AIRCRAFT ENGINE CRANKSHAFT OPTIMISATION

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SHRNUTÍ

Tento článek popisuje část návrhového procesu klikové hřídele vznětového dvoudobého motoru s protiběžnými písty, při níž je pro zadané průměry a délky čepů hledán optimální tvar ramen hřídele. Je zde popsán postup vzniku několika variant, které jsou následně podrobeny kritickému zhodnocení, a dále mechanismus výběru výsledného nejlepšího možného návrhu.

Výsledný tvar klikové hřídele je v poslední fázi podroben výpočtu únavové životnosti, jakožto finálního kroku návrhového procesu.

KLÍČOVÁ SLOVA: KLIKOVÁ HŘÍDEL, TOPOLOGICKÁ OPTIMALIZACE, ÚNAVOVÁ ŽIVOTNOST

ABSTRACT

This article presents part of the crankshaft development of a two-stroke compression-ignition engine with contra-running pistons where, for invariably specified diameters and pin lengths, the optimal crankshaft shape is searched for. The process of creating several options which are then subjected to critical evaluation followed by the selection mechanism for the final best possible design is described.

As the final step of the design process the resulting crankshaft shape is examined by calculation of fatigue life.

KEYWORDS: CRANKSHAFT, TOPOLOGICAL OPTIMISATION, FATIGUE LIFE

1. INTRODUCTION

Internal combustion engines, in a relatively unchanged form and as we know them today, have been used by people for more than a century. A number of alternative arrangements have been produced and can be found in recent decades, which have had a greater or lesser degree of success in introducing changes to the mechanism for heat-to-mechanical energy transformation (e.g. a rotary piston engine). However, the typical structure with crankshaft and connection rods is still the most widely used.

One of the possible alternative cranktrain arrangements is offered by an engine with contra-running pistons in a common cylinder liner (Figure 1). It is a two-stroke compression-ignition engine and its application seems to be valuable particularly in an aerospace industry due to its compactness resulting in low mass and an ability to burn a variety of fuels.

An engine with contra-running pistons is frequently characterized by two crankshafts which are placed against each other and coupled via a gear system. In this particular case a two-cylinder engine is taken into consideration and therefore two connecting-rods are attached to each crankshaft. The unusual use of two crankshafts brings, theoretically, twice

the mass and moment of inertia, which has to be eliminated, to a large degree, during a thorough development process (in practice it is slightly less because this design offers a larger swept volume for the cylinder unit compared to a cranktrain with a fixed cylinder head). However, against this is the requirement for high stiffness due to the high combustion

FIGURE 1: Powertrain scheme

OBRÁZEK 1: Schéma hnacího ústrojí

FIGURE 2: Initial design
OBRÁZEK 2: Výchozí návrh

FIGURE 3: Ideological designs; cast (left) and forged
OBRÁZEK 3: Ideové varianty; odlitek (vlevo) a výkovek

pressures. The crankshaft is clearly one of the key components in this engine with a huge impact on its overall dynamics and fatigue life. Therefore, it is worth paying attention to it.

The whole process of crankshaft development comprises a series of steps. Nevertheless, not all of these are presented in this article because some basic dimensions are already given from the initial design. It is also assumed that the original crankshaft shape meets all the requirements before the optimisation, which means that the improved parameters will also comply with the requirements. Shape optimisation represents a part of structural optimisation which combines mechanics, variational calculus and mathematical programming to obtain a better design of structures. In this article the shape optimisation includes changes in size, design, and topological parameters. Size (dimensions) means the size of the individual parts of the structure and the construction as a whole. The design (i.e. shape in its essential meaning) refers to the boundary surfaces of the object's partial volumes (for example main pin, crankwebs, etc.). Topology then represents the number and placement of holes, the connection area, etc. In its most general settings, shape optimisation of continuum structures should consist of a determination for every point in space, whether there is material in that point or not. [3]

An optimal structure should provide optimal material distribution with respect to given loads and boundary conditions. The most appropriate design for selection is defined by the best effectiveness of the whole structure under external conditions, when all criteria are met.

