraulic performances, pump efficiency, total head, power input and required net positive suction head (NPSHR) if cavitation is take into account. These phenomena also generate stronger unsteady flow conditions, such as stall, wakes, turbulence and pressure fluctuations, which affect the overall mechanical behaviour of the pump with vibration, noise and radial and axial forces on the rotor.

Some of these aspects will be treated by Prof. G. Pavesi in particular for impeller volute and diffuser interactions.

Typical hydraulic characteristics of a pump for a given rotation speed, N, are represented by the relationship of the pump total head, the pump power input and the pump efficiency, with the pump flow rate. This is shown schematically in figure (11) extracted from reference [10].

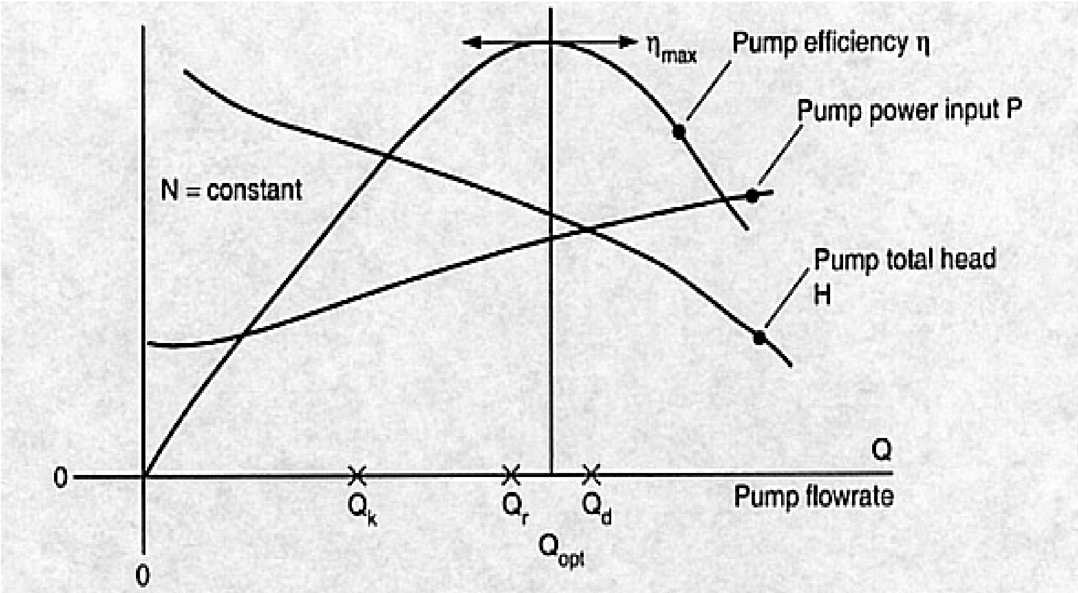


Figure 11: An Example of the Hydraulic Characteristics of a Radial Flow Pump at a Given Speed of Rotation.

The shape of the different characteristic curves depend mainly on the impeller and the diffuser geometrical parameters, on the specific speed Ns, or type number K, as shown in figure (12) from reference [10].

