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DESIGN AND FATIGUE ANALYSIS OF VALVE SPRING USED IN TWO WHEELER

[1]Bharathrinath Gorakhnath Kadam, [2]P. H. Jain., [3]Swapnil S. Kulkarni
[1]ME-Mech/Design Engg-2ndyr, College of Engineering, Osmanabad, M.S., India
[2]Asst. Prof. College of Engineering, Osmanabad, M.S., India
[3]Director, Able Technologies India Pvt. Ltd., Pune, M.S., India

ABSTRACT

Springs are mainly used in the industry as members absorbing shock energy as well as for restoring the initial position of a part upon displacement for initiating a given function. Compression springs are coil springs that resist a compressive force applied axially. In every two-wheeler valve springs play an important role in controlling the breathing in internal combustion engines. The valves are mechanically opened by a camshaft, via valve lifters or tappets, and closed by the valve springs. The analysis would indicate the prominence of the design parameter and its effect over the performance of the spring in terms of fatigue life. The work would be pursued to conclude with a proposal specifying the suitable values (levels) for the significant parameters identified for this study. References are drawn from the literature review as well as the industry.So it is important to calculate the fatigue life and reduce fatigue failure in its intended working period (i.e. before 3, 00,000 cycles). A typicalhelical compression spring configuration of two wheeler horn is chosen for study.

KEYWORDS: *Fatigue Analysis, Valve Compression Spring, Geometric Modeling, Life Analysis, Two Wheeler Valve.*

I. INTRODUCTION

A spring is an elastic object used to store mechanical energy. Springs are elastic bodies (Generally metal) that can be twisted, pulled, or stretched by some force. They can return to their original shape when the force is released. In other words it is also termed as a resilient member. A spring is a flexible element used to exert a force or a torque and, at the same time, to store energy. The force can be a linear push or pull, or it can be radial, acting similarly to a rubber band around roll of drawings. . Depending on the application, a spring may be in a static, cyclic or dynamic operating mode. Typically, the valve spring is subjected to cyclic loading with a maximum expected frequency of usage at about thousands of cycles per a day. Considering the design life of a two-wheeler at about 15 years, the spring should withstand a cyclic compression loading for about a million times. It has been reported by the warranty/ maintenance department that frequent complaints are being received over the failures of these springs well within their intended life span. The springs must be designed for reliability. The springs must be designed to withstand the cyclic loading during operation.
Fig.1 show the schematic representation of helical spring acted upon by a compressive load F.

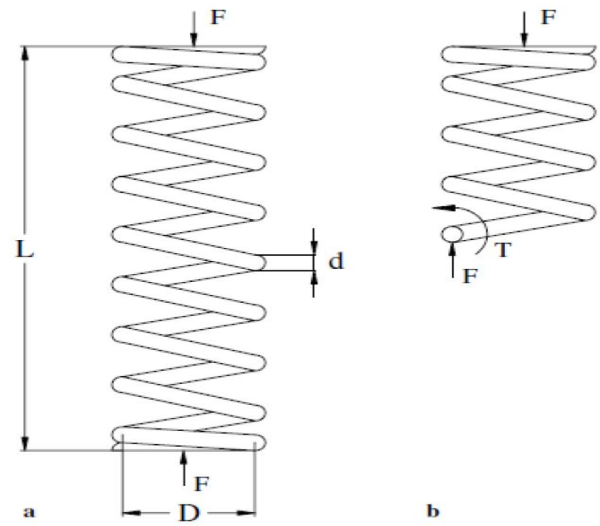

Fig.1 Helical spring with axial load F.(a and b)

Now imagine the spring is sectioned at some point, fig.1(a). Internal forces are generated to maintain the remaining portion shown in Fig. 1(b) in equilibrium. A direct shear force F and a torque T appear. The maximum shear stress on the wire can be calculated by the following equation .

$$\tau_{max} = \pm Tr_J + FA \ (1)$$

Where J is the polar moment of inertia, A is the cross section area and r is the wire radius. The first term corresponds to torsion contribution and the second term is the direct shear stress. Considering a wire with circular cross section of diameter $d(= 2r)$, the last equation may be reduced to

$$\tau = 8FD\pi d^3 + 4F\pi d^2 = \mathrm{K}\ 8FD\pi d^3$$

Where D is the mean coil diameter, K is the Wahl factor, C is the spring index defined by

$$\mathrm{K} = 1 + 0.5C,\ \mathrm{C} = Dd$$

When the external load, F, is variable, the induced stress is variable as well. The mean stress τm and the amplitude τa are defined by

$$\tau_m = K_s\ 8F_m D\pi d^3 \text{ and } \tau_a = K_b\ 8FaD\pi d^3$$

Where Ks and Kb are correction factors due to curvature andFm and Fa are the mean load and load amplitude, respectively.

