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Energetic, structural, thermal and fatigue analysis of heavy duty process pumps.

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Abstract. Design of heavy duty process pumps usually based on the end user requirements. Operating conditions of pumps in the system dictate technical solution to reach high performance pump design. Pumps for special application like nuclear power plants, petroleum, petrochemical and natural gas industry should reach very high design criteria and have to fulfil requirements of different international standards for pumps. Usually energetic and cavitation characteristics are necessary issues of the development procedure. In this paper structural analysis that include thermo-mechanical loading and fatigue phenomena are also considered, because they are very important for estimation of long service life. Repeated thermo-mechanical loading and unloading which leads to fatigue of pumps are obtained using unsteady Computational Fluid Dynamics (CFD) with taking into account also thermodynamics equations. Complete numerical analysis is done for an example of centrifugal pump with the specific speed around n_q=24. The results show energetic characteristics, thermal stresses and deformations and maximal number of operation cycles for safe and reliable operation.

1. Introduction

Design of heavy duty process pumps usually bases on end user requirements. Operating conditions of pumps in the system dictate technical solution to reach high performance pump design.

Pumps for nuclear power plants and pumps for special application should reach very high design criteria. API 610 and ISO 13709 international standards specifies requirements for pumps for petroleum, petrochemical and natural gas industry.

Apart of standards design requirements, design of heavy duty process pumps for high temperature should meet a lot of specific design criteria as follows:

- Life time
  Pump should be designed at least for 20 years operation with 3 years uninterrupted operation (API 610 standard requirement) or 30 years operation for nuclear application.
- Operation cycles
  Operation cycles depend of pump application and end user requirements.

In this paper analyses is done for the following operating conditions:

Table 1. Operating conditions

Nominal operating condition	No. of occurrences
Heat up and cooldown of 50 °C/h	1000
Power (pressure) increase and decrease with step of 10 %	10000
Power (pressure) pulsation from 100% to 0% in one step	150
Power (pressure) increase and decrease with step of 10%	50000

- Reliability characteristics - critical reliability – probability of failure
  Reliability of the pump design should consider the time frame of probably failure compared with critical reliability.
- Design without pre-warming
  Design without pre-warming is possible for pumps with limited sizes. For large size pumps pre-warming design should provide proper heat distribution.
- Minimize thermal distortion
  Design should minimized thermal distortions. Important is to control all thermal deformations, elongations, radial and axial clearances which influence the pump reliability. All thermal deformation should be handled for wide temperature range.
- Handle thermal stresses
  Thermal stresses should be calculated according to the material characteristics limits.
  Using the quality materials of construction risk of high thermal stresses is lower.
- Optimize the temperature distribution and heat transfer
  Optimize the outside and inside geometry of pump parts is the key parameter for proper temperature distribution and heat transfer. In application of heavy duty process pumps we meet all types of heat transfer and deep analyse of each one will provide the quality design.
- Design the thermal barrier and cooling system
  Design the thermal barrier is required in many application of heavy duty process pumps. Design limits of cooling system should prevent the thermal shocks.

A quality prepared structural analysis, thermal analysis and fatigue analysis will help to fulfil design requirements.

In this paper we will show the example of complete numerical analysis for end suction pump designed for following operating conditions:

Specific speed	$n_q = 24$
Design pressure	100 bar
Design temperature	400 ° C
Life time	min 20 years
	3 years continuous operation

Material class selection is C-6 according to API 610 (12% CHR steel for casing, impeller and shaft).
Forces and moments on flanges are according to API 610.

1.1. Pump flow analysis

In pump industry the development procedure is very important for the competition with worldwide producers. The results of the new product must be obtained in as much as possible short time. The development process can be either experimental or numerical oriented. If the numerical analysis is chosen, the results can be obtained quite fast, depends on the size of computational grids and type of numerical analysis. In the literature the majority of examples use the steady state numerical analysis to predict energetic and cavitation characteristics [1]. The complete computational time using steady state analysis is very short, but the accuracy [2] of the results is not always acceptable if the model test want to be replaced successfully. The physics of the flow in the pump shows that the flow is decelerated and have strong diffuser effect. The consequence of above mentioned issues is the high probability of vortices, back flow and flow separation occurrence, especially near the impeller blades. In all cases the flow is very unsteady and using steady state analysis, it is not possible to obtain the converged solution for all operating regimes. The accuracy of steady state results is usually very questionable.