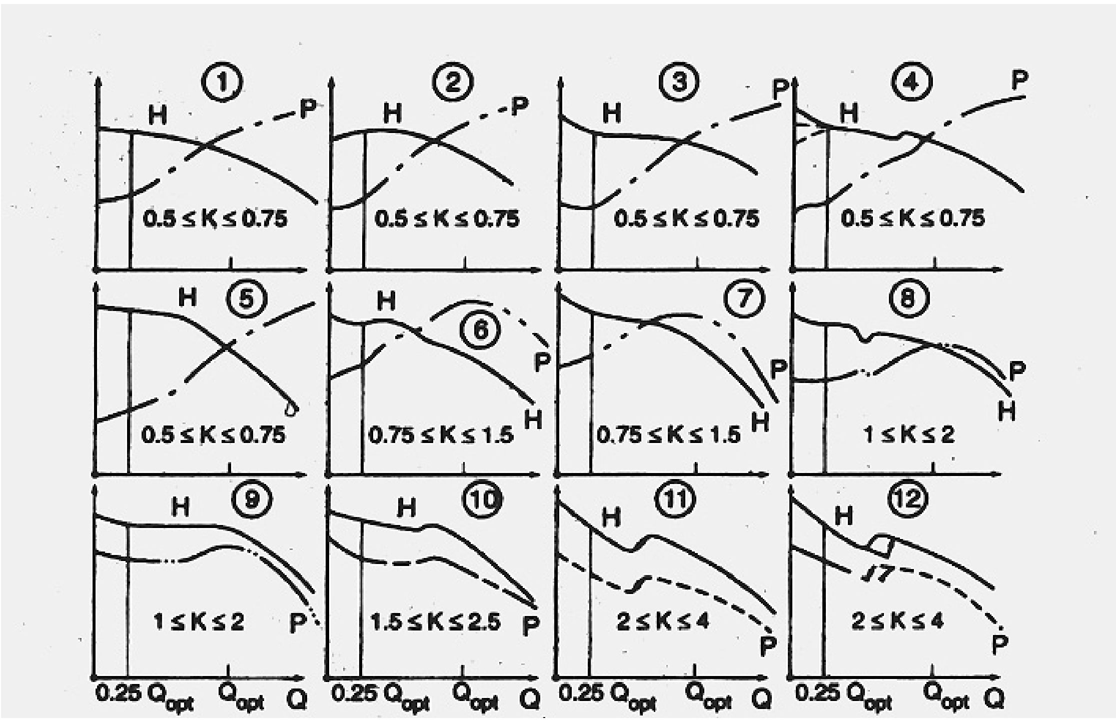


Figure 12: Various Shapes of Head-Flow and Pump Power Input Flow Characteristics at a Constant Speed of Rotation according to the Type Number K of the Pump.

Off design conditions are often related to recirculation problems at partial flow rates.

Recirculation or reverse flow can be defined by the fact that the fluid goes through a flow cross section with velocities havins opposite arithmetic signs. This can, for example, be observed in various part of the machine, but the most important effects of recirculation are encountered when such a recirculation zone appears across the revolving surfaces containing the leading or trailing edges of the impeller, or the diffuser blades. In a properly designed pump, these recirculations occur at partial flow rate. Those associated with the rotating blade passages have a particular strong influence on the overall pump behaviour (hydraulic characteristics, pressure fluctuations in the pump and the installation, vibration and noise).

Figure (13) presents schematically, for a meridional section in a radial flow pump, the difference between a well- developed flow near design conditions and a pertial flow with recirculation in the inlet and the outlet part of the impeller. Figure (14) illustrates what can be observed in different types of pumps when recirculation is present.

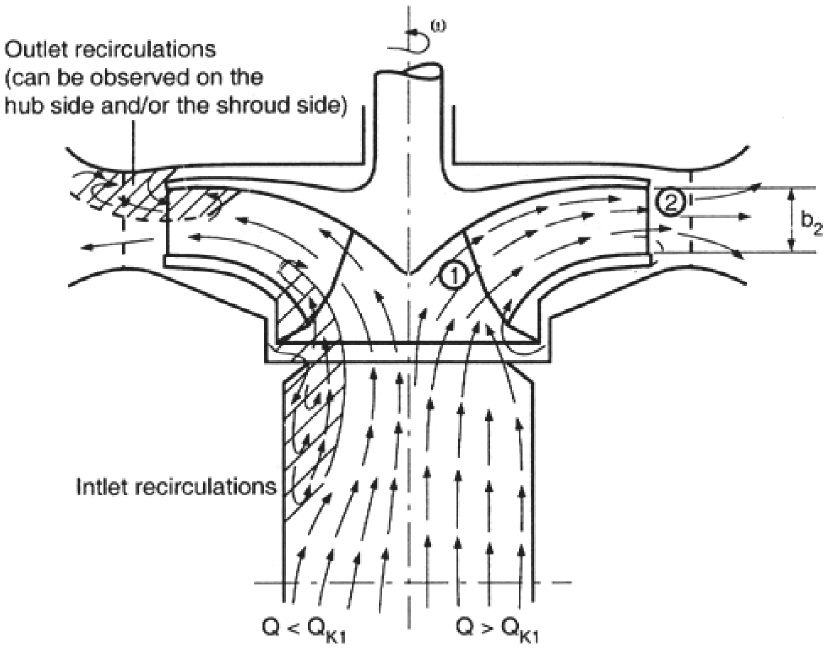


Figure 13: Schematic Illustration of the Flow in a Meridional Cross-Section of a Radial Flow Pump: in Near Design Conditions (right-hand side of the figure); and in Partial Flow Regimes (left-hand side of the figure).