II. LITRATURE REVIEW

Dakshraj Kothari, Rajendra Prasad Sahu and Rajesh Satankar[1] in this paper describe static and fatigue analysis of conventional leaf springs made of SUP 9 & EN-45 Material. These springs are comparing for maximum stress, deflection and stiffness as well as fatigue life. The CAD models are prepared in CATIA and analyzed by using ANSYS 12.1. Computer algorithm using C++ language has been used in calculating maximum stress, deflection and stiffness. Calculated results are comparing with FEA result. SUP 9 springs has lower value of maximum stress, deflection and stiffness incompare to EN45 spring. Predicted fatigue life of SUP 9 spring is higher than EN45 spring. Although, marketprice is much lower than Sup 9 spring.

Gajanan S.Rao and R.R. Deshmukh[2] This paper describe fatigue analysis of compression spring used in two wheeler horn. In most malfunction problems of two wheeler horns is due tofatigue failure of spring in warranty period so it is important to calculate the fatigue life andreduce fatigue failure in its intended working period. Lifeanalysis determines fatigue life of spring by the safe stress and corresponding pay load of thehelical compression spring. This work describes fatigue analysis of the helical compressionspring is performed using NASTRAN solver and compared with analytical results. The preprocessing of the spring model is done by using HYPERMESH software. The present workattempts to analyze the life using MSC fatigue software and also verified by experimentation.

S.S. Gaikwad and P.S. Kachare [3] in this paper describe static analysis of helical compression spring used in two wheeler horn. Static analysis determines the safe stress and corresponding to pay load of the helical compression spring. The static analysis is performed using NASTRAN solver and compared with analytical results. The preprocessing of the spring model is done by using HYPERMESH software.

William H. Skewis [4] discussed spring reliability factors, as springs tend to be highlystressed because they are designed to fit into small spaces with the least possible weight andlowest material cost and required to deliver the required force over a long period of time. Thereliability of a spring is related to its material strength, design characteristics, and theoperating environment. Corrosion protection of the spring steel has a significant impact onreliability and so material properties, the processes used in the manufacturing of the spring, operating temperature and corrosive media must all be known before any estimate of springreliability can be made. Spring reliability is also directly related to the surface quality and thedistribution, type and size of sub-surface impurities in the spring material.

V.B. Bhandari [5] book of "Design of Mechanical Elements" include, spring chapter. In this chapter we will discuss the more frequently used types of springs, their necessary parametricrelationships and their design.

III. DESIGN OF VALVE SPRING

1. Mode of loading: Cyclic loading
2. Outer diameter of coil, D_o= 26 mm
3. Inner diameter of coil, D_i= 19 mm
4. Wire diameter, d_i= 3 mm
5. Pitch, P= 8 mm
6. Mean diameter of coil D = 24 mm
7. Number of active coil, N=4.25Nos
8. Total Number of coil N_t=5.25Nos.
9. Free Length, Lo=P*N_t=42 mm
10. Solid Length Ls= d*N_t= 15.75 mm
11. Max.ForceF_{max}=100 N
12. Min.ForceF_{min}= 0N
13. Spring Index C =D/d =8 mm
14. Necessity of guide: Compression spring may buckle at low axial force for this reason spring needs guide its necessity is checked by,

$$\frac{Free\ length}{Mean\ coil\ diameter} \leq 2.6$$ ……….(Guide is not required)

$$\frac{Free\ length}{Mean\ coil\ diameter} \geq 2.6$$ ……….(Guide is required)

For Valve compression spring, $\frac{10.2}{4.35}$ = 2.34 <2.6 . Hence, guide is not required.

