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Reliability Optimization of Non-Automotive Engine Valve Using Finite Element Analysis and Design of Experiments

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Abstract: Engine valves, designed and manufactured to work at high temperatures and pressures under repetitive load cycles, have to be capable of operating for long periods – thousands of times a minute – and at extreme temperatures. Worn valves will cause issues with an internal combustion (IC) engine. Ideally, valves should be checked and changed in line with the recommendations in the engine's manual, but at times, valves wear or break much before their stipulated lifetime. This can be attributed to non-optimal application of design, materials, manufacturing method, heat treatment, assembly style, or, quite simply, abuse of the final product by the end user. Engine valves experience a wide range of stresses during the operation of the engine, which lead to impact failure or fatigue failure. The valve thereby experiences macroscopic or microscopic defects, which render it imperfect or unusable. Most of the effects of a potential failure can be eliminated in the pre-manufacturing stages using proactive concurrent engineering, which incorporates (re)design and analysis of the valve using simulative conditions with any available past causal data, and design experiments to run trials between multiple factors across levels, to significantly reduce the occurrence of the same failure in the future. This paper seeks to systematically optimize the reliability of IC engine valves using Finite Element Analysis (FEA) and Design of Experiments (DoE).

Keywords — Design of Experiments, Engine Valve, Valvetrain, Finite Element Analysis, Fatigue Failure, Engine Valve Material, Reliability Optimization

I. INTRODUCTION

A. Engine Valve Fundamentals

Valves play an important role in an internal combustion (IC) engine's functioning. They are responsible for letting air into the main cylinder in the first stroke of the four stroke process and for allowing exhaust gases to escape on completion of the combustion cycle, which occurs in the fourth phase. A valve train is a mechanical system that controls operation of the valves in an internal combustion engine, whereby a sequence of components transmits motion throughout the assembly. Every four-stroke engine has at least one intake and one exhaust valve. A multivalve engine is an engine that has more than 1 intake and 1 exhaust valve. The reasons manufactures do this is because more valves equals lighter setup and more air flow. The more valves you have, the more air you can displace, resulting in higher possible RPM and HP. The intake valve allows the air/fuel mixture to enter the cylinder on intake stroke and the exhaust valve allows the used up exhaust gases to leave on exhaust stroke. They also must seal in compression and combustion against the valve seats on compression and power stroke. Valves must be made as light as possible to avoid valve float at high engine RPM. The intake valve is almost always larger than the exhaust valve, and this is because it is much harder to get air into the cylinder than it is to get the air out. Valves are usually of the poppet type (Fig.1), although many others have been

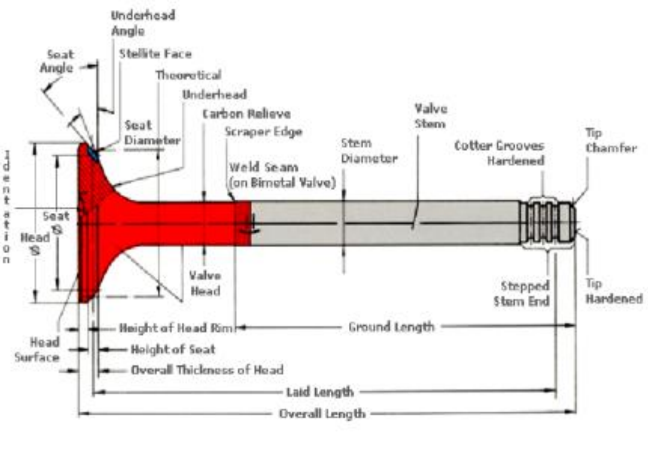

Fig.1: Valve terminology

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Fig.2 represents typical valves along with their mating parts - spring, oil seal, retainer and locks. These work in tandem to regulate the flow of charge into and out of the cylinder.

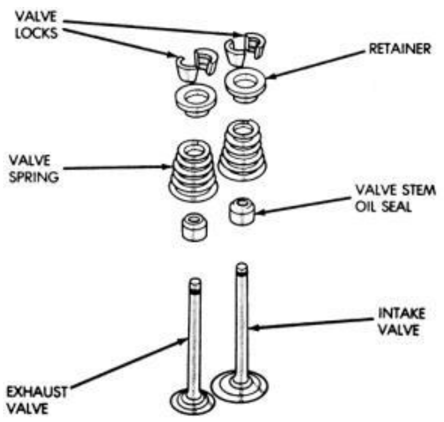

Fig.2: Valve and mating parts

B. Failure Types and Forces Acting on Valve

An engine valve can fail due to either or both of the two reasons - (i) impact, (ii) fatigue. The crack that results and the lines propagated on the broken surface indicate what kind of a failure the valve has experienced. It is best described using Fig.3.

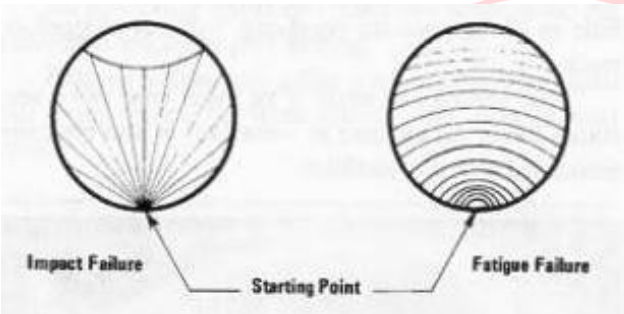

Fig.3: Types of valve failure

The forces experienced by the valve due to which it undergoes such aforementioned failure are best described using Fig.4.

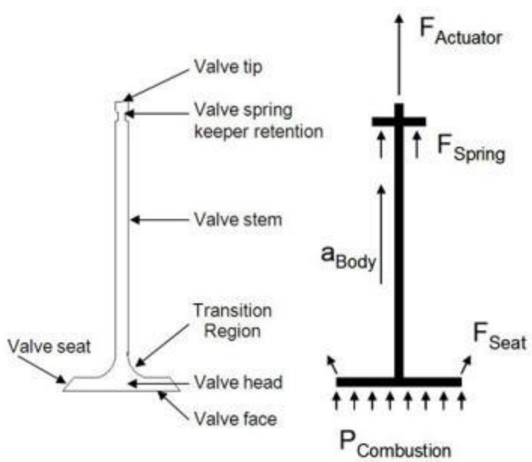

Fig.4: Forces experienced by the valve

C. Problem Statement

About 17 valve damage complaints based out of CI engines (diesel engines) from a non-automotive client have been raised, mostly from the same geographical territory. All complaints have the same valve damage issue – fracture at the valve profile-stem intersection neck region (between the valve underhead and the valve stem).

The outcome has been observed only with exhaust valves, mostly in the 4^{th} cylinder of the engine. There is no case of valve-piston striking. Almost all complaints have been raised after hardly 200-1140 operating hours (as against the mean manufacturing durability (or MTTF) of 10^8cycles, i.e., ~2800 operating hours at rated rpm of 1200).

The problem is the repeated occurrence of this damage issue despite the conformance to client design specifications, material microstructural norms and application of the right heat treatment method to the aforesaid region. There is qualitative and quantitative data available to consider, while further data can be derived/referred. Optimization of the valve's design based on the stresses experienced by it is needed to provide a long-term solution to the company.