Shape optimisation described in this article focuses mainly on the shape of crankwebs connecting individual pins of the crankshaft.

2. INITIAL DESIGN

The design of a 3-main-bearing crankshaft created at the Brno University of Technology [1] was used as the initial shape for optimisation (Figure 2). This initial design resulted from the requirements for manufacturability and fatigue life.

Another feature of this crankshaft is the absence of counterweights. The crankshaft location and dimensions of other cranktrain parts (mainly pistons and connecting-rods) do not allow for sufficient static moment of the counterweights. Previous analyses showed that even the biggest feasible counterweights are not very advantageous for balancing improvement and reduction of main bearings load. An unconventional material for counterweights (e.g. tungsten alloy) has not been taken into account due to its cost [1].

3. DESIGN PROCESS OF IDEOLOGICAL VARIANTS

To determine the limit values of the mass and stiffness two designs have been developed (Figure 3) with regard to general fundamentals of crankweb shaping. The main feature is the proposed production technique.

For the first variant (casting), the main aim is to achieve the lowest mass. The other variant (forging) is, conversely, designed for maximum stiffness (torsional and bending).

3.1 VARIANT "CASTING"

If the crankshaft material is considered homogeneous, the mass is directly proportional to the volume. In this case, the objective of mass minimizing is to slim down the crankwebs within the context of casting technology while sufficient stiffness has to be maintained.

The hollow pins are the specific feature of this variant, which is an effective method for better utilisation of material. Moreover, there are smaller stress differences. A truly significant lightening is achieved mainly in the crankpins; the main pin hollows are limited by the main and crank pin offset.

Crankwebs are considerably lightened by concave sidewalls in the transverse plane and by the drafts in the longitudinal crankshaft plane. Therefore, this design looks less sturdy compared to the original shape.

3.2 VARIANT “FORGING”

Unlike the cast variants intended to provide a solution with the lowest mass, the forged variant was designed with high rigidity in mind. Torsional stiffness is supported by a convex shape of crankwebs in the perpendicular plane; bending stiffness is supported by minimum chamfers in the longitudinal plane of the crankshaft.

4. TOPOLOGICAL OPTIMISATION

In order to obtain the objectively most advantageous crankweb shape with no influence of a designer, the topological optimisation tool incorporated in FEM solver ANSYS was used. This topological optimisation of a given body under required load can be performed by two methods. The first one consists of material removing of low-loaded volumes about the nominal and predefined value of these volumes, while by the second method the value of removed material is defined relative to the whole volume before optimisation.

The topological optimisation ANSYS module allows optimising of the shape under several load conditions acting at the same time – which is advantageous especially in the case of multi-axial crankshaft load.

Actually, this optimisation is performed as an iterative process in order to observe the material removing in sub-steps for better understanding of design trends. Another reason for using these sub-steps is the need to avoid the reduction in volume important for other reasons, rather than just for strength (e.g. supporting surfaces for the axial bearing, etc.).

For topological optimisation a substantially bigger initial volume, bigger than the expected result, is recommended. In spite of the volume of space given by a crankcase and other cranktrain parts, the initial crankweb shape (Figure 4) is designed as a relatively not very large solid owing to a number of sub-step reductions.

During the iterative process each sub-step of topological optimisation, in the form of finite element body, is captured and exported into a CAD modeller where a new shape of crankweb is designed and further used for discretisation and the next sub-step performance in the optimisation process (Figure 5). This process was terminated when the removal of additional material would cause a breach of the conditions defined by the external load.