(i.e. no need to consider effect of buckling)

15.Wahl's stress factor Ks = $\frac{4(8)-1}{4(8)-4} + \frac{0.615}{8}$ = 1.188

16.Shear stress : $\tau = \frac{8FD}{\pi d^3} Ks$

$\frac{8*100*24}{3.14*3^3}$=$\times$1.18 = 261.54 N/mm^2

17.Axial Deflection : $y = \frac{8FD^3 \times i}{Gd^4}$ = $\frac{8*100*24^3}{4*3^4} * 6$=5.27 mm

18.Mid Range shear stress $\propto_m = \frac{Ks*8*5*24}{\pi*3^3}$=13.07 Mpa.

19.Spring material EN47 (Chrome Vanadium wire) values of

A=2005 MPa. &Exponent (Constant) m = 0.168

20. Sut = A/dm
    = 2005/20.168
    Sut = 1784.6 Mpa
21. Torsional Yield Strength, Ssy = 0.45Sut = 803.07 Mpa
    Modulus of Elasticity, G = 77200 Mpa
    Spring Rate,
    k = G*d4/ (8D3N)
    = 77200*24/ (8*243*4.25)
    k = 2.63 N/mm
    Deflection, y = F/k = 9.5mm
22. Spring Safety factor, ns = Ssy/τa = 3.78
    Fractional Overrun, ξ= 0.15
23. Spring Force,
    Fs = (1+ξ) Fmax = 57.5N
    Weight of spring,
    W = π2d2DNγ/4
    = π222*24*4.25*82*10-6/4
    W = 0.08N
24. Fundamental Frequency,
    f = (1/2)*√ (g*k/W)
    = (1/2)*√ (9.81*2.35/0.08)
    f = 264.17 Hz
25. Amplitude Endurance Strength, Ssa = 241 Mpa
26. Midrange Endurance Strength, Ssm = 379 Mpa
27. Factor of Safety, n
    1/n = (τa/Sse) + (τm/Sut)
    1/n = (212.21/352.84) + (212.21/1784.6)
    n = 1.39
    Constant,
    a = (f* Sut) 2 / Sse
    = (0.78*1784.6)2/352.84
    a = 5491.53
    Constant,
    b = - [log (f* Sut)/ Sse]/3
    = - [log (0.78* 1784.6)/ 352.84]/3
    b = -0.199
28. Finite life,
    N = ((Sf/n)/a) (1/b)
    = ((242.73/1.39)/5491.53) (-1/0.199)

N = 3.35 x 107 Cycles

IV. MODELLING AND ANALYSIS OF VALVE SPRING

Reason to use the FEA in valve spring design is it reduces error caused by the simplification of equations. An FEA based design begins with the selection of the element type, how the model should be constructed, how accurate the results should be, and how fast the model should be run. The most accurate FEA results can be obtained by creating 3D modelof a valve spring and its followed by meshing the parts with a 3D solid element. Finer meshing with higher order elements in general will produce the most accurate results.

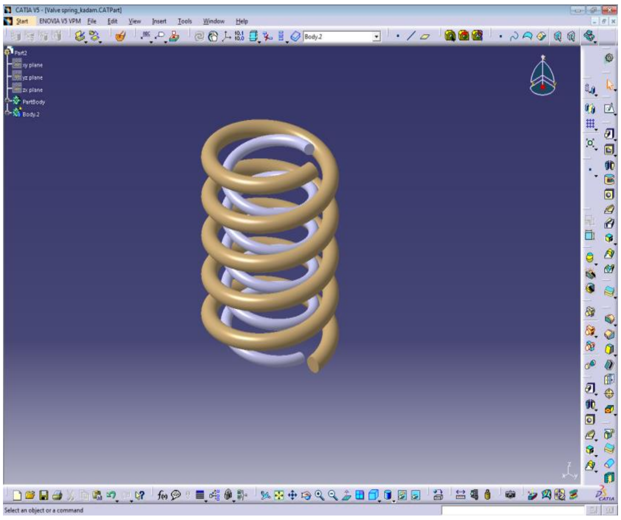

Fig.2 CATIA Model of valve spring.

3D geometry is created using CATIAV5.Gometric modelling CATIA is used for computer-aided design.3D geometry is created using CATIAV5.Gometric modelling CATIA is used for computer-aided design. Geometric modelling is concern with the computer compatible mathematical description of the geometry of an object.

Analysis is carried out by using HYPERMESH as pre-processor and NASTRAN as solver and HYPERVIEW as post-processor.