D. Literature Review

A valve can fail due to various reasons. Defective valves are often overlooked as a crucial possibility for valve train failures, but it historically ranks as the number two cause, after thermal and mechanical overstress. A study that was conducted by a leading valve manufacturer found that as many as 1 out of every 5 (20.7%) valve failures resulted from defects in the valves themselves. Any valve will eventually wear out if driven enough kilometers, but many valves fail long before they should, because of burning or breakage. The consequent stresses acting on the valves make way for either impact failure or fatigue failure. ***(Lembhe N. S. et al., 2016)*** [14] observed that the design of the valve depends on many parameters like behaviour of material at high temperature, vibrations, fluid dynamics of exhaust gas, oxidization characteristics of valve material and exhaust gas, fatigue strength of valve material, configuration of the cylinder head, coolant flow and the shape of the port. On experimentation, it was found that valve radius fillet plays an important role in valve failure and should be carefully selected. Similarly, the results for selected valve radius as proposed by ***(Mamta R. Zade, 2016)*** [18] showed good improvement considering allowable stresses. ***(Karan Soni et al., 2015)*** [15] alluded that experimental investigations are often costly and time consuming, affecting manufacturing time as well as time-to-market, while developing valves for optimal properties. An alternative approach is to utilize computational methods such as Finite Element Analysis, which provides greater insights on stress and strain distribution across the valve geometry as well as possible deformation due to structural and thermal stresses. The results obtained through

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numerical analysis suggested that the valve design can be optimized to reduce its weight, without affecting permissible stress and deformation values. To back up the conclusions, ***(M. L. Pang et al., 2015)*** [38] showed that valve stresses can be reasonably analyzed using finite element methods, and, that the stress state of the valve under spring and combustion pressure load, temperature profile load and impact closing speed in normal operation still remains elastic, the energy transfer during the valve impact event is mainly conservative, and the valve stem maximum stress/strain is mainly linear with valve impact closing velocity. The obtained result from the work done by ***(S. K. Rajesh Kanna et al., 2015)*** [16] proved that the deformation due to loading and deformation due to thermal expansion of the valves will not seal the gap between the valve and the engine block. So the wear on the valve stem will be reduced. The maximum and minimum principal stress acting on the engine values were found to be within the allowable limits of the alloy and thus the valve life increased. As noted by ***(Naresh Kr. Raghuwanshi et al., 2012)*** [20] fatigue failure is the main cause of valve failure. The fatigue strength is significantly decreased with increase of temperature. It was also observed that the focus is on light weight engine valves which can operate without failure at high temperatures and can sustain stress cycles up to 10^9. Most of the companies in engine manufacturing found field related problems of valve and valve seat in gas engines also, and the seat angle along with the valve material play a significant role in handling the stresses developed, as concluded by ***(Rohit T. Londhe et al., 2014)*** [25].Improper selection of fillet radius leads to exhaust valve failure. Thus, fillet radius plays important role in exhaust valve failure and should be carefully selected, as per the observations of***(Snehal S. Gawale et al., 2016)*** [13].***(Y. S. Wang et al., 1996)*** [26] found that wear increases when number of cycles increases, increases as load increases, but decreases as temperature increases only after formation of protective oxide film, thereby reducing adhesive wear. A comparison of the microstructure of failed valves and new valves by ***(Ajay Pandey et al., 2014)*** [22] revealed that the size of grains, grain boundaries, and distribution of carbide particles is affected by high temperature operating conditions and has a serious impact on the useful life of the valves by not only adding to crack initiation and its propagation but by influencing the wear pattern also. It was suspected by ***(KeyoungJin Chunet al., 2006)*** [29] that mileage and speed will also have an adverse effect on the optimal functionality of valves, and was ultimately proven that the wear depth of the valve and seat insert increases sharply in accordance with a rise in the speed (Hz), whereas wear occurred rapidly in the initial stages of the test and wear depths increased less significantly with the cycle number (mileage), when it was increased from average to a factor of 3. Transient heat transfer studies carried out by ***(Ravindra Prasadet al., 1991)*** [31] showed that the temperature of the hottest valve points may reach 700°C at which the strength of the material of the valve will significantly decrease. The effect of cooling media temperatures is dominant at low loads while the hot gas temperature is dominant at high loads. The considerable temperature gradient existing in the valve body will lead to excessive thermal stresses which can cause valve break down. The results of the work done by ***(Lucjan Witek, 2016)*** [34] show that the dynamic stresses at the moment of valve closure can be about 5 times larger than the static value. It was also observed that just after cold start of the engine the bending stress in the valve with carbon deposit increases. The exhaust valve can freely rotate in the engine head, so the carbon deposit can be embedded in different part of the valve seat during the work of the engine. It can explain the initiation of more fatigue cracks in the valve stem. According to the result of thermal stress analysis done by ***(Haruki Kobayashiet al., 1997)*** [35], it is considered that one of the reasons for valve head fracture is the tensile stress caused by temperature distribution, and that the temperature difference on the valve under transition conditions is greater than that under constant conditions. The test sample alloy taken for failure analysis by ***(Oh Geon Kwonet al., 2004)*** [37] showed that the alloy over-aged significantly and the fatigue properties dropped due to the over aging and the over temperature. Multiple fatigue cracks initiated and propagated to failure. A crucial point to note was that there was no indication from the evidence available that the valve failed as a result of being incorrectly manufactured. Structural analysis for multiple materials produces excellent result by treating the problem as coupled field analysis, as proven by ***(G. Ragul et al., 2018)*** [39]. Impact wear studies of a common material done by ***(Takeo Ootani et al., 1995)*** [48]are also considered to gauge the criticalities of fatigue loads viz-a-viz impact loads. It was observed that, without sliding speed, the wear volumes of the specimens were very small (under 3 mm^3), and for impact with sliding, the wear rates of specimens tended to increase with increase in sliding speed. Furthermore, the valves impact the valve seat inserts at an oblique angle in an exhaust gas environment. Looking towards the mode and root causes of intake and exhaust valve failures, while designing the valve, important factors should be taken into account, according to ***(Yuvraj K. Lavhaleet al., 2014)*** [45]. These important factors are chemical composition of valve material, engineering dimensions and tolerances, operating temperatures, duty cycle, equipment applications, atmospheric condition, HP rating, pick torque rating, RPM rating. Besides of design parameter, operation and maintenance plays very important role for the failure of valves. For cause of fatigue failure, care should be taken likewise over speeding of the engine, foreign material entry during induction, hydraulic locks, etc. For thermal loading

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engine operating temperature is the key factor so intake & exhaust gas temperature should be controlled by tracking of overloading, fuel quality and cooling system performance. In case of wear failure valve adjustment and cleanliness of fresh air is most important. Dust entry in intake port is greatest enemy which causes early wear. Poor compression, scale formation, are the prime sources of valve failure due to corrosion-erosion. Performing Engineering Design Optimization (EDO) often requires knowledge about the stage of design, design variables and their minimum and maximum limits (independent variables), constraints, measurement of the design performance (dependent variables), design parameters and relationships between the independent and dependent variables (i.e., a design evaluation model). The design variables are dependent on the level of product definition available at the different stages of the design optimization. The design stages can vary from conceptual or preliminary design, and from configuration to detailed design. Design variables are expressed in either quantitative or qualitative terms. EDO has evolved with time from a totally manual process to computer-based approaches. The paper by ***(Rajkumar Royet al., 2008)*** [46] proposes a classification of the optimization problems based on two viewpoints: design evaluation effort and degrees of freedom. These viewpoints are relevant for mechanical engineering problems and show the major issues in optimization. Validation of reliability of the optimized design can be done by modes such as inter-rater reliability, test-retest reliability, internal consistency reliability and split-half reliability. ***(Necip Doganaksoy et al., 2013)*** [47] showed that, for reliability testing of parts or products meant for long life cycles in a short duration of time, the techniques employed are use-rate acceleration, degradation testing and accelerated life testing. The results obtained from one or more techniques so listed can be used to extrapolate the output for analysis conclusion purposes.