Another problem at steady state analysis is the usage of appropriate sliding interface [3] between rotating and stationary parts. Usually the number of impeller blades is quite low and the sliding interface can have a strong influence on the final results. That is why the unsteady calculations are necessary to obtain the reliable results [4]. The results of unsteady analysis and the comparison with the experimental results are presented in the paper. The comparison is done just for one pump geometry with specific speed $n_q = 24$, but quite similar results can be expected also at different geometries or different specific speed.

2. CFD analysis

*2.1. Computational domain and grid*
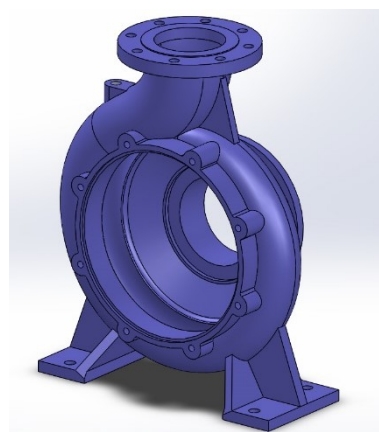
The main task was making a hydraulic design of the new pump impeller and spiral casing with good energetic characteristics and also construction of the complete pump including housing with long service life.

The geometry of analyzed pump is presented on the Fig. 1. The computational domain is slightly different because an inlet and outlet pipe is added to minimize the influence of boundary conditions. The same geometry is used for all: CFD analysis, FEM analysis and fatigue analysis.

Figure 1. Geometry of the pump casing

Only wet surfaces of the pump housing, impeller and spiral casing was taking into account for the computational grid for CFD analysis.

CFD analyses were starting with course computational grids with about 2.5 million elements and finish with very fine computational grids using more than 13.5 million elements (Fig. 2).
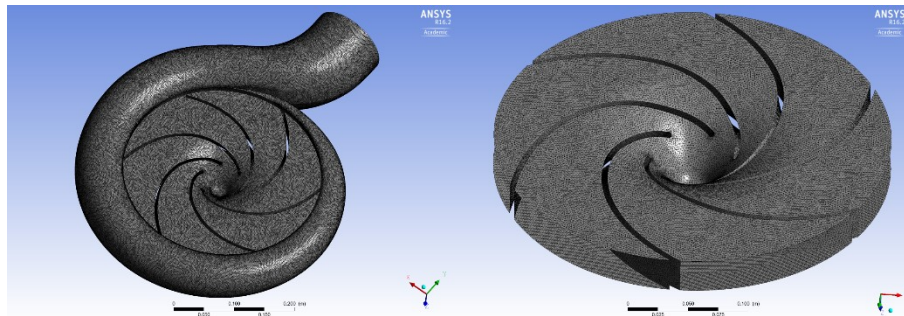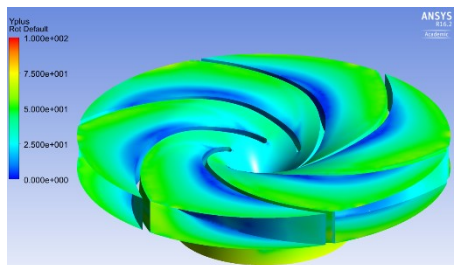
Figure 2. Computational grid

For analysed geometry the quality computational grids were generated in order to find out the accuracy of steady state and unsteady calculations and compare the numerical and experimental obtained energetic characteristics for five operating regimes.

The quality of computational grid is an important condition for accurate numerical flow analysis results, particularly in case of dominant decelerating flow in the pump where the flow is highly unsteady in majority of operating points.

Grid refinement is very important near the impeller walls. Besides the grid refinement, the special attention has been done on the grid quality parameter y^+, on mesh orthogonality and on expansion and aspect ratio. In case of analysed pump, the y^+ in the almost whole computational domain is between 20 and 50. Just in the very small area, which is not relevant on the accuracy of the results, the y^+ exceed 50 (Fig. 3).