The final shape of the crankweb resulting from the topological optimisation (Figure 6) is only slightly modified and used in the new crankshaft design. In spite of the small corrections performed during the optimisation process, the final shape of the crankweb can be declared the intended result of the topological optimisation algorithm. The crankweb is composed of relatively complex surfaces which are particularly noticeable on the sides. Generally, this shape can be described as one not very usual,

FIGURE 4: First interaction step of topological optimisation
OBRÁZEK 4: První interakční krok topologické optimalizace

FIGURE 5: Topological optimisation process
OBRÁZEK 5: Proces topologické optimalizace

FIGURE 6: Final shape after topological optimisation
OBRÁZEK 6: Finální tvar po topologické optimalizaci

rather robust, and relatively hard to manufacture. Further doubts over this shape may arise when detailed comparison is made with crankshafts of modern internal-combustion engines.

As mentioned above, the shape corresponding to the topological optimisation cannot be considered the optimal in all regards and therefore cannot be recommended for prototype manufacturing. Nevertheless, some features of this shape are used in subsequent steps, namely the proportion of rounded drafts in the longitudinal crankshaft plane, and the size of the “bulge” on the web sides.

Here a question arises: what is the cause of the failure of this design process? First of all, in itself the definition of the word "optimisation" means to target a partial recovery in the design process, not as a "wildcard" mechanism for the entire development. Only an almost finished component should be optimised and slightly "tuned" because of the changed objectives of the design, newly acquired knowledge, experience and the need to meet the requirements of new machine tools, etc. In general, the shape is as variable as the system of conditions allows. Simply shaped bodies are suitable for all automatic computational processes, but a crankshaft requires a rational view of the design process. This combination of methods is sometimes referred to as engineering optimisation.

5. DEVELOPMENT OF USABLE FINAL SHAPES

Designs that have been established so far can provide limits for individual variables (e.g. mass, torsional stiffness). The limits, within which the intended shape should appear, are different, but at the same time useful when it comes to setting the values of fatigue, strength, etc. Moreover, they can help us to detect the impact of individual shape elements on the final features of the crankshaft, and also as a basis for selecting the manufacturing technology. The following steps are based on the results arising from topological optimisation ("TOPO"), as the base shape (default) for further treatment. Subsequently, various modern shaped elements were selected and their application should lead to the achievement of contemporary modern crankshafts, while their size, position and final form will be derived based on the topological optimisation.

FIGURE 7: Detail of topological optimisation on the specific shape feature
OBRÁZEK 7: Detail topologické optimalizace na význačném tvarovém prvku

FIGURE 8: Final design
OBRÁZEK 8: Finální návrh

These should bring a weight reduction for the TOPO design. Hollow crankpins (for the cast design) and ribs on crankwebs (for forging design) have been selected (Figure 8). Dimensions and location of these elements were determined using topological optimisation (Figure 7), which required very small changes between steps to capture the appropriate ratio between the individual engineer's and software's adjustments.

5.1 COMPARISON IN TERMS OF MASS AND STIFFNESS

As mentioned in the text above, five different crankshaft concepts are suggested: two ideological, one arising from pure topological optimising, and two designs combining both approaches (designer's contribution and topological optimisation, i.e. engineering optimisation). All of these concepts are subsequently analysed with respect to three main parameters: mass, torsional stiffness and bending stiffness. The selected materials are 42CrMo4 for the forged version and GJS - 800 for casting.

The boundary conditions used for the stiffness calculation are not intended to accurately simulate operational loads, but to reflect the purpose of these calculations, i.e. just for comparing the variants. For torsional stiffness investigation, zero degrees of freedom were set at the flange of the crankshaft rear end while torque was applied along the first main pin axis. For bending stiffness, the crankshaft was cut in the middle of two adjacent main pins, thus one crank throw was actually bended. The main pins were attached to springs connected to a single point in the centre of the pins. Load was applied by forces acting on the crank pin.

These analyses (Table 1) reveal no surprising results, but highlighted the potential of the optimisation process to improve the properties of an initial design. Another important finding is clear from the performed comparison: various shape elements have a different effect on the torsional and bending stiffness, i.e. the ratio between the mass and the torsional and bending stiffness should arise from the requirements on the crankshaft, and thus be part of inputs for the optimisation process, rather than its result. However, such requirement considerably complicates the system input parameters for a fully automated optimisation process, and it is one of the reasons for creating a higher number of sub-steps.