II. METHODOLOGY

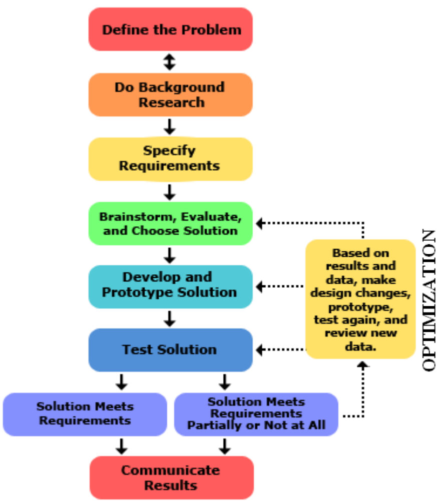

Fig.5: Product design methodology

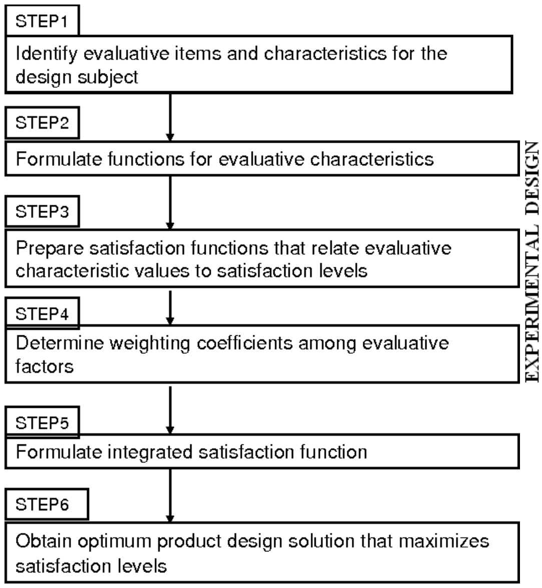

Fig.6: Product design optimization methodology

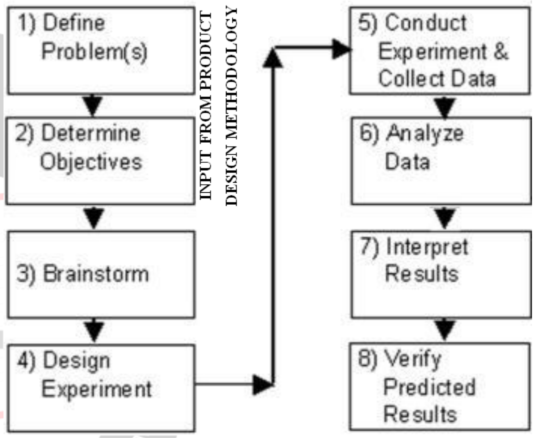

Fig.7: Experimental design methodology

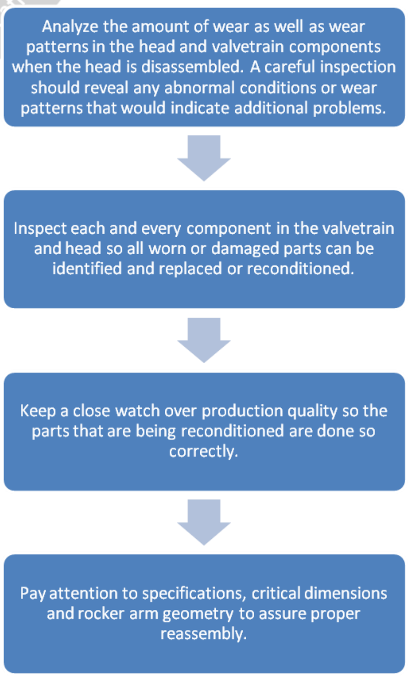

Fig.8: Sustainable valvetrain maintenance and repair

(post-project follow up)

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The entire methodology involving this project is divided into four separate stages shown by four separate images as above.

Fig.5 depicts the first stage of the overall methodology - the product design process.

It shows how the product (in this case, the engine valve) is brought to life from the concept level to the end product. Here, the loop linking the partial/non-optimal compliance of the valve design to the test solution, development and prototype solution and the initially brainstormed solution is called the optimization loop, and is a critical part of this project.

Fig.6 shows the optimization process loop as highlighted in Fig.5, discussed earlier.

It takes the most important parameters into consideration for the experimental design to be done on the same, for bringing about the most optimal design of the valve for the given operating conditions. The satisfaction functions that relate evaluative characteristic values and satisfaction levels work on a confidence level of 95% on a 3-factor 2-level DoE template.

Fig.7 highlights the experimental design methodology mentioned in Fig.6 as one of its components.

Here, the inputs, i.e., the problem definition and the determination of objectives are taken from the product design process, i.e., Fig.5, as the inputs to the product design itself stand as the critical factors of the experimental design involving that very product - the engine valve.

The data is collected and statistical experiments are conducted based on the following parameters of the engine valve that can be changed keeping the other engine components the same:

- Profile Radius
- Profile Angle
- Seat Angle

Fig.8 encompasses all steps that need to be taken to maintain and repair valvetrains after the implementation of the suggestive conclusions of this project. Although the points do not come under the direct objectives of the goals of this project, they are crucial for the longevity of the life of the engine valve.

While the design of the valve is being optimized and its reliability checked, the associated components in the valvetrain play a vital role due to mating and the subsequent wear and tear. Hence, post-project follow-up is inevitable to sustain the results of this study.

III. PHYSICAL ANALYSIS

Among the raised complaints, some valve samples were sent by the non-automotive customer to the company, of which a sample is shown in Fig.9. These valves were analyzed for their failures, along with the data such as hours run, application model, complainant location, etc. in full consideration.

Fig.9: Failed valve sample

It was observed that all the valves had failed at a very similar distance from the valve tip end, which lies in the neck region, and not at the merging point of the profile turning and stem grinding where the step mark is expected to be present. Hardness and Microstructural study did not show any abnormality which could have initiated the failure. However, the initial prognosis was that the repeated stem stress (as evidenced from the groove contact pattern) due to higher seating force might have been the cause for this failure.

Further, the samples and mating parts were observed in ‘as-received’ condition, as shown in Fig. 10:

Fig.10: Failed valve and mating parts sample

The company had received broken exhaust valves along with spring, cotter and retainer assembly for the root cause analysis. Some samples also underwent bending in the stem region which might have occurred due to the post-fracture damage. A more detailed observation also revealed the following details:

Fig.11: Most complaints were involving the 4th cylinder

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Fig.12: No stamping was observed on piston top face

Further, every single rocker tip was found to be okay, every single FIP (fuel injection pump) seal was intact, and in each case, except the failed valve, the remaining assembly was unaffected. This created a necessity for a deeper study using microscopic imaging.

Fig.13: Regular contact pattern was seen on the valve end

Fig.14: Abnormal contact pattern was seen on the first spring groove

Macro analysis of the sample revealed the following details:

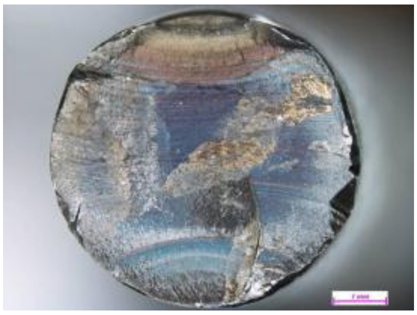

Fig.15: Fracture lines on breakage region

The fracture surface indicates that the fracture mode is fatigue in nature. Radiating lines from the point of 6 o' clock position confirm that point as origin and the final fracture zone is at 12 o' clock position. This was a low nominal mild-stress concentration based unidirectional bending fatigue failure, as shown in the classification in Fig.16.

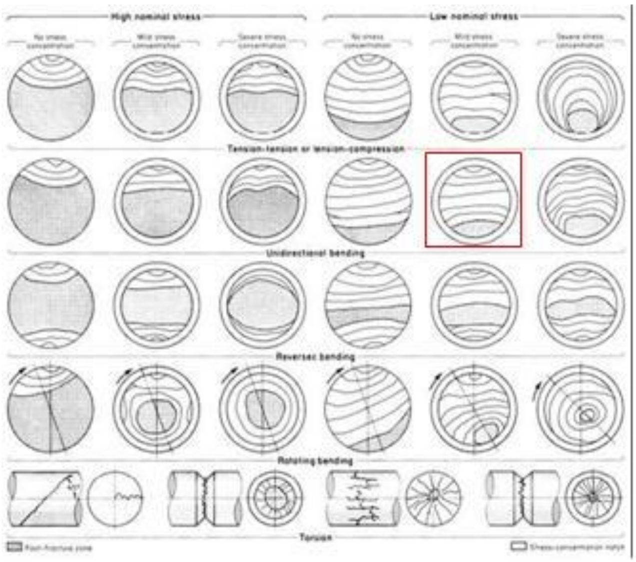

Fig.16: Types of fatigue failure of circular cross sections

To reassure the absence of any material defects, a microstructural analysis was also carried out, as follows:

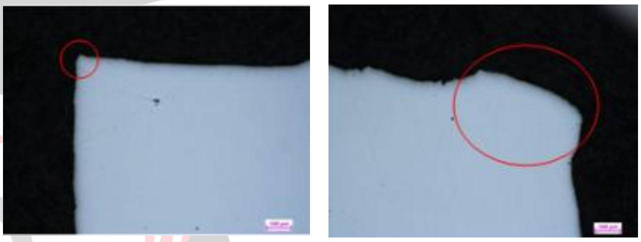

Fig.17: No abnormality was observed in fracture initiation and termination regions; Inclusion content of steel was also within specification

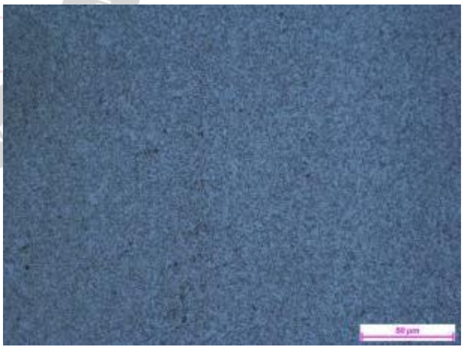

Fig.18: The core microstructure showed uniformly distributed fine tempered martensite

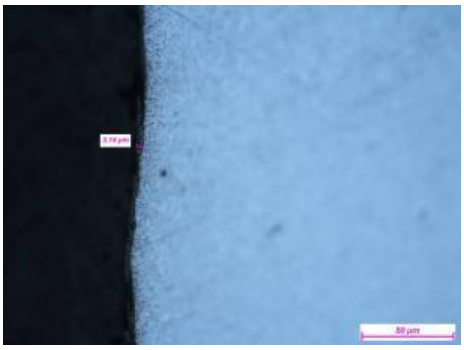

Fig.19: The oxidation depth on the valve neck area was found to be 3.14 micron, which is very minimal and can be negligible

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The fracture occurred in the valve neck area which is at a distance of 1.0-2.0 mm before profile turning end zone. No step mark was observed in the fracture zone in any of the valves. Valve contact pattern analysis with respect to stem, end and groove inferred that the groove had undergone severe contact stress which deformed the groove even to the extent of material pull out in certain cases, despite the groove dimensions meeting specifications.

From the above analysis results, it could be inferred that the step mark was not the cause for this failure, as the fracture did not occur at the merging point of profile turning and grinding. However, from the fracture surface characteristics, it was inferred that repeated stem stress was the cause for this fracture, as the fracture had occurred in the high cycle fatigue mode.

This repeated stress also reflected in the groove contact pattern, which was evidenced from abnormal contact pattern, even to the extent of material removal. Hence, this repeated stem stress might have arisen from the following conditions during service:

- Too high seating velocity.
- Valve bounce due to over revving.
- Weak springs.
- Incorrect valve train clearance.
- Excessive valve lash.

IV. Computer Aided Design & Analysis

In order to conduct further computer aided analysis (FEA) on the valves to find the effect of the operating and loading conditions on the valve, the computer aided design of the valve was done for various iterations in crucial dimensions, i.e., profile radius, profile angle and seat angle. Due to the functional constraints in the original engine design, only the following could be modified, if need be. Both 2D and 3D designs were prepared using the valve design critical characteristics specified in the drawing - profile radius, profile angle and seat angle.

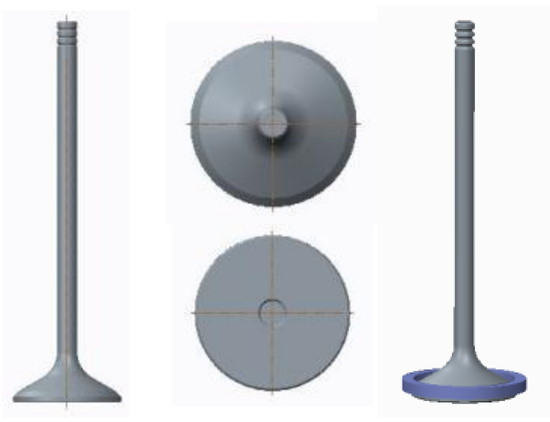

Fig.20: 3D CAD Modelling of the valve in consideration

Fig.21: 2D cross section and 3D sectional view of the model

Appropriately loaded 2D and 3D structural analyses were conducted for the existing mean value of dimensions and the results were taken as shown in Fig.22 and Fig.23.

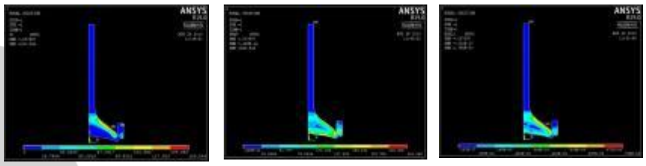

Fig.22: Sample 2D loaded structural analysis to find principal mechanical stress, von Mises stress and principal elastic strain

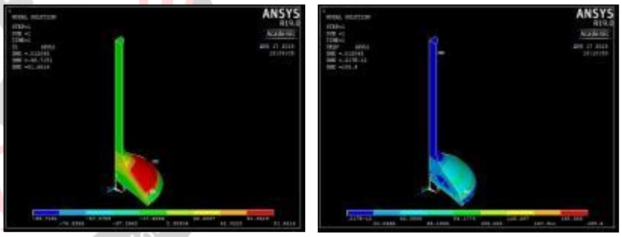

Fig.23: Sample 3D loaded structural analysis to find principal mechanical stress and von Mises stress

Appropriately loaded 2D and 3D thermal analyses were also conducted as shown in Fig.24 and Fig.25.

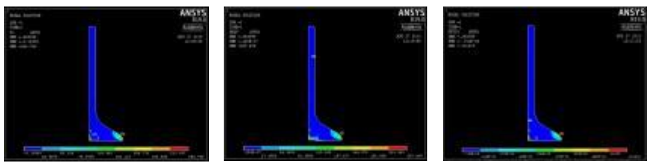

Fig.24: Sample 2D loaded thermal analysis to find principal thermal stress, von Mises stress and principal elastic strain

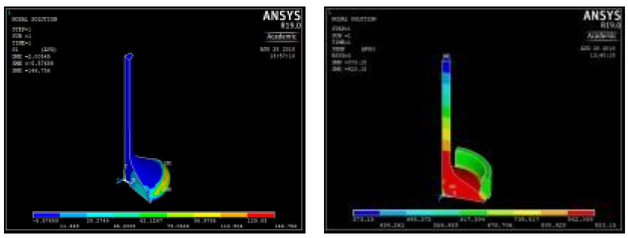

Fig.25: Sample 3D loaded thermal analysis to find principal thermal stress and temperature distribution

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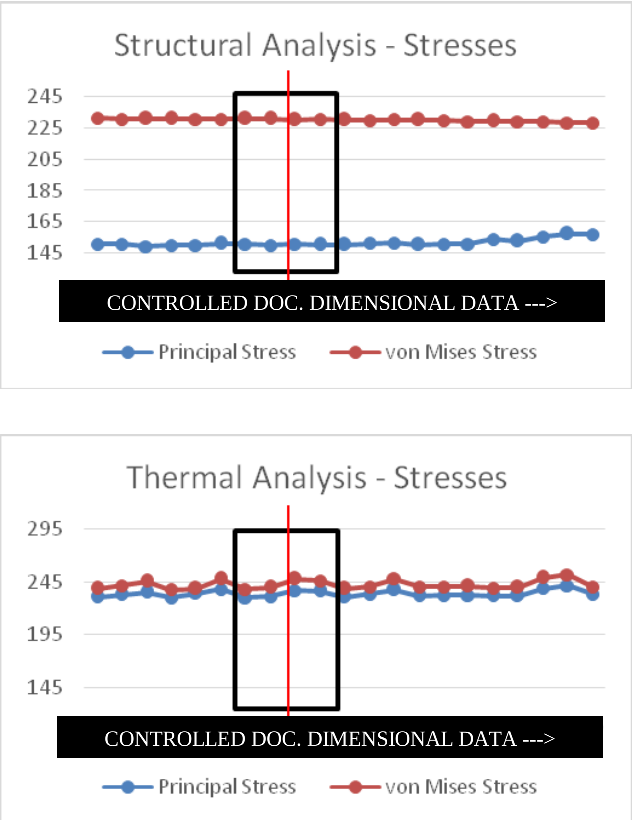