The computational grid was done separately for impeller and spiral casing. Special attention on the mesh quality was paid to the impeller grid generation and this is the main reason that the number of elements in the impeller is around 10 million in the finest grid.

Figure 3. Distribution of y+ at the impeller

2.2. Results of CFD analysis

The numerical analysis in pump (Fig. 4 and Fig 5) has been performed for five different flow rates presented on the table 2.

Table 2. Operating regimes

Operating point (OP)	Flow rate
A	0.73 Q_{opt}
B	0.87 Q_{opt}
C	1 Q_{opt}
D	1.13 Q_{opt}
E	1.26 Q_{opt}

The important issue in CFD analysis is turbulence model. In our case the k-omega SST turbulence model was used for steady state analyses and SAS SST turbulence model for unsteady analyses. The

length of the time step is 0.0002 s for all operating points except for operating point D where the length of time step is 0.001 s.
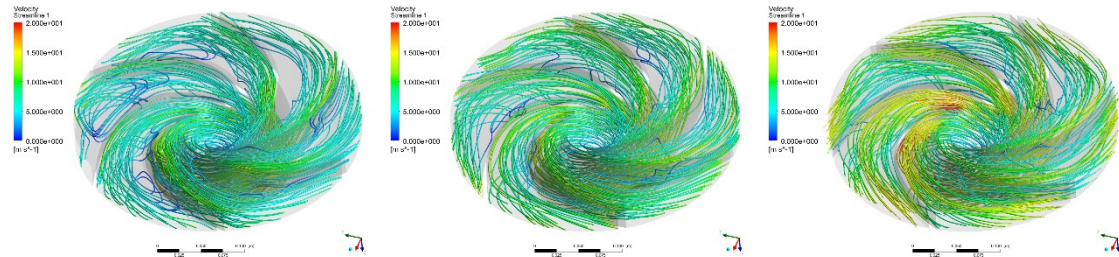

Figure 4. Unsteady flow presentation with streamlines – OP A, C, E (from left to right)
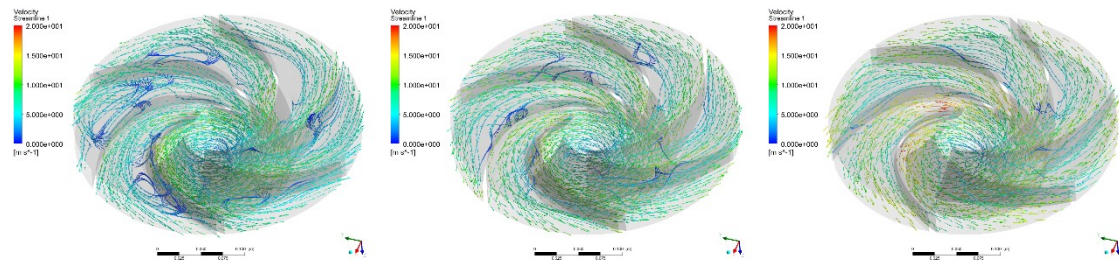
Figure 5. Unsteady flow presentation with velocity vectors - OP A, C, E (from left to right)