TABLE 1: Torsional and bending stiffness comparison

TABULKA 1: Porovnání torzní a ohybové tuhosti

DESIGN	MASS [kg]	TORSIONAL STIFFNESS $\times 10^3$ [N.m.rad ⁻¹]	RATIO T. STIFFNESS/MASS $\times 10^{-3}$		BENDING STIFFNESS $\times 10^6$ [N.mm ⁻¹]
			[N.m.kg ⁻¹ .rad ⁻¹]	[%]	
INITIAL	8,6	130,2	15,1	100	19,9
IDEOLOGICAL CAST	7,7	141,7	16,1	106,6	16,2
IDEOLOGICAL FORGING	11,6	145,1	17,5	115,9	22,8
TOPO	9,7	147,4	15,2	100,7	20,9
FINAL CAST	8,8	141,7	16,1	106,6	18,9
FINAL FORGING	8,3	145,1	17,5	115,9	22,4

6. FATIGUE LIFE ANALYSES

A crankshaft is a component exposed to loads varying locally and over time having a large number of cycles. Therefore, a fatigue life analysis cannot be omitted during its development. Typically, places with the lowest (minimum) safety factor are identified and then evaluated for the entire range of operating speeds.

For the purposes of this study, a comparison of crankshaft concepts, a relatively conservative approach for load definition is applied. This is because the highest indicated pressure cycle in conjunction with the highest engine speed (4150 rpm) are considered, even though the highest indicated pressure is not reached at the highest engine speed. Previous analyses have proved that the original crankshaft is capable of operation and therefore meets all terms of fatigue life [1]. If an innovative concept achieves a higher level of safety factor than the initial one in further analyses, this concept can be considered a satisfactory one – also for the whole engine operating range. For two designs (forging and initial for comparison) safety factor k_p was computed by two methods: analytic-numerical method LSA and using FEMFAT max software. Boundary conditions were chosen with respect to the assignment, and only to the approximate accuracy of the design calculations: zero degrees of freedom were set at the flange of the crankshaft while the maximum torque was applied along the first main pin axis and a radial force from maximum in cylinder gas pressure was applied on one of the crank pins. Crankshaft bearings were simulated by a set of springs with stiffness approximately corresponding to the stiffness of a hydrodynamic bearing for a given speed.

6.1 LSA METHOD

This method (LSA = Local Stress Analysis) evaluates two loading conditions that give rise to the maximum and minimum deformation. Stress properties are detected by FEM for these conditions (load states), focused in particular on changes in the layers just below the surface (because of stress gradient impact). The differences between results indicate sensitivity to fatigue damage. The two load states are also modelled by a more conservative approach that makes it possible to expect lower safety factors than would be appropriate to the real load. The first load state of the crankshaft is defined as simultaneous action of maximum radial force and maximum torque. The second one is defined only by minimum torque. This method uses the approach based on the relative stress gradient computed by FEM solver and defined by:

$$\chi_R = \frac{1}{\sigma_{\max}} \left(\frac{d\sigma}{dx} \right),$$

where χ_R is relative stress gradient, $\sigma_{\max}$ is maximum stress and x is a coordinate.

Using the FEM calculation method only local stress, not nominal, is obtained. And therefore, using ratio β/α is necessary [2]:

$$\frac{\beta}{\alpha} = 1 + \sqrt{\chi_R} \cdot 10^{\left(0.35 + \frac{R_e}{810}\right)},$$

where α is stress concentration factor, β is fatigue notch factor and R_e is yield strength.

Distinctive characteristics of stress are determined by well-known formulas:

$$\sigma_{e\max} = \text{sign}(\sigma_{1a}) \cdot \sigma_{VMa},$$

$$\sigma_{e\min} = \text{sign}(\sigma_{1b}) \cdot \sigma_{VMb},$$

$$\sigma_{ea} = \frac{\sigma_{e\max} - \sigma_{e\min}}{2},$$

$$\sigma_{em} = \frac{\sigma_{e\max} + \sigma_{e\min}}{2},$$

where $\sigma_{e\max}$ and $\sigma_{e\min}$ are maximum and minimum stress, σ_{1a} and σ_{1b} are the first principal stress for the first and second load set, σ_{VMa} and σ_{VMb} are the von Mises stress for first and second load set, σ_{ea} is stress amplitude and σ_{em} is the mean stress.