Fig.26: Stress (along Y-axis, in MPa) plots for varying profile radii (along X-axis, in mm) taken during structural and thermal analyses using FEA

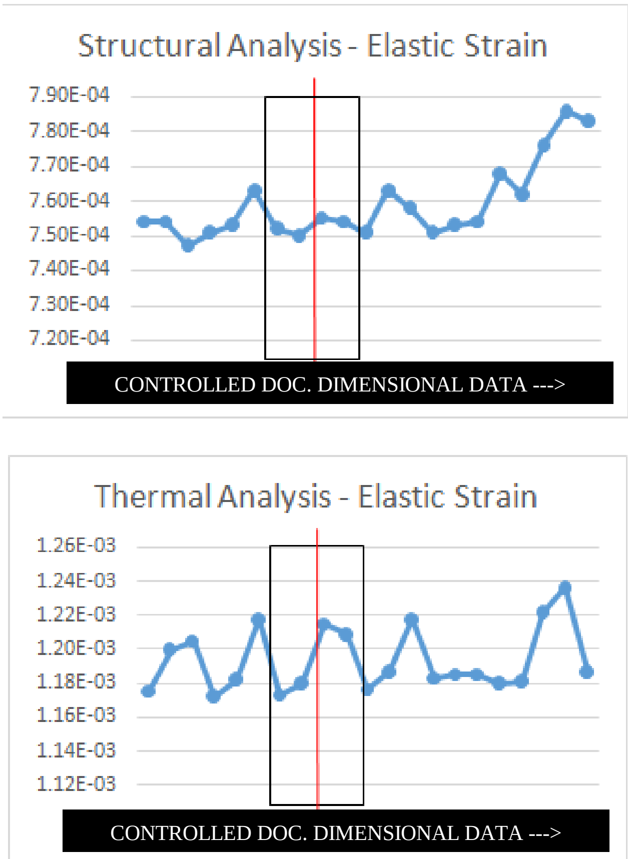

Fig.27: Elastic Strain (along Y-axis, in mm/mm) plots for varying profile radii (along X-axis, in mm) taken during structural and thermal analyses using FEA

Further, due to the possibility of damages (due to manufacturing, assembly, logistics, etc.) also causing failures, additional analyses were also carried out to simulate issues encountered at the stem, the profile and due to bending in general (which is the core cause of the failure – unidirectional bending).

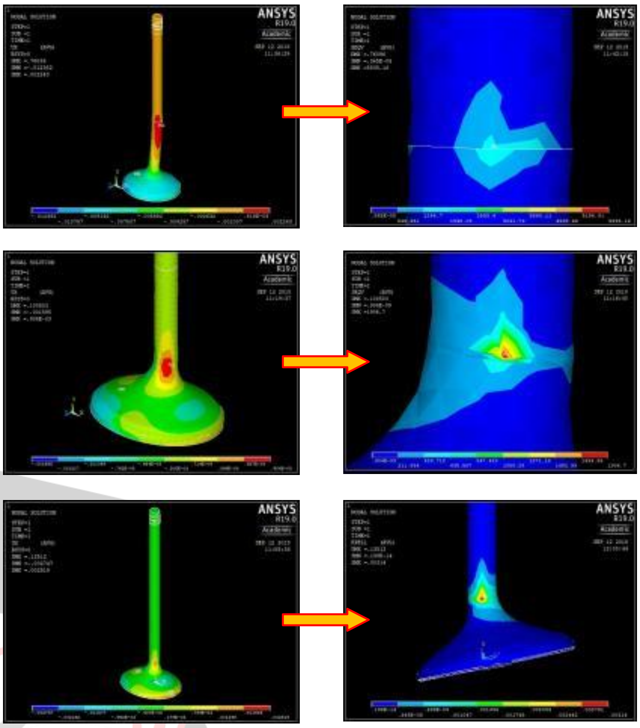

Fig.28: FEA output of analysis on damaged valve (broken at stem, fillet region and due to unidirectional bending respectively), showing X-component of nodal deformation and the stress distribution

The fatigue beach marks observed as the output due to unidirectional bending was exactly the same as what the physical analysis results showed, as seen in Fig.29, which also depicts the deformed + undeformed output to see the extent of deformation comparatively.

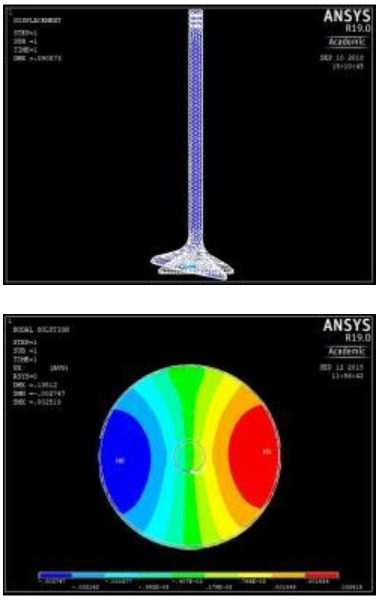

Fig.29: FEA output of analysis on damaged valve due to unidirectional bending (ANSYS: def+undef), and, the beach marks observed at the broken region

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V. EXPERIMENTAL DESIGN

A. Initial structural analysis output (varying profile radius only) - DoE input #1

Profile Radius (mm)	Elastic Strain (mm/mm)	Principal Stress (MPa)	von Mises Stress (MPa)	O/P
[redacted] CONTROLLED DOC. DIMENSIONAL DATA →	7.54E-04	150.845	231.299	
[redacted]	7.54E-04	150.73	230.497	
[redacted]	7.47E-04	149.227	230.811	
[redacted]	7.51E-04	149.98	230.813	
[redacted]	7.53E-04	150.118	230.526	
[redacted]	7.63E-04	151.34	230.449	
[redacted]	7.52E-04	150.701	230.555	
[redacted]	7.50E-04	150.222	230.657	
[redacted]	7.55E-04	150.844	229.846	
[redacted]	7.54E-04	150.436	230.352	
[redacted]	7.51E-04	150.476	230.011	
[redacted]	7.63E-04	151.388	229.756	
[redacted]	7.58E-04	151.723	230.121	
[redacted]	7.51E-04	150.484	229.883	
[redacted]	7.53E-04	151.017	229.646	
[redacted]	7.54E-04	150.675	228.901	
[redacted]	7.68E-04	153.846	229.17	
[redacted]	7.62E-04	152.889	228.957	
[redacted]	7.76E-04	155.643	228.72	
[redacted]	7.86E-04	157.375	228.039	
[redacted]	7.83E-04	156.871	228.107	

Present design
Recommended design

Table 1: ANSYS output data for structural analysis varying only the profile radius by 0.5 mm per iteration

B. Initial thermal analysis output (varying profile radius only) - DoE input #2

Profile Radius (mm)	Elastic Strain (mm/mm)	Principal Stress (MPa)	von Mises Stress (MPa)	O/P
[redacted] CONTROLLED DOC. DIMENSIONAL DATA →	1.18E-03	230.137	238.382	
[redacted]	1.20E-03	232.5	241.5	
[redacted]	1.21E-03	235.109	245.168	
[redacted]	1.17E-03	229.854	237.515	
[redacted]	1.18E-03	233.928	238.164	
[redacted]	1.22E-03	237.951	247.411	
[redacted]	1.17E-03	230.008	237.64	
[redacted]	1.18E-03	230.949	239.372	
[redacted]	1.22E-03	236.759	247.459	
[redacted]	1.21E-03	236.107	245.756	
[redacted]	1.18E-03	230.559	238.348	
[redacted]	1.19E-03	233.029	240.378	
[redacted]	1.22E-03	237.417	247.794	
[redacted]	1.18E-03	231.656	240.094	
[redacted]	1.19E-03	231.88	240.288	
[redacted]	1.19E-03	231.95	240.496	
[redacted]	1.18E-03	231.407	239.119	
[redacted]	1.18E-03	231.386	239.459	
[redacted]	1.22E-03	238.623	248.456	
[redacted]	1.24E-03	241.414	251.667	
[redacted]	1.19E-03	233.043	240.205	

Present design
Recommended design

Table 2: ANSYS output data for thermal analysis varying only the profile radius by 0.5 mm per iteration

C. 3x2 DoE for optimal Principal Stress, von Mises Stress and Elastic Strain

Experimental design is conducted for the three parameters - profile radius, profile angle and seat angle - across two levels - maximum and minimum - to determine the optimal and most feasible solution to the issue.