Fatigue study was performed by using MSC-FATIGUE software as well as ANSYS. Then import the geometry from CATIA software to HYPERMESH.

V. MESHING

Meshing involves division of the entire model into small number of pieces called element. It is convenient to select the hex mesh because of high accuracy in result. To mesh the valve spring the element type must be decided first. Here element type is solid.Fig.3 shows the meshed model of coaxial valve spring.

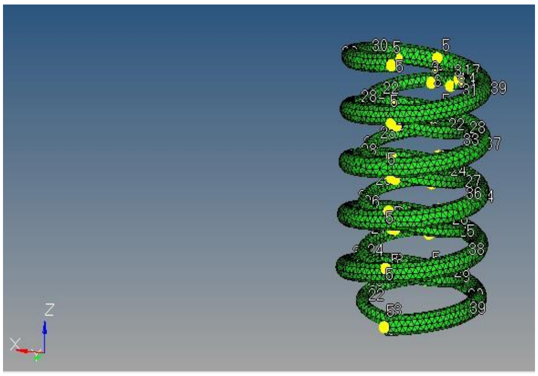

Fig.3 meshing

VI. NASTRAN

Static analysis is carried out by using MSC-NASTRAN software. And all the output result from this software are read in HYPERWORK software. Nastran file contains the result of the analysis such as displacement, max. shear stress and von-misses stresses.

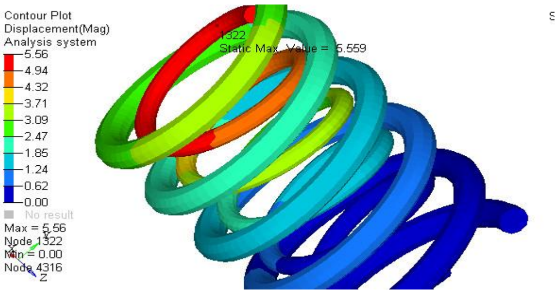

Fig.4 displacement

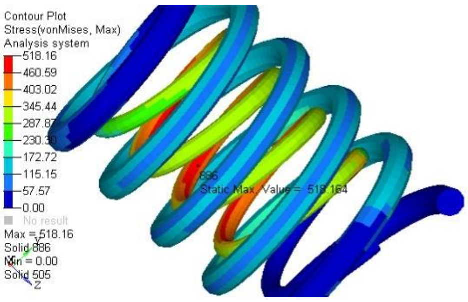

Fig.5 max.shear stress

VII. Fatigue Life

After NASTRAN solver the result file and the input deck file is imported in the MSC FATIGUE software for fatigue prediction of a given valve spring.

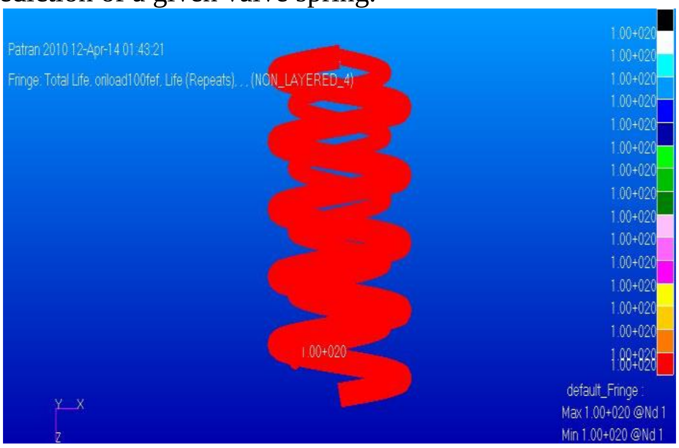

Fig.6total life

Fatigue study is performed to find life of valve spring in terms of number of cycles. This fatigue study was performed by using msc-fatigue software. The total life of valve spring is $1*10^{20}$.
For life analysis and life enhancement of valve spring propose new design along with other influencing factor like changing the pitch of spring. Here existing pitch of the spring is 8 mm and modified pitch of the valve spring is 6 mm.
Similarly static analysis of modified coaxial spring is carried out by Nastan and then importing to the same Msc. Fatigue which shows the life of modified pitch spring as shown in figure.

VIII. Modified Valve Spring

According to existing work modified spring is designed by changing the pitch of the spring.
Existing pitch of the spring= 8 mm and modified pitch of the spring= 6 mm.