The resulting safety factor is given by [4]:

$$\frac{1}{SF} = \frac{\beta}{\alpha} \frac{\sigma_{ea}}{\sigma_c \cdot \eta_\sigma \cdot \nu_\sigma \cdot f_G} + \frac{\sigma_{em}}{R_m},$$

where SF is safety factor, η_σ is survival probability, f_G is correction factor, σ_c is fatigue limit stress, R_m is ultimate strength and ν_σ is size factor.

6.2 FEMFAT MAX

Using specialised software for fatigue safety factor evaluation more accurate results can be achieved as this computation removes the simplifying assumptions associated with the LSA method. However, the accuracy of the results depends on the quality of the input data as in the previous case. In order to compare the outputs of these two methods the load cases are applied identically. Another advantage connected with employment of specialised software is a comprehensive solution for the whole component which provides the possibility of

DESIGN	METHOD	FATIGUE SAFETY FACTOR
INITIAL	LSA	1,038
	FEMFAT	1,063
FINAL FORGING	LSA	1,104
	FEMFAT	1,107

TABLE 2: Fatigue safety factor

TABULKA 2: Součinitel bezpečnosti vůči mezi trvalé pevnosti

rendering the results based on its volume, not only in one point as in the previous case.

As Table 2 shows, fatigue safety factors computed by FEMFAT are almost identical to the previous computational method. Unfortunately, this may not be the rule because, especially for LSA, the result is significantly influenced by the choice of empirical variables.

7. CONCLUSION

In general, topological optimisation discussed in this article cannot bring any substantial changes in cranktrain dynamics behaviour and it is only a part of the whole engine design process. On the other hand, modern combustion engines have been improved to nearly the utmost possible forms and finding ways to improve their parameters even further, or save on overall costs, whether manufacturing or other processes, requires a great effort.

Suitably performed optimisation of machine parts may be beneficial in terms of better utilisation of material and production cost savings. However, it is very important that the optimisation has been conducted from the beginning in the particular direction leading towards the predetermined objectives.

Whereas these changes in overall behaviour of the engine due to topological optimisation are not significant, testing is performed especially on a parameter of the fatigue safety factor by a simplified version of the calculation method which should definitely be taken into account because it reveals the strengths and weaknesses of individual designs during the design process. In the specific case of crankshaft optimisation presented in this article, the newly developed shape concept shows significantly improved torsional and bending stiffness without accompanying mass increase and therefore the set goals can be considered achieved. The next appropriate step would be to compare all designs using a complex computational model of the cranktrain dynamics, i.e. a virtual engine.

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LIST OF ABBREVIATIONS AND SYMBOLS

FEM	Finite Elements Method
LSA	Local Stress Analysis
χ_R	Relative stress gradient
$\sigma_{\max}$	Maximal stress (in general) [MPa]
α	Stress concentration factor [1]
β	Fatigue notch factor [1]
R_e	Yield strength [MPa]
R_m	Ultimate strength [MPa]
σ_{1a}	First principal stress, first load set [MPa]
σ_{1b}	First principal stress, second load set [MPa]
σ_{VMa}	Von Mises stress, first load set [MPa]
σ_{VMb}	Von Mises stress, second load set [MPa]
$\sigma_{\max}$	Maximal stress [MPa]
$\sigma_{\min}$	Minimal stress [MPa]
σ_{em}	Middle stress [MPa]
σ_{ea}	Stress amplitude [MPa]
σ_c	Fatigue limit stress [MPa]
η_σ	Probability of survival [1]
f_G	Correction factor [1]
ν_σ	Size factor [1]
S_F	Safety factor [1]

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