SYSTEM IDENTIFICATION AND LOAD ANALYSIS
OF A 20 M HIGH SMELTING FURNACE

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Abstract
At a 20 m high smelting furnace used for copper recycling strong vibrations occurred caused by the burner operation.
To assess the vibration load the following investigations were done:

- modal analysis based on FEM calculations,
- finding measuring points from the results of the FEM calculations,
- vibration measurement for system identification and estimation of the maximum dynamic amplitudes,
- adjusting the FEM model with the measured eigenforms,
- presetting the deformation in the FEM model from the measured dynamic amplitudes to estimate the stress,
- examining the stresses for operational stability,
- propose counter measures to reduce the stresses.

1 Introduction
1.1 Vibration in structures of civil engineering
In most cases the loads in structures of civil engineering are static or cyclic and no mass inertia is involved. Floor vibrations due to railway- and street-traffic are only important for the residents or sensitive equipment – not for the safety of the building. By contrast plant constructions are subject to permanent or periodical vibrations due to machines or processes. In these cases, especially with high forces, resonance of force and structure connected to less damping like in steel structures, the amplitudes reach high values and fatigue behaviour has to be investigated.

1.2 Case study of a copper furnace with strong vibration caused by the burner
The case study presented here looks at a smelting furnace used for copper recycling. Smelting copper ore usually is an exothermic process, but in a recycling furnace it is necessary to heat the copper scrap with an oil or gas burner. The vibration is caused by a feedback loop of the system blender-burner-flame-combustion chamber. Under certain boundary conditions of blender and burner the flame is instable, vortexes are produced and pressure fluctuations affect the combustion chamber. It is a self-controlled vibration actuated by the flame instability [2], [3], [4], [5]. Due to the pressure fluctuation the plant construction moves in forced vibrations or in resonance vibrations.
In this case there is a gas or oil burner lance used. Fuel and air are mixed in a jet resulting in a jet diffusion combustion. Other burners are working with premixing combustion. In both cases vibration occurs. Figure 1 shows the structure of a methane jet diffusion flame as a simple example. When trying to reduce vibrations in plant constructions there often are limitations to what can be investigated and altered. In this case the burner must not be investigated.

1.3 Plant construction – copper furnace

The furnace is a tall cylinder nearly 20 m high lined with chamotte slabs on the inside and cooled with water form outside. There is a top cover with water filled cooling pipes and the beginning of the flue gas pipe. The burner is formed like a lance and hangs in the furnace.

The copper scrap is filled into the furnace, heated by the burner to smelt and than separated by casting copper, other metal fractions and slag. This procedure takes some hours, but the strong vibrations occur only for a limited time – which is important for the fatigue investigation.

2 Modal analysis – FEM calculation

2.1 Model

First modal analysis was done with FEM calculations. The construction is only fixed at the bottom on a block foundation. The furnace is made of steel. To protect against the heat the furnace is lined with chamotte slabs. From these material combination a uniform behaviour is not to be expected. For simplicity the FEM model is reduced to the stiffness of the steel and the combined mass of the steel and the chamotte slabs.

2.2 Eigenforms and Eigenfrequencies

The fundamental eigenfrequencies are close together starting at 5.2 Hz and 5.3 Hz both with breath modes, followed by two bending modes at 6.2 Hz and 6.7 Hz. Figures 2-3 show the 3rd and 4th modes in isometry and top view.
3 Vibration measurements

3.1 Measuring points and technique

From the results of the FEM calculations the measuring points were defined. To cover the eigenforms it was necessary to use measuring points on the top in radial and tangential directions and in the circular cross-section in radial direction. The measuring points are shown in Figure 4-5.

The measurement technique used consists of electrodynamic velocity sensors, an 8-channel measuring amplifier, an A/D-converter, a measuring software and a hard disk. The measuring amplifier from Dr. Kebe Scientific Instruments has an analogue compensation of the sensor response curve in amplitude and phase according to the rules of DIN 45669 with a linear behaviour in a frequency range from 1 Hz to 315 Hz. Because of the measuring amplifier it was enough to use a 12-bit A/D-converter from National Instruments. To have sufficient data for analyses like differentiation or integration, the data were collected with 2048 samples per second. To monitor, automatic trigger and collect the signals on hard disk the measuring software from Dr. Kebe Scientific Instruments was used.

The environmental conditions in the plant are difficult because of high temperature and smoke. The possibilities to mount the transducers were limited and the wiring was difficult. Especially taking into account a measuring duration of one week with non stop plant operation.
Figure 4: lateral view of the furnace and measuring points

Figure 5: Top view of the furnace and measuring points

3.2 Measuring results

In the frequency domain there is a single wide frequency band starting at 4.5 Hz and ending with 7 Hz. Because of the spectra of turbulent flow from the burner there is no single excitation frequency. The furnace structure shows its fundamental eigenfrequencies in this range mentioned above. The furnace construction lined with chamotte slabs and smelting copper will yield a high damping.
For system analysis the measurement results were analysed detailed in the time domain. In Figure 6 the time and frequency domain is shown for four channels. It was necessary to compare the signals of all channels with each other and with the results of the FEM calculations. In the end it was found that vibrations were forced in the 3rd and 4th eigenform.

For present projects with colleagues we are using the software Artemis for operational modal analysis. Operational modal analysis enabled us to interpret the structure motion more reliably. But one has to keep in mind that operational modal analysis requires more measuring points which increases the effort.

