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## Fatigue Design for Scroll Compressor Wraps

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### ABSTRACT

Understanding the mechanical behavior of compressor components under cyclic load (as the compressor runs) is one of the most important considerations when designing the compressor. Properly modeling and predicting the mechanical behavior in Fatigue has an influence in the resulting compressor reliability. Thus, the Fatigue design of the scroll involutes using Finite Element Analysis (FEA) coupled with experimental measurements provides confidence about the reliability of the compressor, starting with the initial steps of involutes design. Such technique allows the reduction of precious development and qualification time.

## 1. INTRODUCTION

### 1.1 Overview

Designing a new product requires to check the resistance to static loading, but also to consider the fatigue phenomenon, which has a lots of influence in case of cyclic loading.

In 1829 Wilhelm ALBERT carried out the first tests regarding the damage of materials by alternative loading. In 1839 Jean-Victor PONCELET used the French word "fatigue" to describe the degradation of the material properties under cyclic loads. And in 1852 August WÖHLER was the first engineer to discover the link between the reliability and a cyclic stress. He used for the first time a diagram of acceptable stress S versus the number N of applied load cycles. This is now known as the SN diagram.

Fatigue is also studied at the microscopic scale. Investigations showed that fatigue destroys the material through cracks propagation. Cracks are initiated by invisible stage I cracks due to shear stress, because locally the geometry leads to a stress concentration which is enough to exceed the elastic limit. Then visible stage II cracks may appear and grow inside the structure of the material. Each cycle of loading contributes to a small progression of the crack.

Several parameters have an influence on the crack growth, like the relation between stress and strain, and the amplitude of tensile or compressive effects during a cycle. Surface finish and surface treatment also play a role.

One surprising consequence linked to fatigue is the overloading effect. Such a sequence applied once can close the end of the crack and this delays or stops the crack propagation for lower loads. In the same way, auto-patching materials have been being developed and this addresses fatigue in priority.

The fatigue analysis applied to compressor parts is useful for all parts which experience a cyclic load. This is a mean to determine the expected lifespan of the product. Because the fatigue analysis relies on the part geometry (i.e. stress concentration) and the choice of material (i.e. microscopic structure), this does not always lead to a heavier part.

## 1.2 Lifespan

All products have a minimum expected lifespan, compatible with a normal use of the product and the customer's needs. The few examples hereafter deal with common systems and this shows how large the differences in expected life are.

Washing machine	2000-2500 washing cycles	about 4,000 hrs
Car	250 000 km @50km/h	about 5,000 – 6,000 hrs
	180000 miles @30mi/hrs	
TV set	no mechanics	20,000 hrs
Danfoss Compressor	10 years @66% running time	60,000 - 65,000 hrs
	15 years @50% running time	

It appears that the demand in lifespan is relatively long for compressors. Machines with a cyclic load are subject to fatigue and this is to be taken into account in order to achieve an adapted lifespan.

## 1.3 History of fatigue analysis with Danfoss

Since early 2000's, Danfoss commercial compressors has implemented these different steps of fatigue analysis :

- Determination of fatigue data by experiments for cast iron
- Determination of an analytical criterion based on lab tests for the wrap
- Building of a computer program which applies automatically the boundary conditions for the wraps over one turn of the crankshaft in a FEA model
- Dang Van fatigue analysis from FEA results with the assumption of a harmonic stress cycle
- Dang Van fatigue analysis for the wraps from FEA results with a real stress cycle

## 2. CYCLIC STRESS INSIDE A SCROLL COMPRESSOR

### 2.1 Geometry

The compression process generates cyclic stress inside the scroll set (fixed scroll wrap and orbiting scroll wrap).

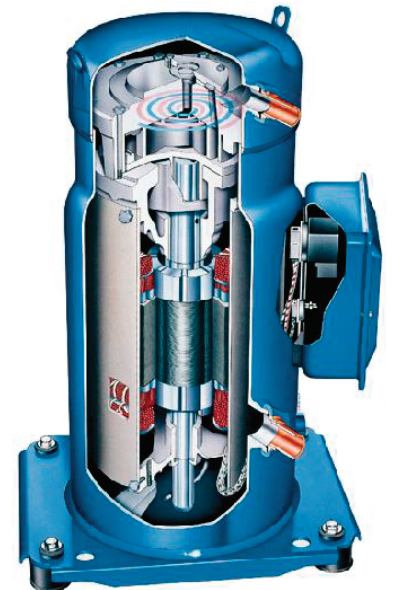
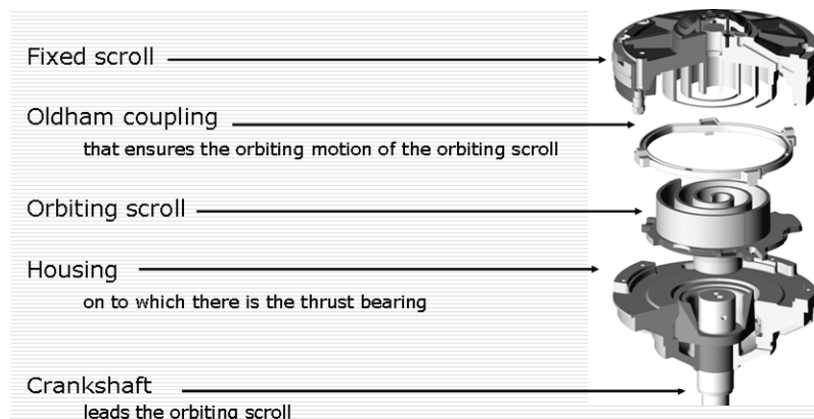


Figure 1 : scroll set assembly

## 2.2 Internal pressure loads

The duties of the involutes come from pressure pockets. Indeed, during a compression cycle, pressure pockets are created inside the involutes.