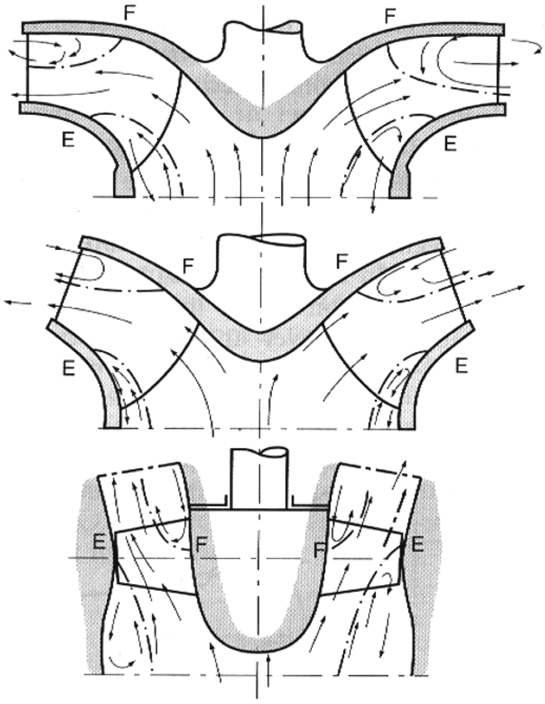


Figure 14: Different Types of Recirculations at Off Design Conditions and for Different Type of Pumps.

At impeller outlet, the small gap between the impeller and the stationary parts induces stronger interactions, and therefore the phenomena become more complex and unsteady. The flow structures that are shown in the previous figures indicate an outlet recirculation zone near the hub. In fact, the flow structure near the outlet part of an impeller is dependant upon the impeller design. In partial flow ranges, there is a strong influence from the stationary parts of the pump close to the outlet of the impeller (for example radial gaps) and reverse flow appear to be a result of the inadequacy of the matching between the outlet of the impeller and its environment. That means that reverse flow can occur either on the hub side or the shroud one, or both sides simultaneously. The development of RANS computer codes for pumps will probably lead in the near future to a better accuracy of these critical flow rate predictions and consequences. Statistical studies clearly show that the ratio of impeller recirculated mass flow over optimal mass flow increases with specific speed (reference [12]).

Off design conditions can also being associated with operational instabilities depending upon the shapes of the characteristic curves for the case of a single pump operation as shown in figure (15).

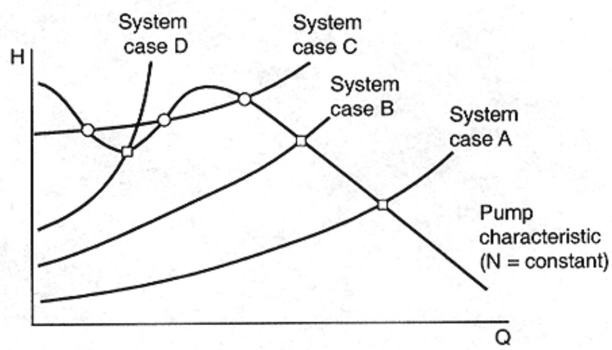


Figure 15: Static Stability Analysis Depending on the Circuit System.

3.2 Flow Structure and Cavitation

A detailed review of cavitation problems within rotodynamic pumps can be found in reference [11].

Cavitation occurs in any part of a machine or a circuit where low local pressure appear and/or with high speed. The type of cavitation depends on its localisation in the machine and upon the physical aspects that have created local low pressure. So, cavitation structures and two-phase mixture may be different whether it appears in pipes, on blade leading edges generally at tip sections for design mass flow, on suction side for high incidence, on pressure side for low incidence, inside tip clearance of non shrouded blades, tip section of propeller and also, at trailing edge and outlet parts of hydraulic turbines etc.

In order to qualify cavitation, one generally use the THOMA's number or cavitation number σ.

Next presentation, which will be given by Prof. J.P. Franc. He will spoke upon basic aspects, physics and control of cavitation including some specific thermodynamic aspects.

As already said, cavitation cause strong instabilities and pump damage. Cavitation instability characteristics strongly depend on the operating conditions of the pump and these is going to be shown in other chapters of this lecture series by Dr. D. Wulff.

We only would like to describe here the general overall consequences of cavitation in a pump.

3.2.1 Radial Pump Configuration

When cavitation is present, one has to take into account the fact that overall performance may be destroyed for certain level of THOMA's number. As a consequence, a critical value of Total Head is generally define as the NPSH (Net Positive Suction Head) which correspond to the margin of the absolute value of the total head above the head equivalent to the vapour pressure of the liquid at a particular temperature.

For a given mass flow rate, pump total head suddenly decrease when the initial total pressure.

Decrease. This is shown on figure 16. Depending on the NPSH value, the spread of the cavitation zone is different, as shown in figure 17, that correspond to an attempt of visualisation of such patterns using a CFD code obtained by Coutier (reference [15]).