From the flow pattern shown on the Fig. 4 and Fig. 5, it is obvious that the flow is real unsteady. Inside the blade channels a lot of vortices and also back flow is detected. Flow conditions also influence on the efficiency characteristics and on the Fig. 6 the time dependent efficiency distribution is presented for five operating regimes. The frequency of the efficiency distribution is the same for all operating points, but the amplitudes and the shapes are different. The difference between minimal and maximal value depends on operating regime and is from 3 % to 18 %.
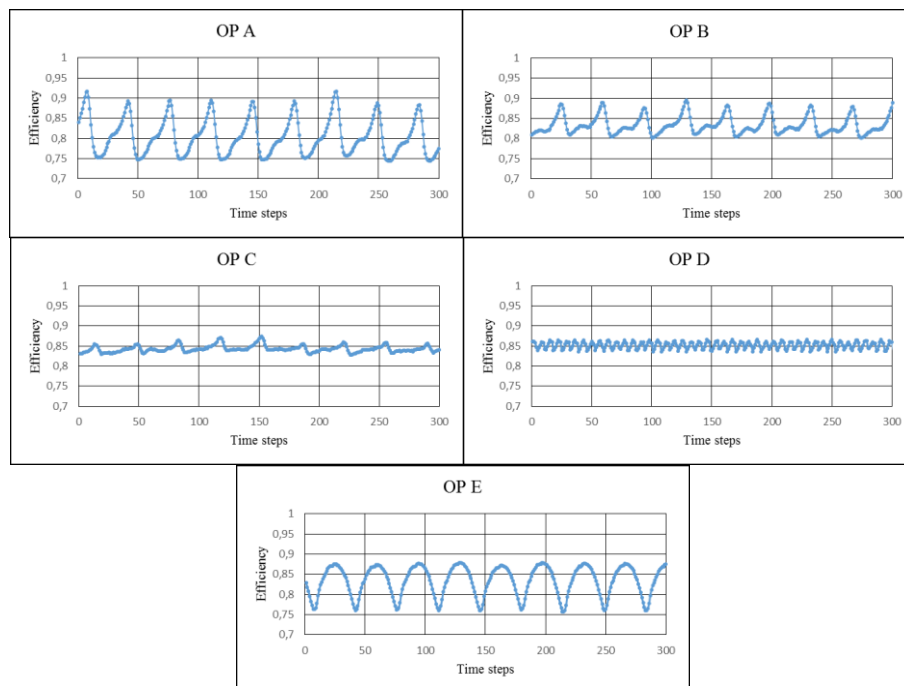

Figure 6. Time dependent efficiency distribution for five operating regimes

The time average value of efficiency is slightly different from the middle value of minimal and maximal results. Because of oscillating nature of the efficiency, it is necessary to make calculation for a reasonable number of time steps in order to obtain reliable averaged result. It is also important to monitor the convergence of the efficiency during calculation, because if only residuals is monitoring, the results can be useless.

The complete efficiency distribution for unsteady analysis is presented on Fig. 7, where experimental results are compared with both average values, minimal values and maximal ones. It is shown that the amplitudes of the efficiency are very low near optimal operating regime, at part load and full load the situation is much different. The comparison between the time average values and normal average values shows that maximal difference is around 2 %. The best result was obtained using time averaging of unsteady results.
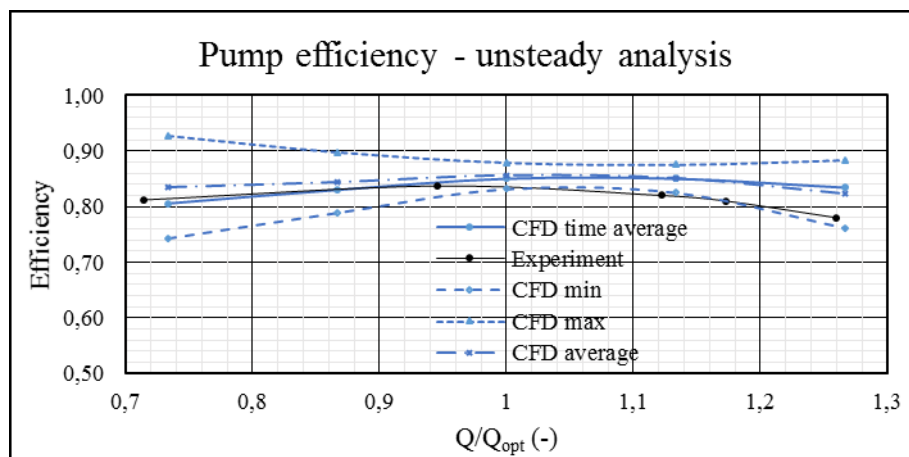
Figure 7. Comparison between experimental and CFD results for unsteady analysis

When the computational analysis is steady state, usually two options can be used, a so called 'frozen rotor' or 'stage' sliding interface between rotating and stationary parts. From the comparison of 'frozen rotor' and 'stage' sliding interface (Table 3) it can be seen that the difference of efficiency ($\Delta\eta$) is from around 2 % to 4 percent.