Inputs from (A) and (B) are taken to find out how the profile angle and seat angle (similar entities) react to varying profile radii (different entity).

With recommended designs being available both sides of the scale, careful 3x2 DoE is conducted to choose the better option among the two.

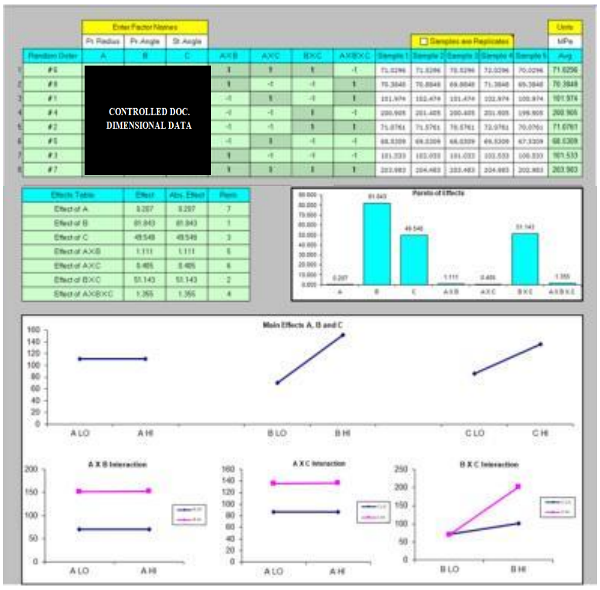

Table 3: Experimental design for Principal Stress

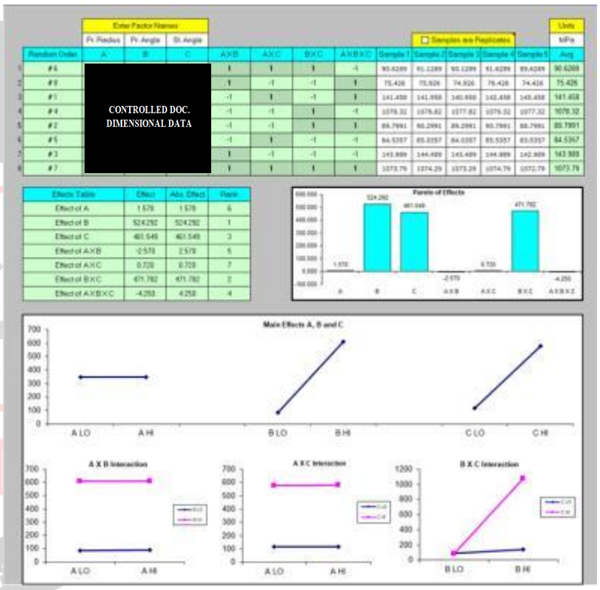

Table 4: Experimental design for von Mises Stress

Table 5: Experimental design for Elastic Strain

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VI. DESIGN OPTIMIZATION

A. Analytical Reliability

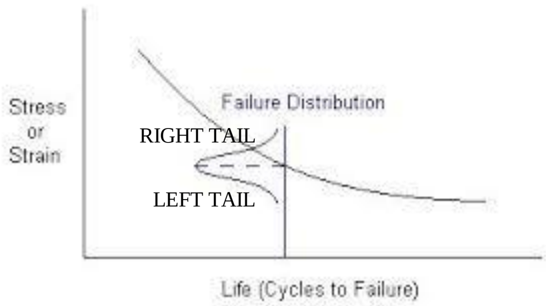

Fig.30: Stress/Strain vs. Life Cycle plot for a normal failure distribution

With the stress and strain values showing no clear trend, i.e., ascending, consistent or descending, reliability is needed to establish the authenticity of the results.

Hence, considering the two-hypotheses method, the output parameters, i.e., principal stress, von Mises stress and elastic strain, are checked for both stability and consistency, before they are validated using the recommended design characteristics.

H_0: There is no significant difference between the values.

H_1: There is a significant difference between the values.

PRINCIPAL STRESS (INTER-RATER RELIABILITY):

Sample 1	Sample 2		Sample 1	Sample 2
71.0296	71.0296	n =	**5**	**5**
71.5296	71.5296	mean =	**71.0296**	**71.0296**
70.5296	70.5296	s.d. =	**0.790569**	**0.790569**
72.0296	72.0296			
70.0296	70.0296			

TWO-TAILED T-TEST	
Lower Critical Value	-2.306004135
Upper Critical Value	2.306004135
p-value	1
Decision	**Do not reject H_0**
ONE-TAILED T-TEST (LEFT TAIL)	
Lower Critical Value	-1.859548038
p-value	0.5
Decision	**Do not reject H_0**
ONE-TAILED T-TEST (RIGHT TAIL)	
Upper Critical Value	1.859548038
p-value	0.5
Decision	**Reject H_0**

Table 6: Inter-rater reliability study for Principal Stress

PRINCIPAL STRESS (TEST-RETEST RELIABILITY):

Sample 1	Sample 2		Sample 1	Sample 2
101.974	101.533	n =	**5**	**5**
102.474	102.033	mean =	**101.974**	**101.533**
101.474	101.033	s.d. =	**0.790569**	**0.790569**
102.974	102.533			
100.974	100.533			

TWO-TAILED T-TEST	
Lower Critical Value	-2.306004135
Upper Critical Value	2.306004135
p-value	0.403498473
Decision	**Do not reject H_0**
ONE-TAILED T-TEST (LEFT TAIL)	
Lower Critical Value	-1.859548038
p-value	0.798250763
Decision	**Do not reject H_0**
ONE-TAILED T-TEST (RIGHT TAIL)	
Upper Critical Value	1.859548038
p-value	0.201749237
Decision	**Reject H_0**

Table 7: Test-retest reliability study for Principal Stress

VON MISES STRESS (INTER-RATER RELIABILITY):

Sample 1	Sample 2		Sample 1	Sample 2
90.6289	90.6289	n =	**5**	**5**
91.1289	91.1289	mean =	**90.6289**	**90.6289**
90.1289	90.1289	s.d. =	**0.790569**	**0.790569**
91.6289	91.6289			
89.6289	89.6289			

TWO-TAILED T-TEST	
Lower Critical Value	-2.306004135
Upper Critical Value	2.306004135
p-value	1
Decision	**Do not reject H_0**
ONE-TAILED T-TEST (LEFT TAIL)	
Lower Critical Value	-1.859548038
p-value	0.5
Decision	**Do not reject H_0**
ONE-TAILED T-TEST (RIGHT TAIL)	
Upper Critical Value	1.859548038
p-value	0.5
Decision	**Reject H_0**

Table 8: Inter-rater reliability study for von Mises Stress

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VON MISES STRESS (TEST-RETEST RELIABILITY):

Sample 1	Sample 2		Sample 1	Sample 2
90.6289	89.7991	n =	**5**	**5**
91.1289	90.2991	mean =	**90.6289**	**89.7991**
90.1289	89.2991	s.d. =	**0.790569**	**0.790569**
91.6289	90.7991			
89.6289	88.7991			

TWO-TAILED T-TEST	
Lower Critical Value	-2.306004135
Upper Critical Value	2.306004135
p-value	0.135577047
Decision	**Do not reject H_0**
ONE-TAILED T-TEST (LEFT TAIL)	
Lower Critical Value	-1.859548038
p-value	0.932211477
Decision	**Do not reject H_0**
ONE-TAILED T-TEST (RIGHT TAIL)	
Upper Critical Value	1.859548038
p-value	0.067788523
Decision	**Reject H_0**

Table 9: Test-retest reliability study for von Mises Stress

ELASTIC STRAIN (INTER-RATER RELIABILITY):

Sample 1	Sample 2		Sample 1	Sample 2
3.28E-04	3.28E-04	n =	**5**	**5**
3.37E-04	3.37E-04	mean =	**3.28E-04**	**3.28E-04**
3.19E-04	3.19E-04	s.d. =	**6.52E-06**	**6.52E-06**
3.30E-04	3.30E-04			
3.26E-04	3.26E-04			