- **Suction - 1<sup>st</sup> orbit**
- As the bottom scroll orbits, two refrigerant gas pockets are formed and enclosed
- **Compression - 2<sup>nd</sup> orbit**
- The refrigerant gas is compressed as the volume is reduced closer to the centre of the scroll
- **Discharge - 3<sup>rd</sup> orbit**
- The gas is compressed further and discharged through a small port in the centre of the fixed scroll

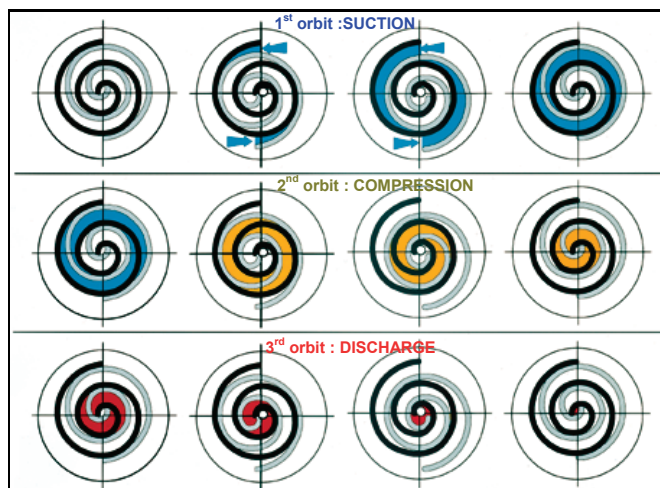
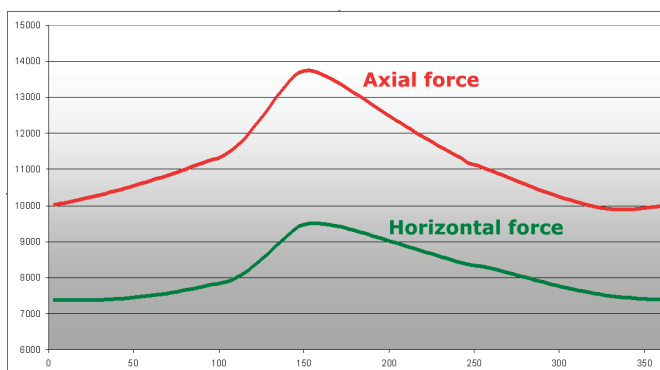
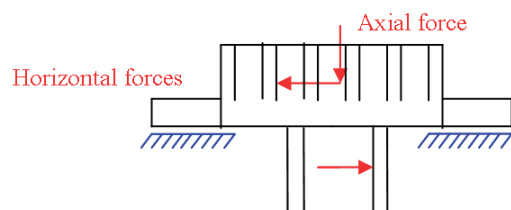


Figure 2 : compression process

## 2.3 Variations of the internal forces

The result of the compressive forces breaks down into a thrust load and 2 horizontal forces, these forces vary during one turn of the crankshaft (as it is shown in the following graph)



Graph 1 : Variations of the internal forces

## 3. FEA SIMULATION MODEL

### 3.1 General points

These mechanical calculations were carried out with Ansys Software. This analysis can be led for both orbiting scroll and fixed scroll.

For orbiting scroll analysis, we assumed that the thrust bearing surface is rigid and that the orbiting scroll is flexible i.e. the orbiting scroll can experience deformation under loading : this is a non linear static mechanical approach. Opposite to the reality, in the FEA model, the thrust bearing surface moves in relation to the orbiting scroll which is fixed.

For the fixed scroll analysis, this is a linear static mechanical approach, the fixed scroll is flexible i.e. the fixed scroll can experience deformation under loading.

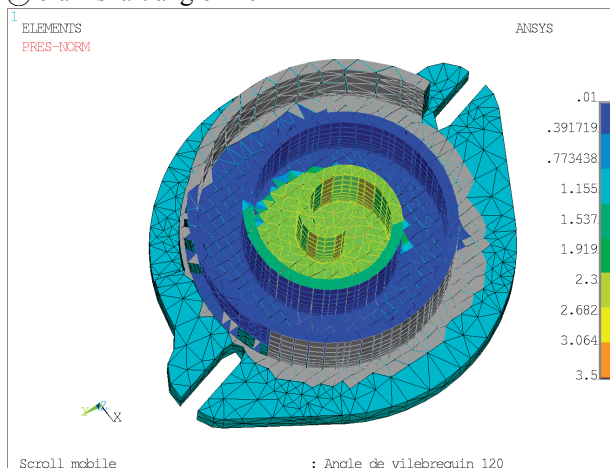
### 3.2 Boundaries conditions : pockets pressures

To simulate precisely a turn of the crankshaft, calculations were carried out every 3° of crankshaft angle. Our thermodynamic simulation software gave the pocket pressure position and pocket pressure for each crankshaft angle. An issue was to apply the boundaries conditions of the pocket pressure for each angle of crankshaft. In fact the meshing of the orbiting scroll remains the same for all calculations, whereas the pocket position and pocket pressure change during a turn of crankshaft. A computer program was written to apply automatically, and independently of the grid, the pressure inside the involutes. The difficulty in this process is that the elements boundaries did not correspond to the pocket pressures boundaries.