Table 3. Comparison of efficiency between frozen rotor and stage sliding interface

Operating point	'Frozen rotor'	'Stage'	$\Delta\eta$
A	0.810	0.853	0.043
C	0.868	0.885	0.018
E	0.852	0.876	0.024

'Frozen rotor' solution was obtained only for one position of the impeller blades without angular shift of geometry.

3. Fatigue

Metal components subjected to fluctuating loading, fail also in case when stresses are essential below the tensile strength because of cracks initiation and theirs propagation. Fatigue is basically caused by fluctuating mechanical and thermal loading. There are two well-known approaches for analysing of components subjected to fluctuating stresses [5]:
-    Stress life approach when stresses are usually low compared to tensile strength,

- Strain life approach which considers plastic deformation of components.

In this paper a stress life approach will be considered. The relationship between loading and fatigue failure is captured with a Stress-Life curve or S-N curve, also known as a Wöhler's curve. Stress-Life curve shows the relation of stress amplitude of failure.
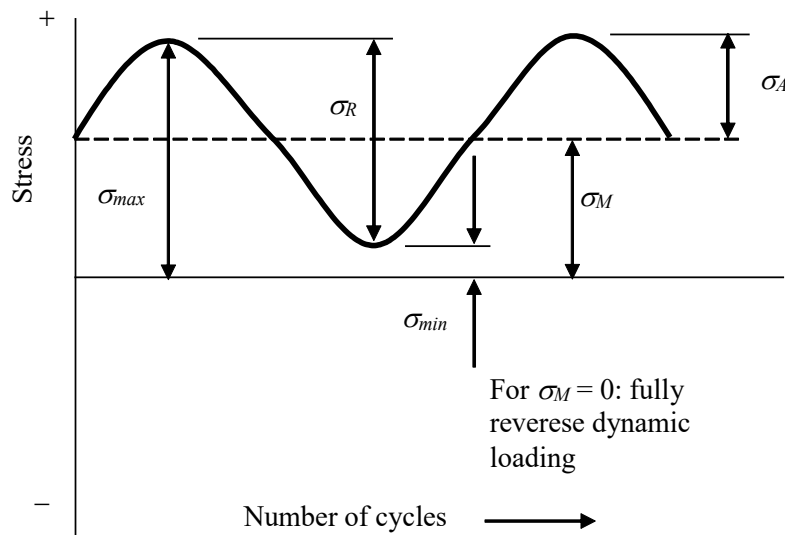
Figure 8. Cyclic loading [5]

Characteristic values (Fig. 8) are defined with maximum (σ_{max}) and minimum (σ_{min}) stress. Stress range is defined as σ_R:

$$\sigma_R = \sigma_{max} - \sigma_{min} , \tag{1}$$

where amplitude stress σ_A is equal to half of stress difference:

$$\sigma_A = \frac{\sigma_R}{2} = \frac{\sigma_{max} - \sigma_{min}}{2} . \tag{2}$$

Mean stress σ_M is half of the sum between maximum and minimum stress:

$$\sigma_M = \frac{\sigma_{max} + \sigma_{min}}{2} \tag{3}$$

Stress ratio R is defined as:

$$R = \frac{\sigma_{min}}{\sigma_{max}} \tag{4}$$

Stress ratio R can be for one way dynamic (reverse) loading equal to $0 < R < 1$, at zero based loading is $R = 0$, while at general reversed loading can be $-1 < R < 0$. Fully reversed loading occurs when an equal and opposite load is applied ($\sigma_M = 0$ and $R = -1$).

Stress-Life curves are strongly dependent on mean stress. Different mean stress correction theories (Goodman, Soderberg, and Gebert) can be used for consideration of different materials and when multiple Stress-Life curves are not available [6]. This is also important in case of non-fully reversed loading.

Loading can be proportional or non-proportional. In case of proportional loading the ratio of principal stresses is constant and the principal axes do not change over the time. Non-proportional loading occurs when the ratio of principal stress is not constant and the principal axes change over the time. This happens in case of alternating between two different load cases or nonlinear boundary conditions (contact between bodies), etc. In presented paper a constant amplitude and non-proportional loading is considered caused by pressure load and temperature changes during pump operation. There are several types of fatigue results that can be considered. In this paper equivalent alternating stress is evaluated and compared with Stress-Life (S-N) curve.