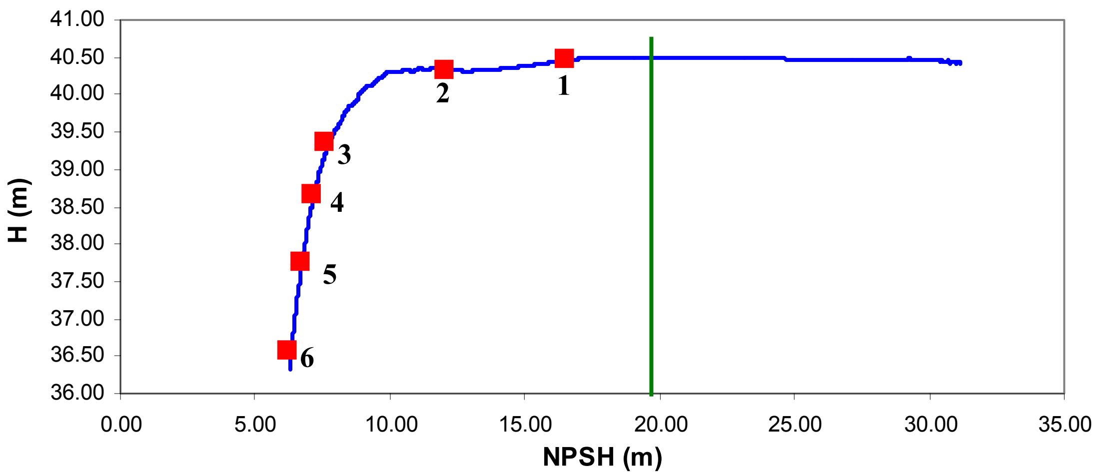


Figure 16: Radial Pump Cavitation Characteristics for Nominal Flow Rate.
The red points correspond to operating points visualised in figure 17.
The green line corresponds to cavitation onset in the pump.