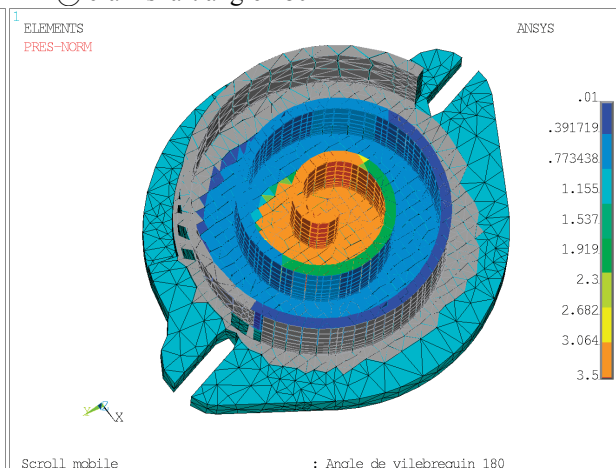
Note : for the following results, only 6 positions out of 120 were post-treated in this document.

Boundaries conditions on an orbiting scroll :

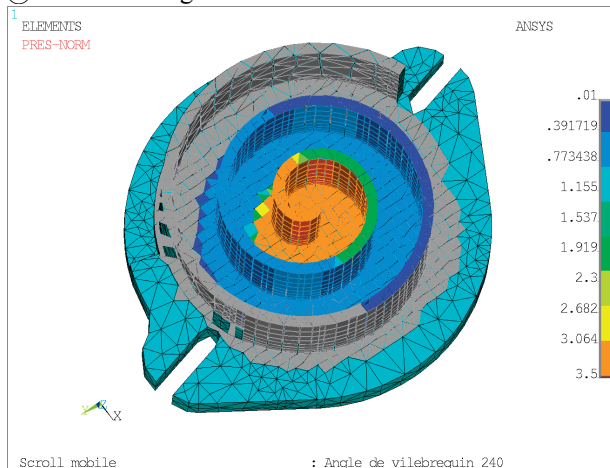
@ crankshaft angle 120°



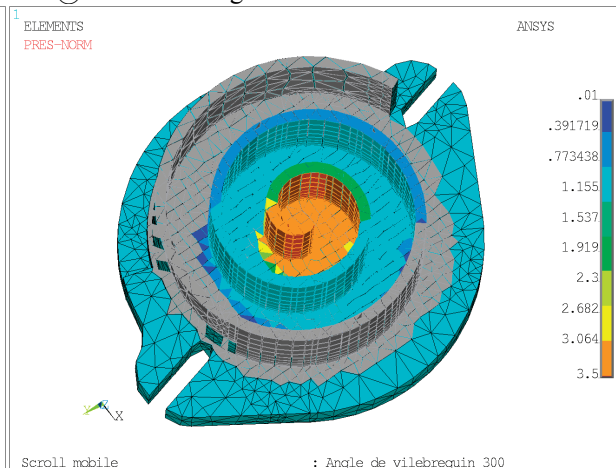
@ crankshaft angle 180°



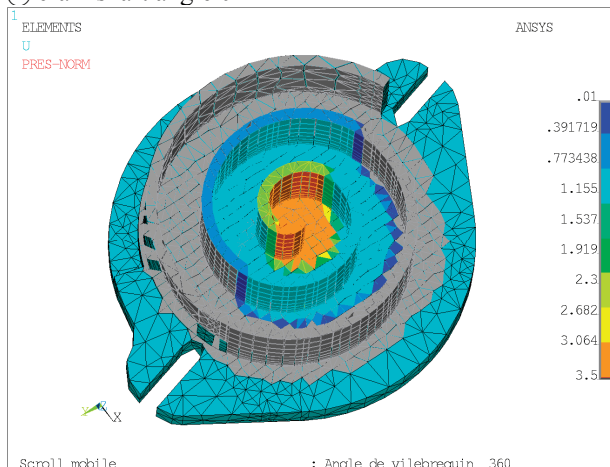
@ crankshaft angle 240°



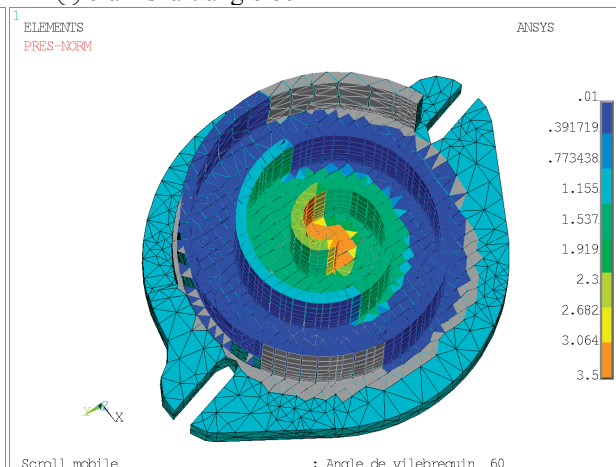
@ crankshaft angle 300°



@ crankshaft angle 0°



@ crankshaft angle 60°



Note : The colors correspond to pressure levels

### 3.3 Employing Submodeling for a specific hotspot

One of the sensitive locations is the connection between the center of the wrap and the baseplate. To obtain a precise stress matrix, we have employed submodeling. This method consists to create a local model with a finest meshing in order to obtain a more precise stress matrix. Moreover all tiny radii or geometrical features can be taken into account in this local submodel.