Boundary conditions and loading have a very big influence on stresses and consequently on fatigue. Adequate boundary conditions used in the model are presented in Fig. 9:
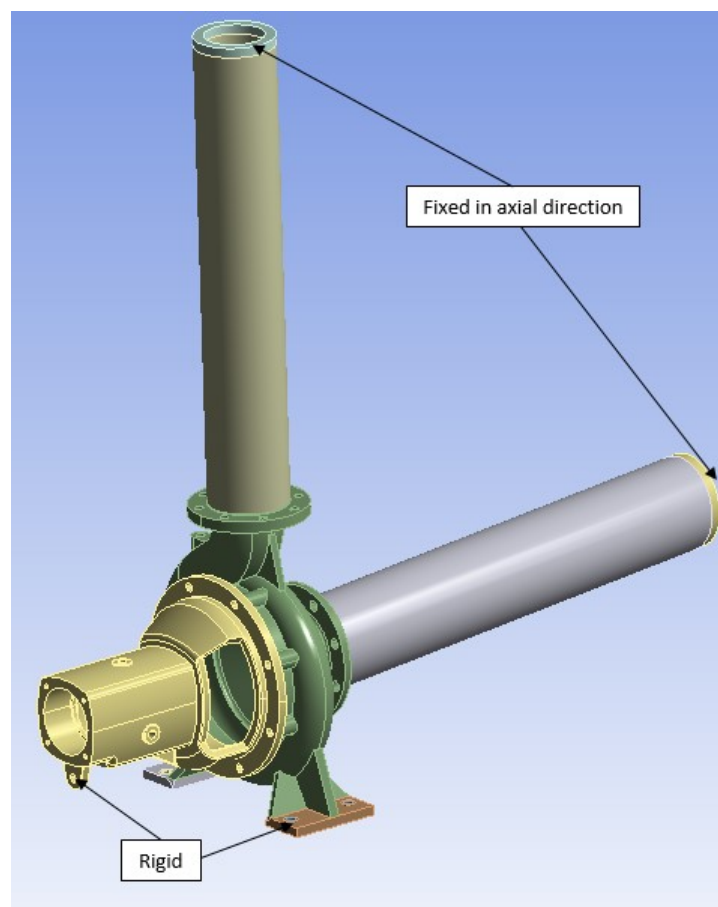

Figure 9. Boundary conditions

Results from Fig. 10 show that sharp edges have negative influence on fatigue because they increase the equivalent alternating stress.

Considering mechanical and temperature loading from Table 1 lead to alternating stress for non-proportional loading and Goodman mean stress theory presented in Fig. 10:
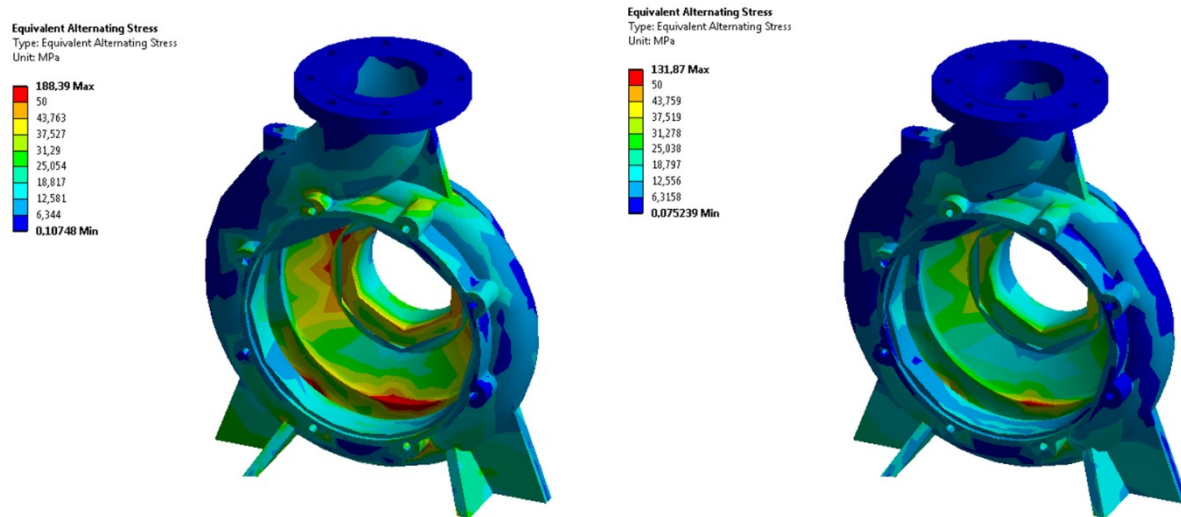
Figure 10. Equivalent alternating stress for non-proportional loading of the pump for two Fatigue Strength (Reduction) Factor - $K_f = 0.7$ (left), $K_f = 1$ (right)