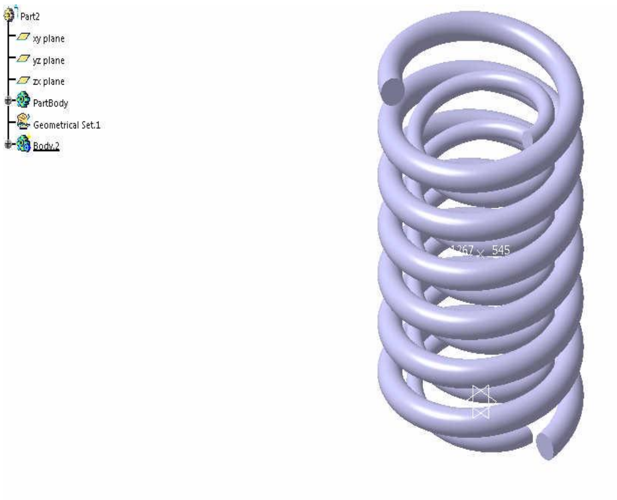

Fig.7 modified pitch coaxial spring

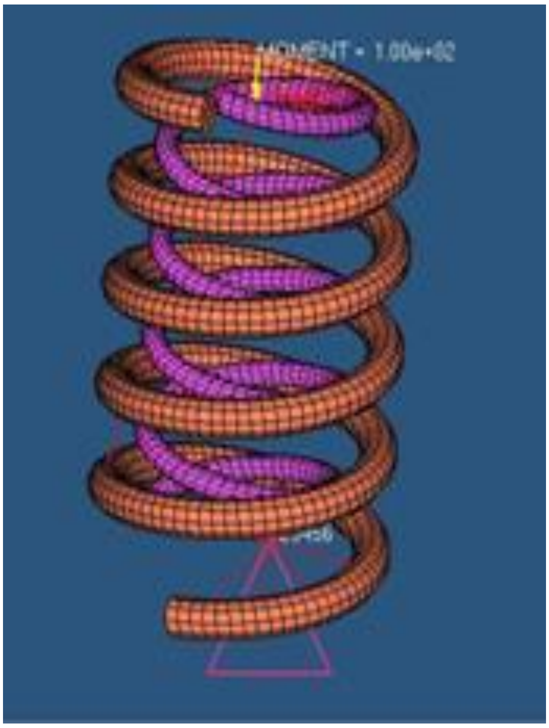

Fig. 8 meshing of modified spring

The following are the material properties of valve spring
Material EN-47, Young's modulus = $2*10^5 N/mm^2$,Density = 7850, Poisson ratio = 0.3

IX. BOUNDARY CONDITION

The valves are mechanically opened by a camshaft, via valve lifters or tappet, and closed by the valve spring. With the dynamic behaviour of valve spring it must be considered that we have a lower resonant frequency than other components, meaning that they can be easily stimulated to undergo resonant vibration. One end of the spring is fixed and other end is connected to load.

X. LOAD APPLIED

The load is distributed equally by all the nodes associated with the center of the spring. To apply the load, it is necessary to select the circumference of the spring centre and consequently the nodes associated with it. Here we apply 100KN load on spring and applied load need to divide with the number of nodes associated with the circumference of the spring centre. The load is applied along the FY direction. Boundary condition and load applied as shown in fig.8

After applying load and boundary condition import HM file to the Nastran solver and run the analysis and thereafter final results are shown below.