The stress at the connection between the wrap and the baseplate depends on the meshing and on the tiny geometrical features. With the global model it's difficult to obtain a good stress matrix representation. It's shown in the graph beside that the variation of the stress along a vertical line is discontinuous while approaching the baseplate due to FEA numerical approximation (cf. red dotted ellipse).

The evolution of a local submodel for the connection between the center of the wrap and the baseplate is shown hereunder.

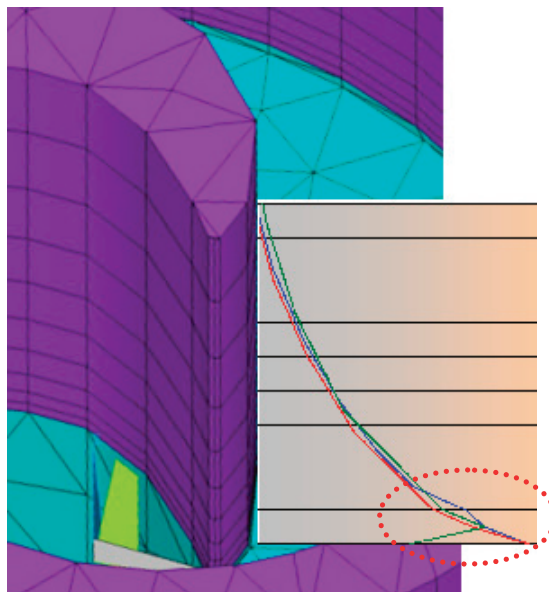
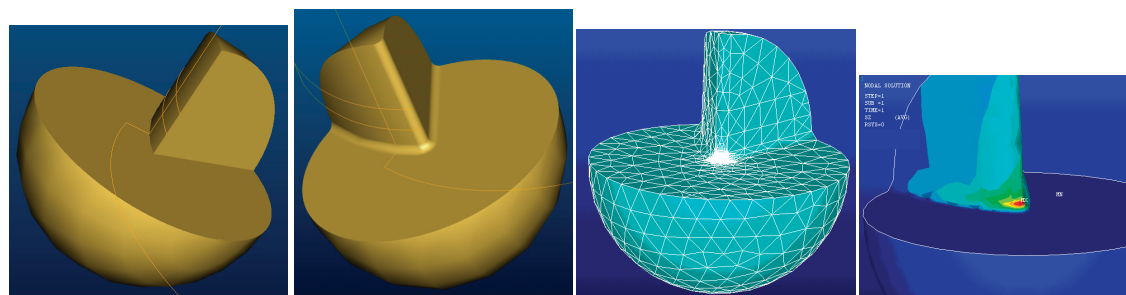


Figure 3 : variation of the stress along a vertical line



1-Simplified geometry

2-Machining tool radius

3-Very fine meshing

4-Precise stress matrix

Figures 4 : submodeling process

## 4. FATIGUE ANALISYS

### 4.1 Fatigue analysis software

nCode DesignLife is used as software tool for fatigue analysis. The system provides fatigue life prediction from finite element analysis. Various fatigue processing models are implemented – SN, EN, Seam Weld, Spot Weld, Multiaxial, transient vibration, linear, static, modal superposition and more to durability and certification analysis.

### 4.2 Fatigue analysis criterion

Following the stress type and intensity, there are different approaches to proceed to a fatigue analysis. It is possible to classify depending on uniaxial or multiaxial stresses, periodic or variable cycles, below or above yield strength of the material.

uniaxial stresses	periodic stresses	Low stress level	Haigh Diagram (or Goodman) and Wöhler curve; Gradient type Method
		Intermediate stress level	Neuber Rules; Smith-Topper Method; stress intensity factor method
		High stress level	Manson-Coffin Curves ; Fracture mechanics
	variable cycles stresses	Low stress level	Same for periodic stresses with Cycle summation method and Damage summation rule
		Intermediate stress level	
		High stress level	
multiaxiales stresses	periodic stresses	Low stress level	Fatigue criteria expressed in stress or strain and based on Wöhler curve; <b>DangVan criterion</b>
		Intermediate stress level	
		High stress level	
	variable cycles stresses	Low stress level	Same for periodic stresses with Cycle summation method and Damage summation rule
		Intermediate stress level	
		High stress level	

Table 1 : Fatigue issues classified by types

### 4.3 Choice of the fatigue criterion

In most of cases mechanical component experiences multiaxial solicitations. For scroll wraps, we deal with high cycle fatigue and the solicitations are multiaxial, periodic but not harmonic. For this type of solicitations, 2 families of fatigue criteria can be used :

- criteria with a global approach where stresses are averaged in a considered volume
- criteria with a critical plane approach where stresses are averaged in a critical face

Moreover, in our case the principal direction of stress matrix could be considered fixed, so it's preferable to choose a criterion with a critical plan approach. A Dang Van criterion has been chosen because automotive industry has been using it for a long time and because only 2 values are needed to define it.