TWO-TAILED T-TEST	
Lower Critical Value	-2.306004135
Upper Critical Value	2.306004135
p-value	1
Decision	**Do not reject H_0**
ONE-TAILED T-TEST (LEFT TAIL)	
Lower Critical Value	-1.859548038
p-value	0.5
Decision	**Do not reject H_0**
ONE-TAILED T-TEST (RIGHT TAIL)	
Upper Critical Value	1.859548038
p-value	0.5
Decision	**Reject H_0**

Table 10: Inter-rater reliability study for Elastic Strain

ELASTIC STRAIN (TEST-RETEST RELIABILITY):

Sample 1	Sample 2		Sample 1	Sample 2
2.89E-04	2.83E-04	n =	**5**	**5**
2.98E-04	2.92E-04	mean =	**2.89E-04**	**2.83E-04**
2.80E-04	2.74E-04	s.d. =	**6.52E-06**	**6.52E-06**
2.91E-04	2.85E-04			
2.87E-04	2.81E-04			

TWO-TAILED T-TEST	
Lower Critical Value	-2.306004135
Upper Critical Value	2.306004135
p-value	0.183698114
Decision	**Do not reject H_0**
ONE-TAILED T-TEST (LEFT TAIL)	
Lower Critical Value	-1.859548038
p-value	0.908150943
Decision	**Do not reject H_0**
ONE-TAILED T-TEST (RIGHT TAIL)	
Upper Critical Value	1.859548038
p-value	0.091849057
Decision	**Reject H_0**

Table 11: Test-retest reliability study for Elastic Strain

SUMMARY:

All T-tests done in this section are of differences between two means (unequal variances).

Table 6 and Table 7 show the culmination of the analyses done on the Principal Stress on the valve. The right tail of the normal distribution of the Principal Stress is to be rejected as higher stress would not enable the valve to provide the results expected of it - thereby making it non-optimal.

Table 8 and Table 9 show the culmination of the analyses done on the von Mises Stress on the valve. The right tail of the normal distribution of the von Mises Stress is also to be rejected as higher stress would not enable the valve to provide the results expected of it - thereby making it non-optimal.

Table 10 and Table 11 show the culmination of the analyses done on the Elastic Strain on the valve. The right tail of the normal distribution of the Elastic Strain is to be rejected as higher stress would not enable the valve to provide the results expected of it - thereby making it non-optimal.

To sum up, the least of the stresses and strains is considered to be safest for the optimal redesign of the valve. Hence the left tail is not rejected in any case (Ref: Fig.30).

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B. Manufacturing Reliability

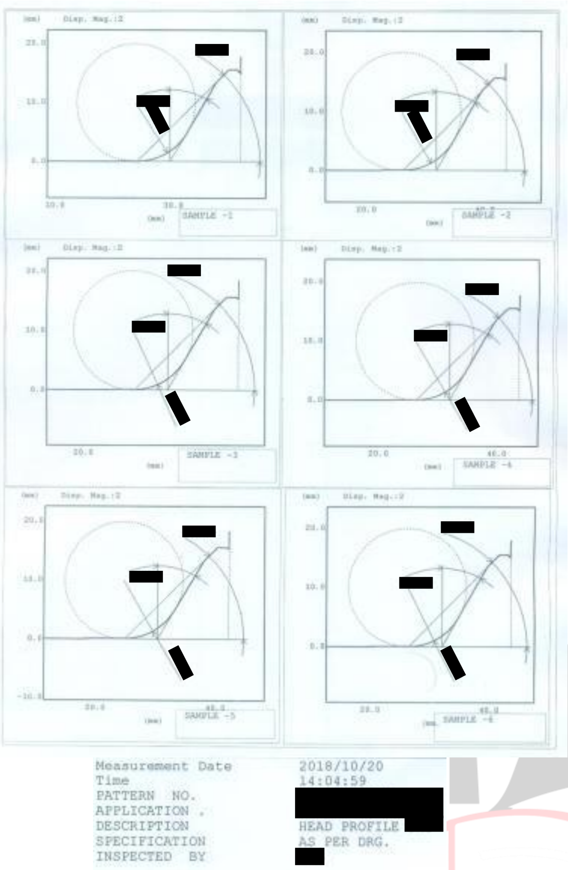

Fig.31: Sample profile parameters measurement statistics shown for the six different boxes of valves studied during final inspection (Note: dimensions are hidden to protect controlled data)

H_0: There is manufacturing reliability for the dimensional parameter.
H_1: There is no manufacturing reliability for the dimensional parameter.

Manufacturing reliability is also checked to ensure there are no major changes to the equipment/machinery that needs to be done to accommodate the changes, if any, and to ascertain whether costs incurred in the manufacture of the new valve do not differ majorly with the existing value.

PROFILE RADIUS:

Sample 1	Sample 2		Sample 1	Sample 2
CONTROLLED DOC. DIMENSIONAL DATA		n =	6	6
		mean =	9.95583	9.98083
		s.d. =	0.040509	0.074759

TWO-TAILED T-TEST	
Lower Critical Value	-2.364624252
Upper Critical Value	2.364624252
p-value	0.494746077
Decision	**Do not reject H_0**

Table 12: Reliability study for Profile Radius

PROFILE ANGLE:

Sample 1	Sample 2		Sample 1	Sample 2
CONTROLLED DOC. DIMENSIONAL DATA		n =	6	6
		mean =	30.01935	30.02602
		s.d. =	0.047434	0.048811

TWO-TAILED T-TEST	
Lower Critical Value	-2.262157163
Upper Critical Value	2.262157163
p-value	0.815763254
Decision	**Do not reject H_0**

Table 13: Reliability study for Profile Angle

SEAT ANGLE:

Sample 1	Sample 2		Sample 1	Sample 2
CONTROLLED DOC. DIMENSIONAL DATA		n =	6	6
		mean =	45.24593	45.25593
		s.d. =	0.039553	0.039553

TWO-TAILED T-TEST	
Lower Critical Value	-2.228138852
Upper Critical Value	2.228138852
p-value	0.670763954
Decision	**Do not reject H_0**

Table 14: Reliability study for Seat Angle

SUMMARY:

All T-tests done in this section are of differences between two means (unequal variances).

To include both sides of tolerance limits of the dimensions of the parameters aforementioned, i.e., +\-, the two-tailed is done as is without considering any tail in specific.

For all parameters - profile radius, profile angle and seat angle - the decision obtained from the analyses was to not reject H_0 on both tails. This implies that the manufacturing standards currently in use can accommodate the changes, if any, and neither the accuracy of manufacture nor the precision of quality checking will be affected by the same.

The inputs fed for this line of tests were completely based on the existing machinery and equipment used to make the engine valve.

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VII. Validation of Optimized Valve