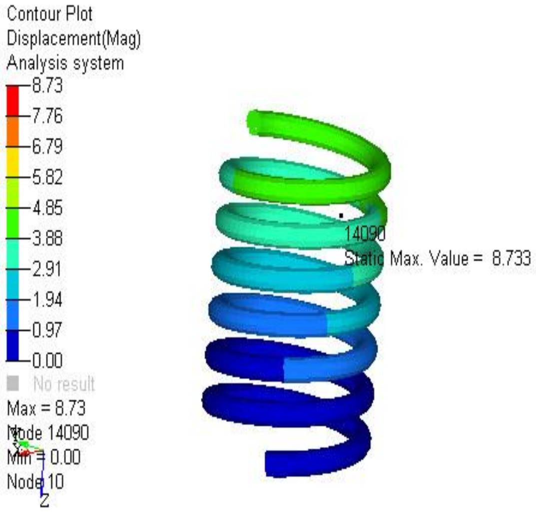

Fig.9. displacement

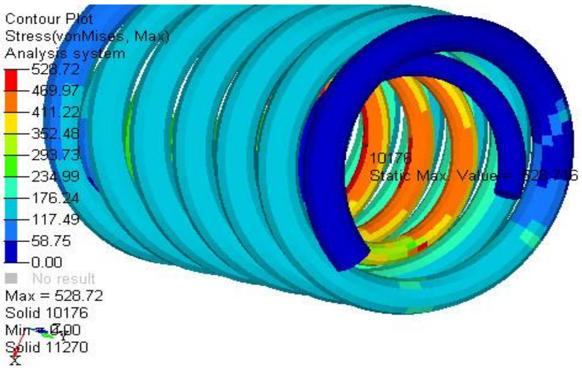

Fig.10. max.shear stress

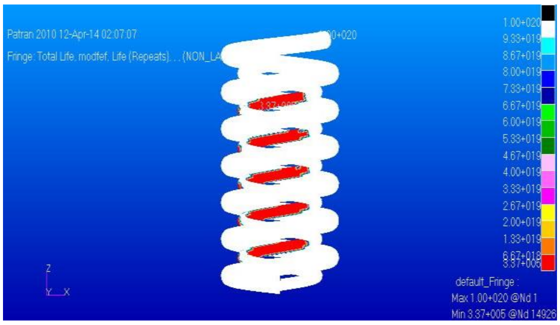

Fig.11.max. life of modified pitch spring.

Using MSC-FATIGUE life is analyzed which can be 3.37 * 1o$^{5.}$ The result obtained is as shown in fig.11. It's life is found to be more than 300000 cycles.

XI. Comparative Result In Between Original And Modified Pitch Spring

	Outer Spring				Inner Spring				Displacement (mm)	Stress (MPa)	Mini. fatigue life
	Total Height	Pitch	Wire Diameter	Tapered angle	Total Height	Pitch	Wire Diameter	Tapered angle			
Original	40	8	3	0	36	7.5	2	0	5.56	716.98	$1*10^{20}$
Pitch Variation	40	6	3	0	36	6	2	0	8.73	528.72	$3.37*10^{5}$

XII. Experimentation

The experimentation for Coaxial spring is carried out using a Fatigue Life Testing SPM. The spring is held in position with other operating conditions identical to the application.Typically, the loading cycles are repeated for a million or more for ensuring that the spring stays intact during its operation.The number of trials is conducted in a very controlled environment with focus on the variables influencing the fatigue life. The trial runs are conducted to ensure consistency/ repeatability of the spring behaviour. The virtual validation of the spring (simulation using software) should address the given problem during the design selection stage. While the spring is made available in the physical form, the trials and testing would address the phase of validation.

Fig.12.life testing spm

Since the cycle time is 1 sec with a maximum dwell of 1 sec, the total testing time using the physical setup should not take more than three weeks after complete test setup is ready.

XIII. Conclusion

The above analysis presents that the valve compression springs becomes quite necessary to do the complete stress analysis of the spring. These springs undergo the fluctuating loading over the service life. In addition, FEM software has been use for performing meshing simulation. Almost in all of the above cases, fatigue stress, shear stress calculation play more significant role in the design of valve compression springs. The Comparison of the theoretical obtained result by the shear stress equation to the Finite Element Analysis result of helical compression springs is the mode of our present work, by this analysis it will possible in future to provide help to designers for design of spring against fatigue condition.

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AUTHORS' BIOGRAPHY

Bharathrinath Gorakhnath Kadam M.E(Mech)Design Engg-II College of Engineering, Osmanabad.

P.H.Jain M.E(Design Engg) College of Engineering, Osmanabad.

Swapnil S.Kulkarni Director, Able Technologies India Pvt. Ltd., Pune. The Company offers Engineering Services and Manufacturing Solutions to Automotive OEM's and Tier I and Tier II Companies. He is a Graduate in Industrial Engineering with PG in Operations Management. With around 20 years of working experience in the domain of R&D, Product Design and Tool Engineering, he has executed projects in the Automotive, Medical and Lighting Industry. His area of interest is Research and Development in the Engineering Industry as well as the emerging sector of Renewable Energy.