### 4.4 Dang Van analysis

The Dang Van method is designed to provide a safety factor calculation for multiaxially loaded components in the endurance regime (large number of cycles to failure). The theoretical basis can be summarized as follows :

- Fatigue crack initiation normally occurs due to repeated plasticity on shear planes in individual grains.
- The most important factor as to whether a crack will propagate past the first grain and go on to cause fatigue failure is the microscopic shear stress in critical grains.
- The ability of a shear crack to propagate is modified by the hydrostatic stress (which can increase damage by opening existing cracks).
- The state of stress in a grain where repeated plasticity is occurring will be affected by a process of shakedown.

The Dang Van analysis process is summarized in this graph :

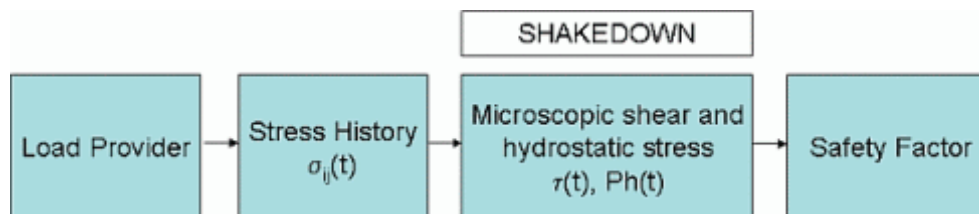


Figure 5 : Dang Van analysis process summary

The Dang Van analysis engine requires a FEA derived stress-time history as input.

The material parameters required for a Dang Van analysis are as follows :

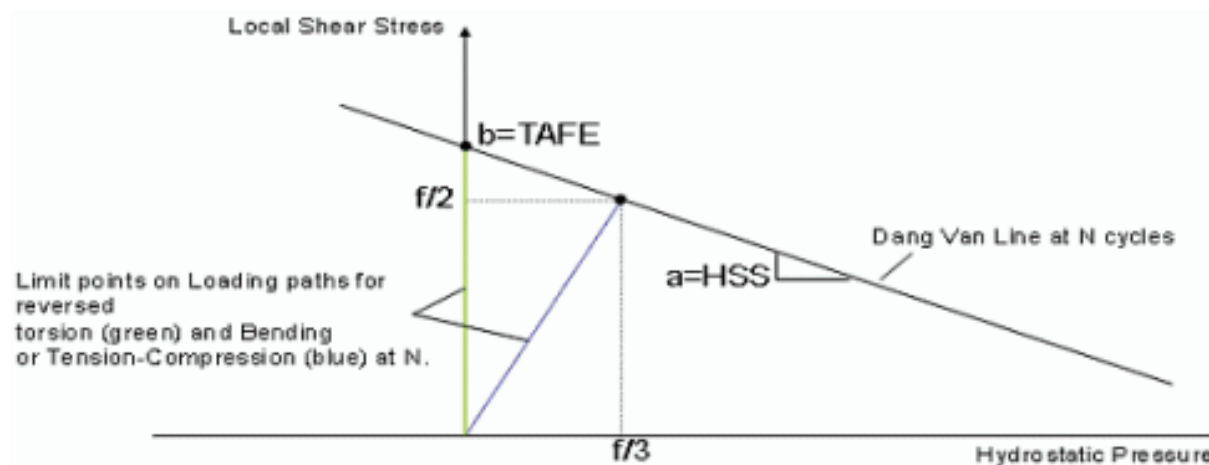
- UTS Ultimate Tensile Strength
- E Modulus of elasticity
- TAFE Type A Fatigue Endurance (MPa). This is the shear stress amplitude at the fatigue limit under pure shear (torsion) loading, usually noted “b” in discussions about the Dang Van method.
- HSS This is the Hydro-Static Sensitivity factor, usually denoted “a” in discussions of the Dang Van method

The Dang Van parameters a and b (TAFE and HSS) define a fatigue threshold condition which is described by the equation  $\tau + a.P_h = b$  where  $\tau$  is the microscopic shear stress and  $P_h$  is the hydrostatic pressure.

The material parameters a and b can be determined if the fatigue stress limit amplitude is known under two different loading conditions, with the most common being uniaxial tension-compression or bending and pure shear, both at stress ratio  $R = -1$ .

In the case of a pure shear test, at the fatigue limit,  $P_h = 0$  and  $\tau = b = \text{TAFE}$ .

For a tension-compression or bending test, at the fatigue limit  $f$ ,  $P_h = f/3$  and  $\tau = f/2$ . This leads to : 
$$a = \frac{3(\text{TAFE} - 0.5f)}{f} = \text{HSS}$$



Graph 2 : Dang Van threshold line

The Dang Van criterion is a three dimensional multiaxial fatigue limit criterion dealing with high cycle fatigue conditions where the crack initiation occurs at a microscopic level. The method assumes that around the fatigue limit, cyclic plasticity will occur in critically oriented individual grains, and therefore local stress tensors need to be considered. These microscopic stresses differ from the macroscopic ones by the presence of a microscopic residual stress field and near the fatigue limit tend towards a pseudo shakedown state.

The Dang Van method states that, at the stabilized state (shakedown state), crack initiation will happen whenever a function of the microscopic stresses exceeds a threshold at any time in a stabilized cycle. The formulation uses a linear combination of the current local shear stress  $\tau(t)$ , creating plasticity, and hydrostatic pressure  $p(t)$ , acting on opening of micro-cracks, to calculate an “equivalent” stress and compare it to a limit.