**Figure 6: Measuring – time and frequency domain**

Over a duration of one week with non stop operating furnace the maximum dynamic amplitude of 90 mm/s peak value was detected. The occurrence of high amplitudes can be calculated from the collected data. In one week there were high amplitudes for a accumulated duration of 60 s. Considering a frequency of 7 Hz there is an occurrence of 420 high amplitude events a week and 21 840 a year. The lifetime of the furnace is given with 25 years, so there will be 546 000 load-cycle changes at the above mentioned high dynamic amplitude.

In the measuring software presently used implemented a function allowing to classify the maximum amplitude between each zero-crossing in the time domain. This function works permanently in the background in real time and stores the occurrences in twenty classes of amplitude ranges. This feature provides the possibility to investigate the fatigue behaviour more detailed with the use of load collectives.
4  Stresses, fatigue analysis and counter measure

4.1 Adjusting the FEM model based on the measured vibrations

In order to estimate the stresses in the steel structure of the furnace first the FEM model has to be adjusted based on the measured vibrations. Therefore the measured signals were analysed in detail in the time domain and compared with the results of the FEM calculations. It was necessary to adjust the influence of the masses from the chamotte slabs and the copper in the furnace.

4.2 Presetting deformation in the FEM model to estimate the stresses

In the FEM model the deformation was preset based on measured displacement. From this the stresses can be calculated. The hot spot of stresses was in a horizontal non-through-welded butt joint. According to Eurocode 3 Design of steel structures [6] for this stress concentration there is an estimated value at $\Delta \sigma = 260 \text{ N/mm}^2$.

4.3 Examining the stresses for operational stability – fatigue analysis

In Eurocode 3 is gives a stress variation range $\Delta \sigma$ against maximum allowed load-cycle changes $N$ and which is shown in Figure 8. A stress variation range more than $\Delta \sigma_L$ has to considered in the load collectives and is called the threshold of fatigue strength. The endurance strength is designated $\Delta \sigma_D$. Between the threshold fatigue strength $\Delta \sigma_L$ ($N_L = 2 \cdot 10^9$) and endurance strength $\Delta \sigma_D$ ($N_D = 5 \cdot 10^6$) there is a power law with $m=5$. The fatigue strength is $\Delta \sigma_C$ ($N_C = 2 \cdot 10^8$), up to $N_D = 5 \cdot 10^6$ there is a power law with $m=3$.

For load-cycle changes at $N = 2 \cdot 10^6$ according to Eurocode 3 for this stress concentration there is a given maximum allowable stress variation range $\Delta \sigma_C = 80 \text{ N/mm}^2$. The estimated for the furnace value is $\Delta \sigma = 260 \text{ N/mm}^2$. The maximum allowable load-cycle change of an amplitude $\Delta \sigma = 260 \text{ N/mm}^2$ can be calculated:

$$N = \frac{1}{\left(\frac{260 \text{ N/mm}^2}{80 \text{ N/mm}^2}\right)^3 \cdot 2 \cdot 10^6} = 58260$$
At about 21 840 load-cycle changes in one year the allowed value of 58 260 is reached in approximately three years. Therefore it was strongly recommended to reduce the stress in the hot spot area.

Figure 8: Standardised Wöhler-curve stresses against load-cycle changes according to Eurocode 3 DIN EN 1993-1 [6]

4.4 Proposed measures to reduce the stresses

The load has to reduced without altering the operation, especially the burner. So the vibration excitation and the quantity of load-cycle change could not be reduced. It was not possible to fix the structure or reinforce the furnace from outside in order to reduce the stresses.

The weak areas are the horizontal non-through-welded butt joints. In these parts the stresses are resulting from the stresses in longitudinal direction and additionally from a torsional strain because of the asymmetric welded joints. The only practical way to reduce the load is to avoid the torsional strain. Therefore the proposed measure is to fill the gaps and make through-welded joints.

The non-through-welded butt joints are on the outside of the furnace wall and the filling gaps on the inside. To protect the furnace against the heat of the smelted copper it is lined out with chamotte slabs. Because of erosion the chamotte slabs have to be changed regularly. During the changing after removing the chamotte slabs the possibility to reinforce the welded joints is given.
5 Conclusion

It was shown that the vibration excitation of the copper recycling furnace is caused by a self-controlled vibration actuated by the flame instability in the burner. The pressure fluctuation is leading to vibrations of the furnace walls. The fundamental eigenforms were calculated with a FEM model and used to define the measuring points. From measured results the maximum amplitudes and the load-cycle change for these amplitudes were given. With the adjusted FEM model a load analysis was done by presetting the dynamic deformations. The load analysis shows hot spots with high stresses at horizontal non-through-welded butt joints. A fatigue analysis shows that for this situation the number of load-cycle changes is limited. To reduce the load altering the operation especially the burner was not possible. The proposed measure was to reinforce the furnace wall by making through-welded butt joints. The weak side of the welded joint are inside and because the furnace is lined out with chamotte slabs not easily accessible. During regular changing of the chamotte slabs it is possible to reinforce the welded joints.

For present and future investigations we are using the technique of operational modal analysis to enabled us to interpret the structure motion more reliably. Working on fatigue behaviour the vibration monitoring we are presently operating uses a permanent classification of the load-cycle change. Starting from the counted load-cycle changes the fatigue analysis could be done considering the load collectives in more detail.

6 References