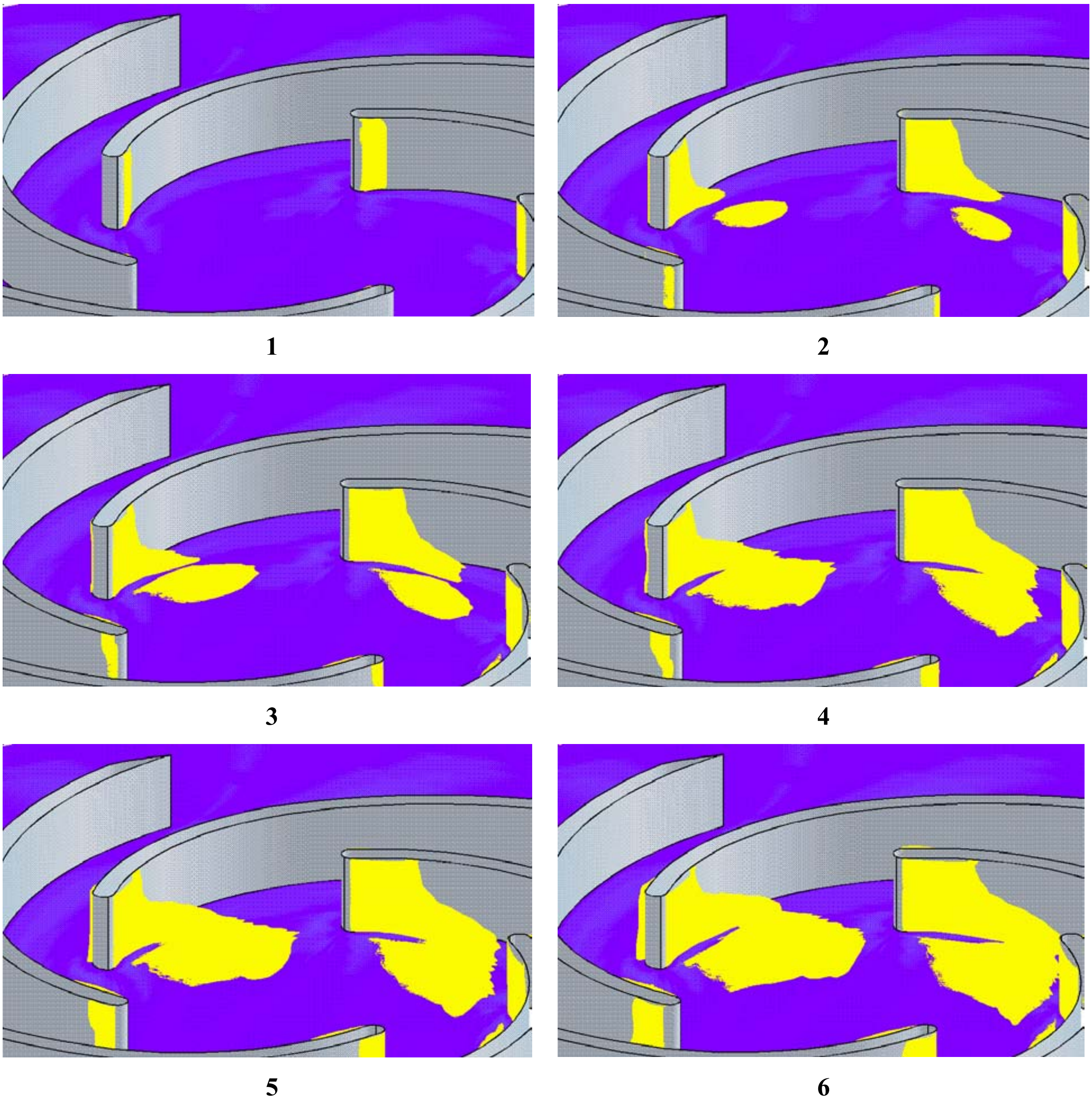


Figure 17: Development of Two Phase Region Obtained with CFD Results for Decreasing Thoma's Values. The numbers refered to positions indicated on figure (16).

Depending of flow rates, the corresponding curves are modified as shown in figure (18).

• From Huzel, D. K. and Huang, D. H.

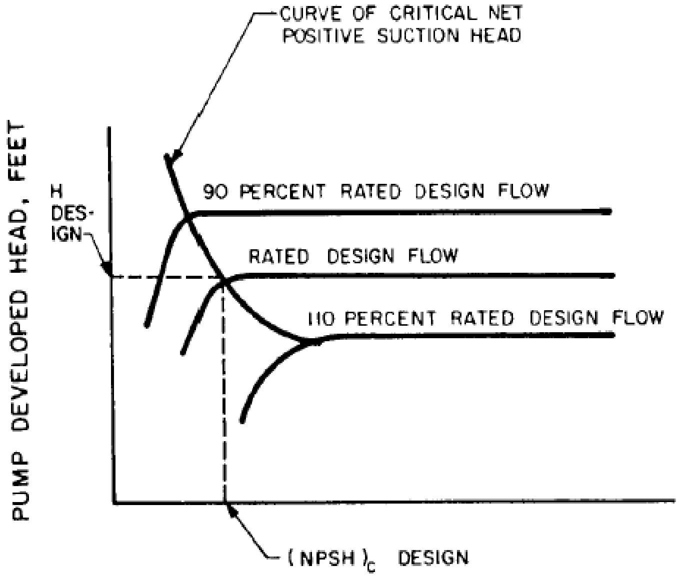


Figure 18: Pump Suction Performances Depending on Mass Flow Rates.

The corresponding curve showing the "curve of critical net positive suction head" presented in the previous figure is shown figures (19) and (20), considering mass flow rates on the x coordinate.

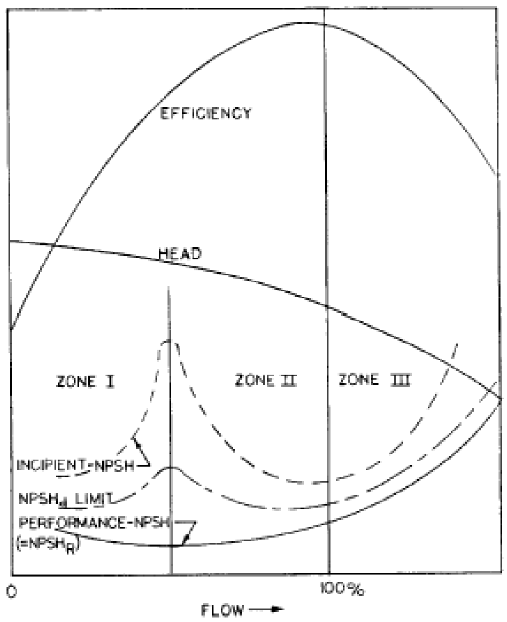


Figure 19: Typical Pump Performance Curve, Showing NPSH Required: a) to Maintain Hydraulic Performance or Pump Head (NPSHR); b) to Limit Cavitation Damage and Therefore Maintain Pump Life (NPSHd); and c) to Prevent Bubble Formation Entirely (NPSHi) from Cooper and Antunes (1983).