This criterion is based on the determination of critical plane where  $|\tau(t)| + \alpha \cdot p(t)$  is worst i.e. highest. The tensor of macroscopic stresses is simplified from 3D to 2D issue by using this critical plane. So Dang Van criterion is a Macro-Micro approach in High-Cycle Multiaxial Fatigue.



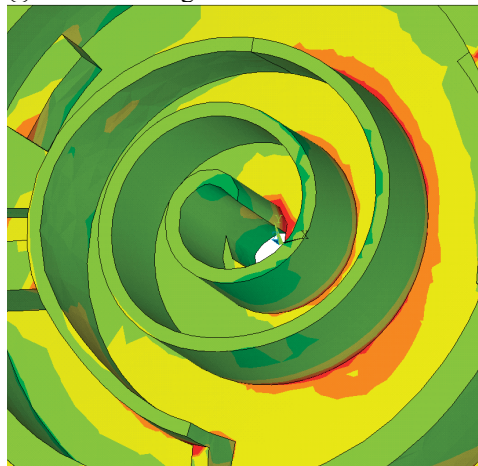
## 5. FEA SIMULATION + FATIGUE ANALYSIS RESULTS

### 5.1 Stresses inside a wrap obtained by FEA simulation

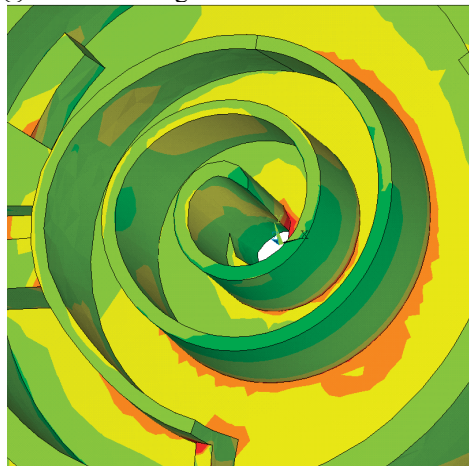
The first result is the visualization of the stresses of the fixed scroll on one turn of a crankshaft.

Note : for the following results, only 6 positions out of 120 were post-treated in this document.