To find an influence of equivalent alternating stress magnitude on fatigue a comparison with Stress-Life (S-N) curve should be done. Influence of loading cycles on properties of structural steel is presented in Fig. 11.
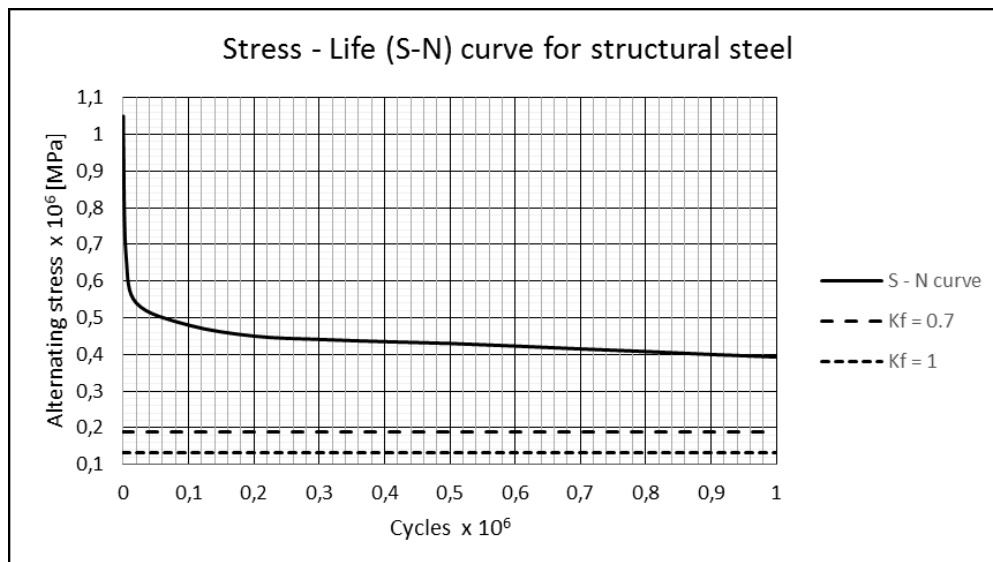
Figure 11. Stress-Life (S-N) curve for structural steel

Comparison between results from Fig. 10 and 11 shows that pump is safe against fatigue for design cycles (Table 1) and also for long time service.

4. Conclusions

Flow in pumps is usually very unsteady and that is why steady state numerical analysis is not enough reliable method in the development process. In our case the comparison of experimental and numerical results of the efficiency is presented.

Experimental results show only the average values unlike the numerical ones, where complete time dependent distribution can be obtained. The fluctuations of efficiency can be quite high, depends on operating point from 3 % to almost 18 % in this particular case. The most reliable results can be obtained using time averaging of results.

Results of the fatigue analysis show that pump can operate in safe fatigue regime for design load cycles. A very big influence on fatigue life have boundary conditions. Fixed rigidity of the pump at connections to other parts of the pump system significantly reduce fatigue life. Therefore also other parts of the pump system were considered in the model to prescribe an adequate boundary conditions. Fatigue life was evaluated using comparison of the equivalent alternating stress and stress-life (S-N) curve where also the Goodman mean stress theory was considered.

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