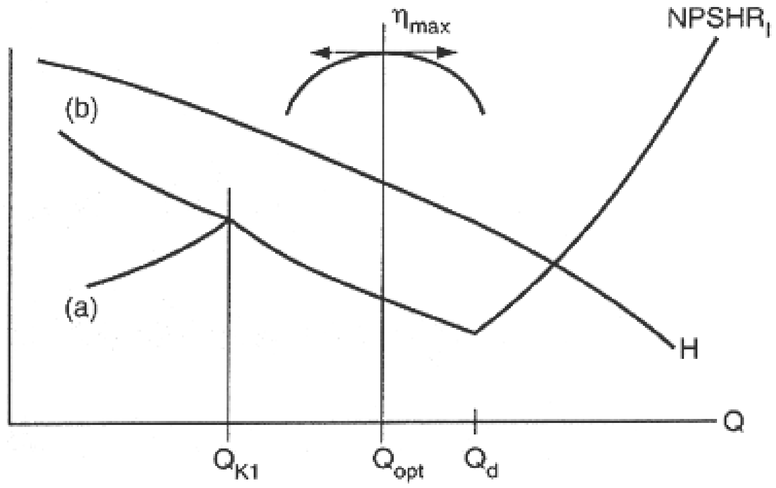


Figure 20: Relationship between the Required NPSH in Incipient Conditions (NPSHR) and the Flow Rate in a Pump at a Constant Speed of Rotation: (a) Straight Pipe in Front of the Pump Impeller Inlet; and (b) Ribs or Elbows in Front of the Pump Impeller Inlet.

3.2.2 Inducers

Inducer is an axial flow type of turbo-machine. A typical geometry is given figure (23). Inducer is generally placed in front of centrifugal pump stages. It is specifically design to be able to suck flow in order to prevent cavitation inside the centrifugal part of the whole machine. As already said, the design criteria for an inducer and specifically for high speed pump do not only depend on mono-phase mean steady flow assumptions, but also on two phase unsteady considerations because of the specific kind of physics that occur in such flows. This cause damages due to instabilities and vibrations that may also influence the entire pump conception criteria especially concerning seals and bearings configurations and designs. This last aspect is going to be developed by Prof. San Andres. On the other hand, cavitation is related with the pump live due to erosion damages. An example of blade shape adaptations is given in figure (24) in order to maintain life pump as large as possible. A lot of work is still done on material either.

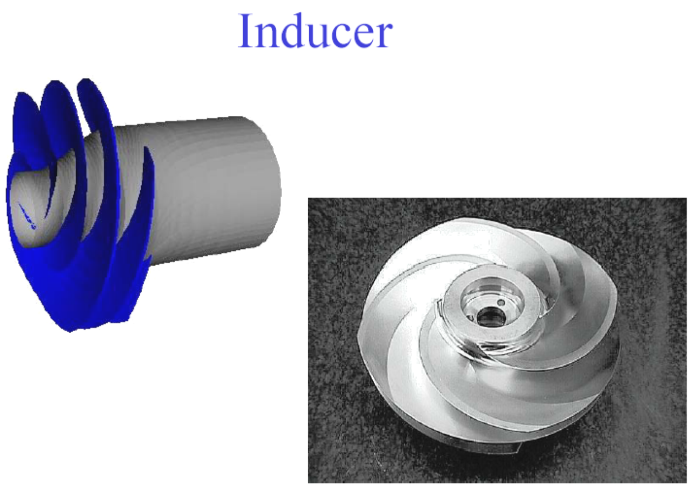


Figure 21: Typical Inducer Geometry.

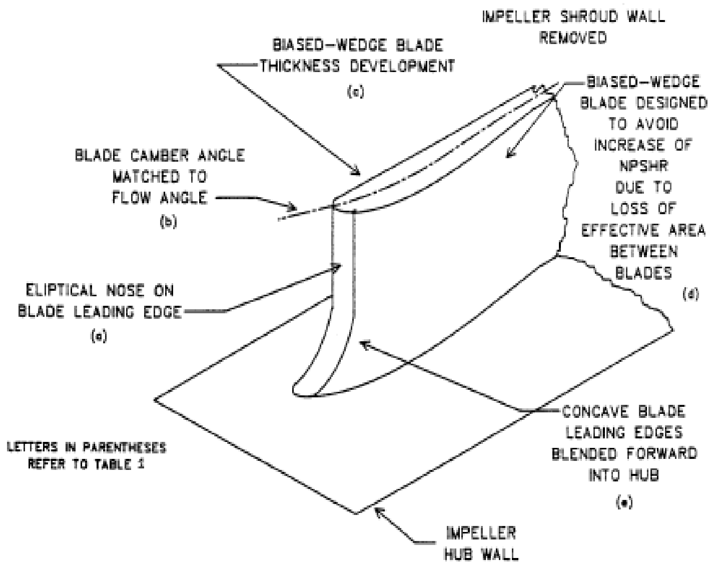


Figure 22: Reduction of Cavitation Damage.