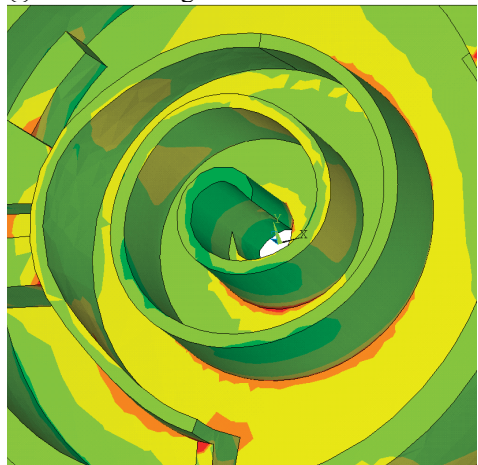
@ crankshaft angle 0°



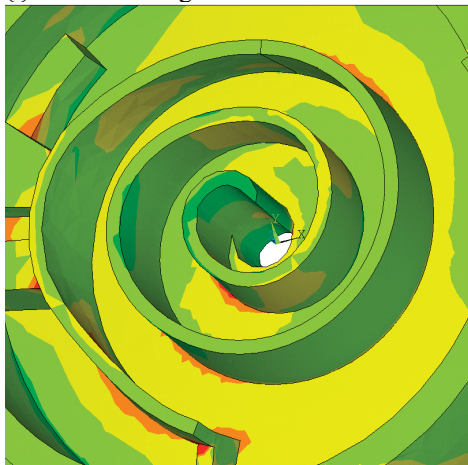
@ crankshaft angle 60°



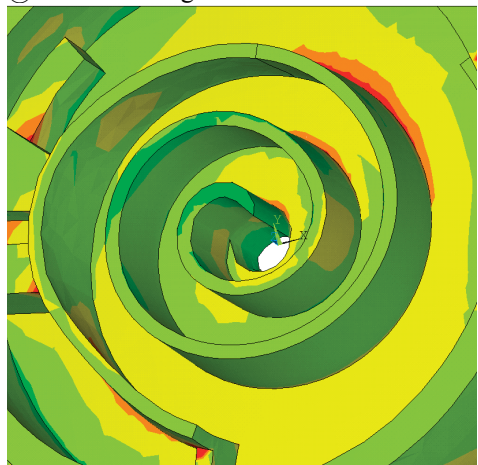
@ crankshaft angle 120°



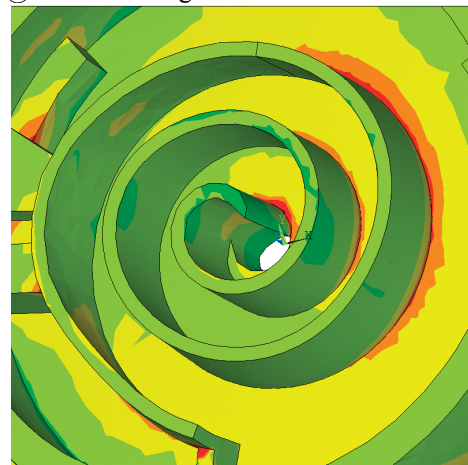
@ crankshaft angle 180°



@ crankshaft angle 240°



@ crankshaft angle 300°

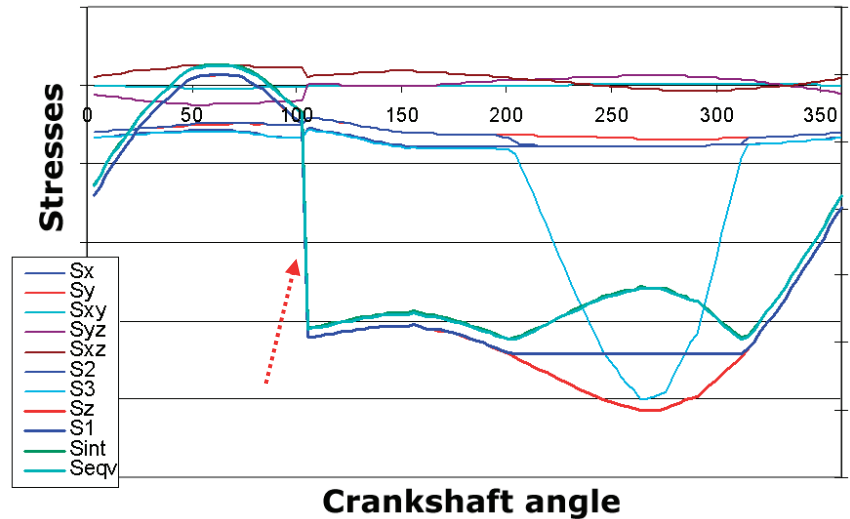


**5.2 Variation of stress matrix components at a node.**

This graph allows to visualize the variation of stresses during one turn of the crankshaft. None of stress tensor components are null, then the multiaxiality of the stress matrix is confirmed.

The variation of the stress is not harmonic. (no sinus shape).

There is during the compressing process a step (cf. red dotted arrow) due to the opening of the compressed gas pocket toward the high pressure volume. At this time, a sealing contact switches to opening.



Graph 3 : Variation of stresses on one turn

**5.3 Variation of stress at the connection between the center of the wrap and the baseplate.**

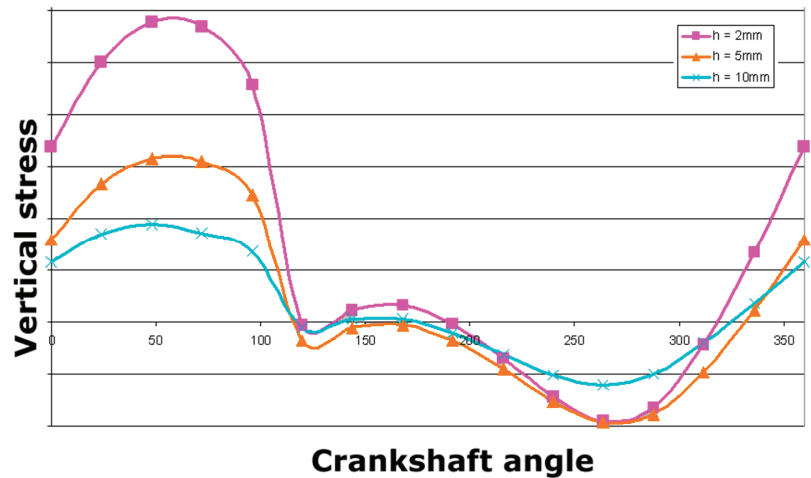
The stress toward the vertical direction is analyzed.

The vertical stress at this location is either positive (traction) or negative (compression) during one turn of the crankshaft.

The three curves correspond at different elevations from the baseplate.

Of course, the stress is more important while approaching the baseplate due to “cantilever effect” of the wrap on the baseplate.

The maximum traction stress is observed at a little rotation just before the opening of the compressed gas pocket toward the high pressure volume.



Graph 4 : Variation of stress at different heights

## 5.4 Dang Van fatigue analysis results.

### 5.4.1 Safety factor

One of the Dang Van fatigue analysis results is the Safety Factor which is defined by :

$$SF = \frac{b}{\text{Max}(\tau(t) + a \times p(t))}$$

From the FEA loadcases results, for each node of the FEA model, this analysis is led to obtain a 3D view of the safety factor in the part. It allows to identify the critical location.

Knowing this, we are able to improve our design to increase this safety factor.

### 5.4.2 Loading path

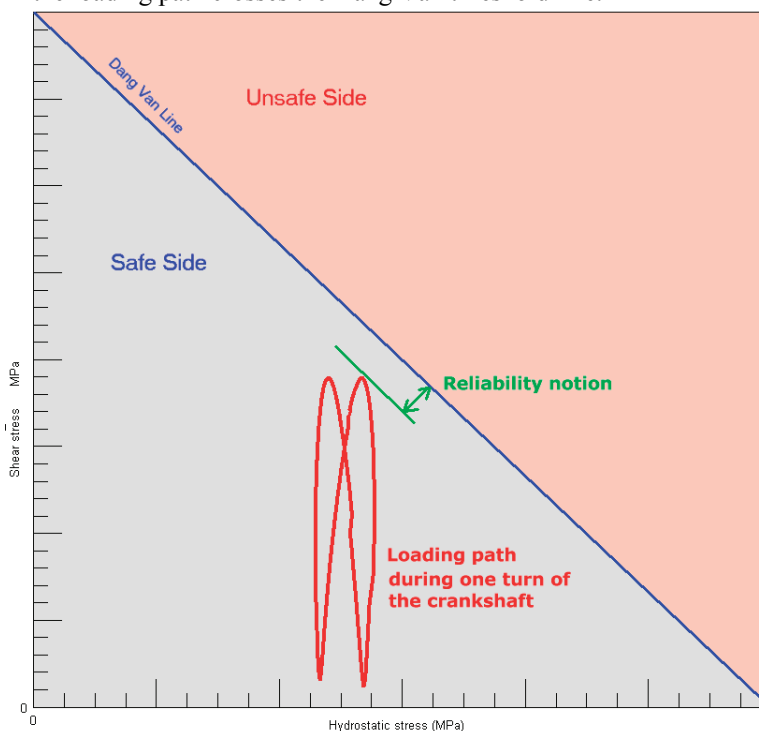
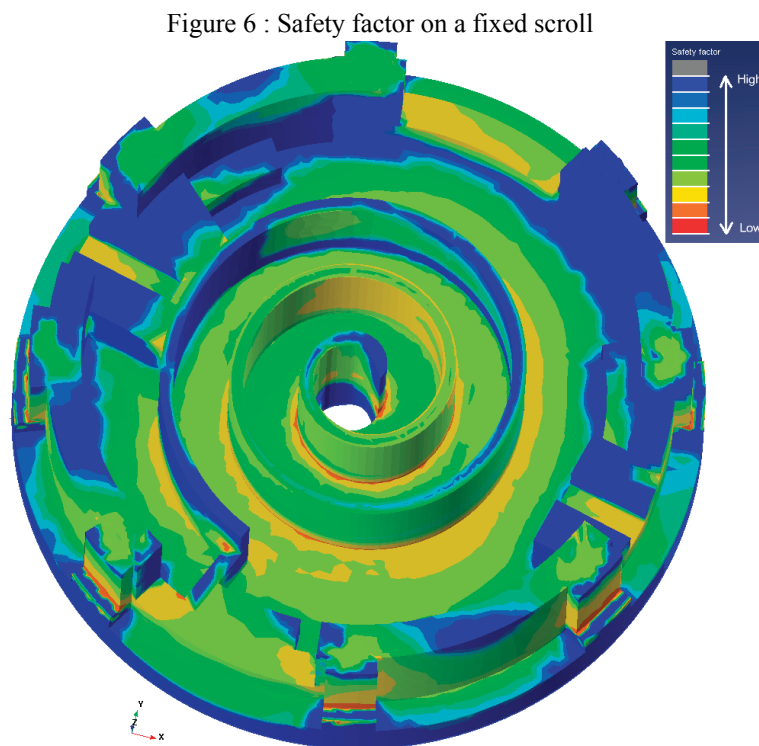
Another output of this analysis is the loading path at the node where the safety factor is minimum.

All FEA loadcases results are analyzed to define at this node the couple  $P(t)$ ,  $\tau(t)$  which define a loading path in the Dang Van diagram. Damage will occur only if the loading path crosses the Dang Van threshold line.

The more the loading path is far from the Dang Van threshold line, the more the reliability is high.

Then a notion of reliability can be deduced from this diagram.

Indeed the reliability can be quantified by the distance between the loading path (in red) and the Dang Van threshold line (in blue).



Graph 5 : Loading path of a fixed scroll on Dang Van graph

## 5.5 Reliability approach

Reliability is studied both with a predictive and experimental approach. As regards the resistance of materials, there are several conditions to match :

- validation of the design, which relies on computations and tests
- quality of the rough material, this is why the permanent control of the material characteristics is compulsory
- quality of the machining, this requires periodic dimensions checking
- field return analysis, a number of compressors are evaluated in partnership with the customer and the product is fully dismantled and analyzed after use.

We use a Pareto tool to sort the various causes of the failure and 8D tool to solve quality issues and thus improve the reliability of our compressors.

The predictive approach requires a large database of all types of products, of which behavior is perfectly known. From this database, it is easier to design a new product, knowing the limits of the material and of the design.

Danfoss Commercial Compressors has got a monitoring system based on the field return compressors. This system is based on the Marciano table which allows to predict the future product reliability from :

- the current field return situation
- the gain of reliability for the implemented improvement actions.

Another strategy which has been deployed is the "test to fail" procedure which allows to determine the extreme limit of each product. If this limit depends on the resistance of materials, then this will be recorded in the database, in order to improve the new products.

The field return database and the test to fail are important inputs, and combined with fatigue simulation open on a reliability analysis with a stress-strain curves method.

## 6. CONCLUSION

The fatigue analysis by FEA simulation is one of the elements to improve the reliability of compressors.

This job requires several steps :

- knowledge of the cyclic loads, which must be fully determined over one turn of crankshaft
- know-how in the FEA strategy and tools, with meshing refinements on the critical spots
- choice and use of a fatigue criterion in the context of compressors, with the associated database for previous calculations and tests

The more knowledge is built, the more efficient the analysis will be. This type of knowledge could be implemented into the software of an expert system in order to help to make decisions.

It enabled us to modify the geometrical shapes and to increase the safety factor, so the lifespan, in order to have the best reliability for all our compressors.

**To sum up, thanks to this fatigue analysis by FEA simulation, we acquired :**

- ✓ **A better knowledge about fatigue strain of our wraps**
- ✓ **A better technique to improve the safety factor and so the lifespan of our compressors wraps**
- ✓ **A better global reliability.**