Inducer's head curve modifications are quite identical to the one observed for radial pumps.

However, the design parameters can strongly modify the flow structures depending on number of blades, stagger angles, meridional shapes, bending and /or leading of the blade, tip clearence, radial equilibrium choices, incidence laws and inlet conditions.

Some of the flow structures are showing on figure (23) in a particular inducer's geometry. More other details can be seen for another inducer in figure (24) with a first attempt of CFD results obtained by BAKIR in reference [8].

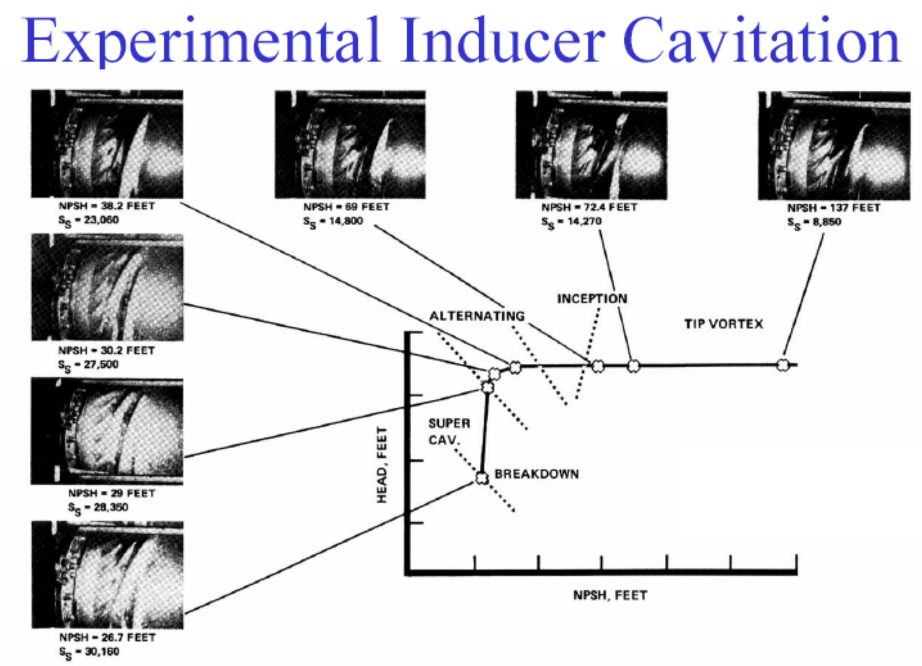


Figure 23: Different Views of Cavitation Flow Patterns.

Two particular sections of this lecture series will be devoted to cavitation. The fist one concerns the numerical modelling of the cavitation and its application to rocket engine turbo-pumps. The presentation will be given by Dr. B. Pouffary. The other one will be given by Prof. Y. Tsujimoto on “Cavitating and non cavitating pump flow”.

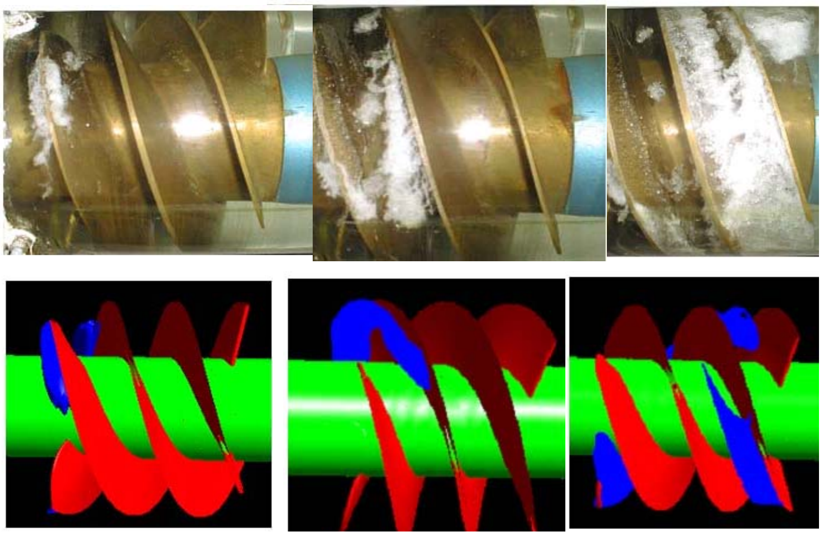


Figure 24: Cavitation Pattern for Relative Mass Flow: qv/qn=1.09 with Variing Thoma’S Number (shown void fraction corresponds to 10% of vapor).

4.0 CONCLUSIONS

While globalisation apparently de-emphasises the technical and fluids engineering aspects that have always characterised the pump industry, technical improvements will continue to be necessary to maintain global competitiveness. Here are some ideas of areas of expertise that must be continuously improved if success is to be achieved in pumps industry:

- Successes can be expected in stabilising the head characteristic of large, high-specific-speed mixed flow pump. CDF has to be a tool to aid in making such improvements.
- CFD also promises to take the guesswork out of predicting the pump performance curves. Missing the performance of a new design has been a costly element of the pump business for a long time.
- Eliminating vibrations and off-design pressure and flow fluctuations is another challenge that still remains, and superior performance in these areas will influence the choice of pumping equipment by customers. In a companion article to that of Ohashi and Tsujimoto (6), S. Kopalakrishnan emphasises the advances that have been made in finding solutions to both rotor-and structure-related vibrations.
- The same author also discusses the fact that structural modal analysis have become both common and essential as an aid to the design and improvement of the pump components and mounting configurations.
- Another important subject is cavitation. Its affects both performance and pump life. In the future, users will routinely impose a life requirement of five or more years against failure of pump parts from cavitation erosion. Cavitation is now studied in research laboratories and this has led to improved blade shapes that have reduced and even eliminates cavitation damage in some important applications. Together with related improvements in materials, this is a major reason for the extended pump life already mentioned especially for high speed machines.

- We have not spoken about another important feature concerning the expected successes in this field. This concerns experimental works that have to be done with sophisticated techniques in order to give more detailed and precise informations.
- This topic will be tackle by Dr. D. Wulff. One have also the possibility to get such informations about new experimental techniques in past or former von Karman lecture series.

5.0 LIST OF REFERENCES

[1] Brennen, C.E., 1994, “Hydrodynamics of pumps”. Concepts NREC ETI and Oxford University Press.

[2] Ohashi, H., Tsujimoto, Y., Hergt, P. and Kopalakrishnan, S., 1997, “Pump Research and Development – Past, Present and Future.” Three papers on the Japanese, European and American Perspectives respectively; 1997 ASME Fluids Engineering Division Summer Meeting.

[3] Cooper, P., 1996, “Perspective: The new face of R and D – A Case Study of the Pump Industry”, Trans. ASME, Vol 118, December, pp. 654-664.

[4] Dahl, T., 2001, “The Evolution of the Pump Business in the Information Age”, Proc. Of ASME FEDSM’01, ASME Fluids Eng. Division Summer Meeting, New Orleans, Louisiana, May 29-June 1, 2001.

[5] Europump, European Association of Pump Manufacturers, “Attainable efficiencies of volute casing pumps”, The European Guides to Advanced Pumping Technology, Vol 2, 1999.

[6] Stoffel, B. and Lauer, B., Summary of the Final Report on the Research Project “Theoretically Attainable Efficiency of Centrifugal Pumps”, provides by the Specialised Section Pumps within the VDMA, Frankfurt, 1994.

[7] Hydraulic Institute. “Efficiency Prediction Method for Centrifugal Pump”, Parsippany, NJ, 1994.

[8] Bakir, F., “Aerodynamique et dimensionnement des turbomachines à écoulement incompressible et cavitant”. Habilitation à diriger des recherches soutenue le 16 novembre 2004. Université Pierre et Marie Curie (ParisVI).

[9] Wuibaut, G., Bois, G., Dupont, P., Caignaert, G. and Stanislas, M., “PIV Measurements in the Impeller and the Vaneless Diffuser of a Radial : Flow Pump in Design and Off–design Operating Conditions”. Journal of Fluids Engineering. ASME, September 2002, Vol 124, pp 791-797.

[10] Europump, European Association of Pump Manufacturers, “Operating Rotodynamic Pumps away from Design Conditions”, The European Guides to Advanced Pumping Technology, Vol 3, 2000.

[11] Europump, European Association of Pump Manufacturers, “NPSH for rotodynamic Pumps.”, The European Guides to Advanced Pumping Technology, Vol 1, 1999.

[12] Rey, R., Guiton, P., Kermarec, Y. and Vullioud, G., “Etude statistique sur les caracteristiques à debit partiel des pompes centrifuges et sur la determination approximative du débit critique.” La houille Blanche 1982; 2/3; 107-120.

[13] Coutier-Delgosha, O. “Modélisation des Ecoulements Cavitants: Etude des comportements Instationnaires et Application Tridimensionnelle aux Turbomachines” Thèse de Doctorat de l’INPG. Spécialité: Mécanique des Fluides et Transferts. LEGI UMR CNRS 5519. Novembre 2001.