A. Configuration Alternative #1

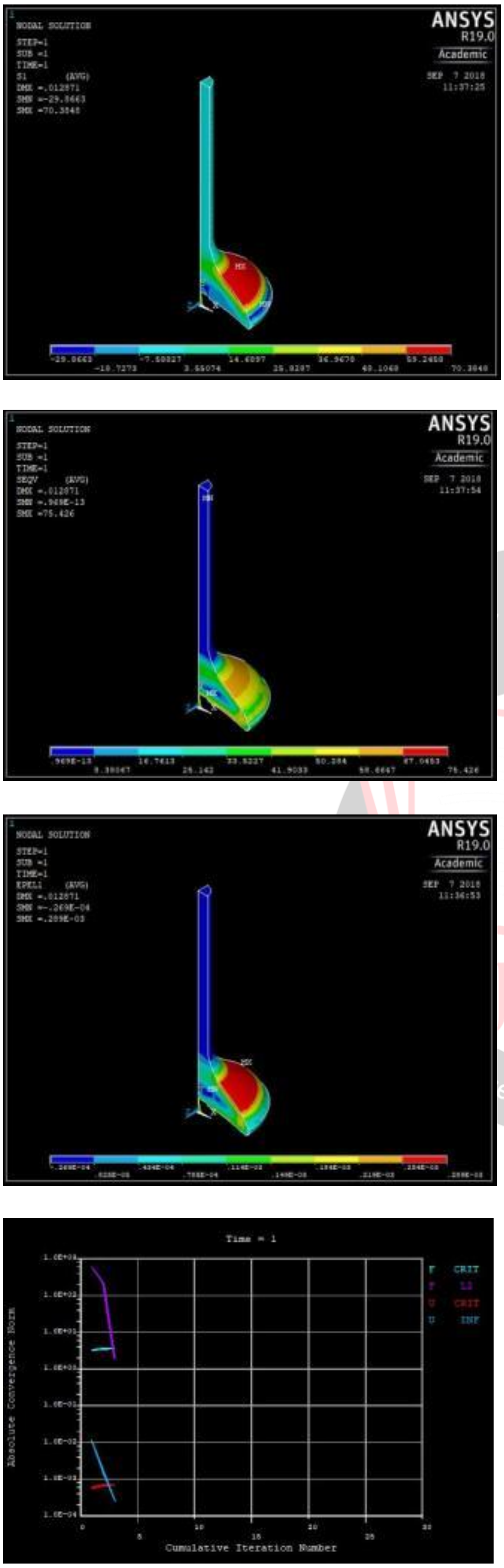

Fig.32: Principal Stress, von Mises Stress, Elastic Strain and Convergence Iteration plot of the alternative #1 obtained from transient finite element analysis, for comparison purposes.

B. Configuration Alternative #2

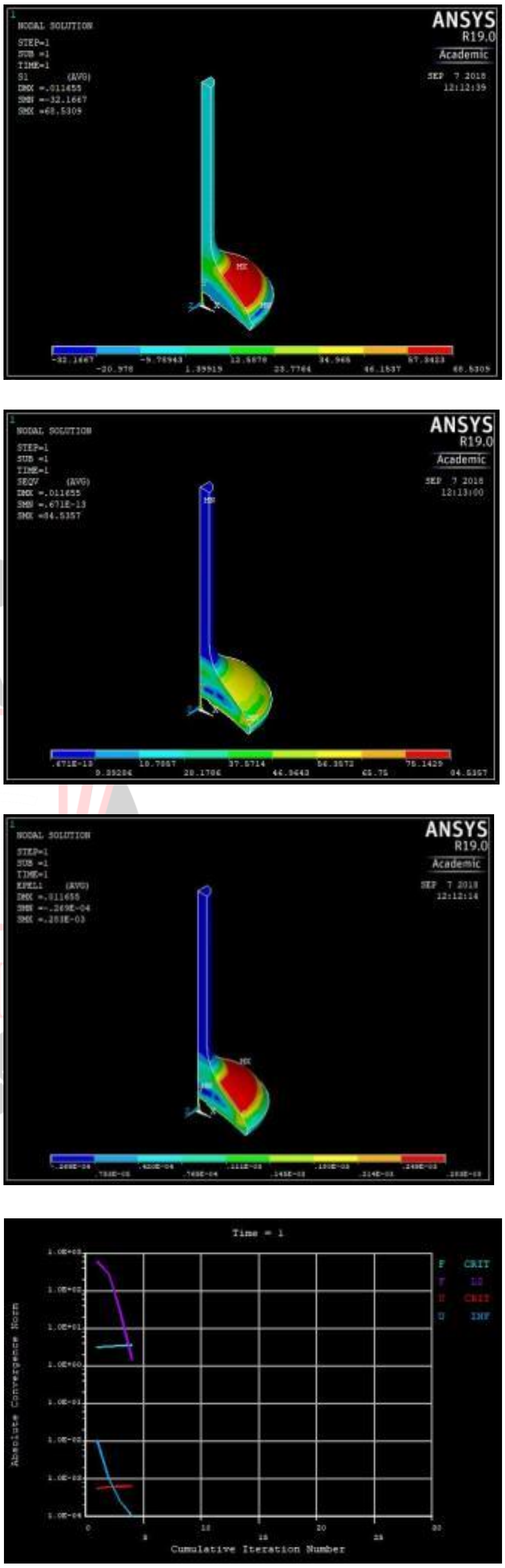

Fig.33: Principal Stress, von Mises Stress, Elastic Strain and Convergence Iteration plot of the alternative #2 obtained from transient finite element analysis, for comparison purposes.

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Comparison of the two shortlisted configurations as replacement for existing design is done to pick the most suitable alternative to solve the problem. Considering no change in the engine specifications, only things in the direct control of the company can be modified - within the existing tolerance limits, to be economical and minimalistic.

According to the analytical and manufacturing reliability studies, the configuration alternative #2 is picked over #1, due to the consistent T-test outputs related to both stresses and strain going in its favour.

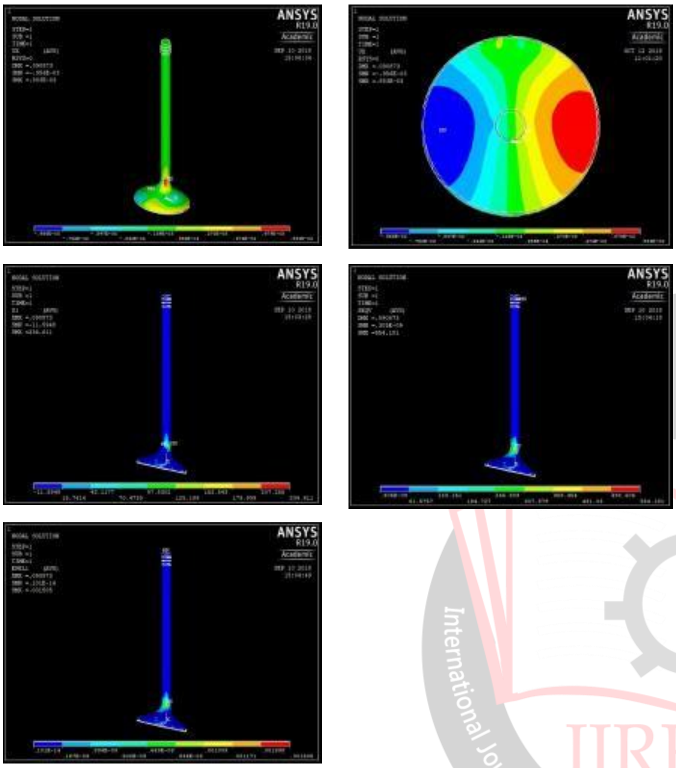

Fig.34: FEA output of analysis on alternate selection of configuration alternative #2 bent valve, showing X-component of nodal deformation, fatigue beach marks on valve due to unidirectional bending stresses which leads to failure, principal stress at the broken region, von Mises stress at the broken region and elastic strain at the broken region

It needs to be noted that, for the same operating conditions, the stress dealt with by the valve is almost halved, and there is a significant difference in the strain value. This means that, although failure is inevitable, the valve will experience reduced degree of bending and therefore have a much longer operational life.

VIII. Material Optimization

SUH3 material has been predominantly used as an inlet valve material, and therefore, the valve in consideration has competitive materials to potentially replace the existing one in this regard. Materials like high temperature titanium, SUH35, etc. are suitable options. Materials that are commonly used for performance valve applications include carbon steel alloys, stainless steels, high-strength nickel chromium-iron alloys and titanium. The alloys that are most commonly used for performance engines include various high chromium stainless alloys for intake valves, Inconel in common, and 21-4N (EV8) for exhaust valves. Exhaust valves may be made from a martensitic steel with chrome and silicon alloys, or a two-piece valve with a stainless steel head and martensitic steel stem.

SUH3's S-N curve and a sample comparison of SUH3 and SUH35 steel for the same application is given below:

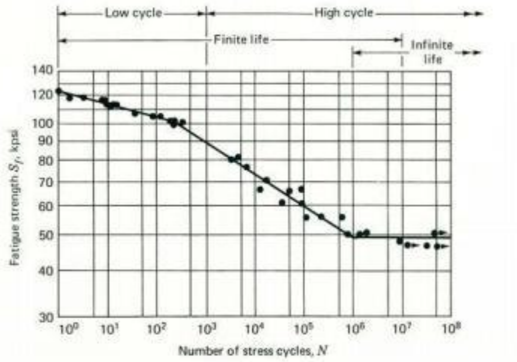

Fig.35: S-N Curve of SUH3 Grade Steel

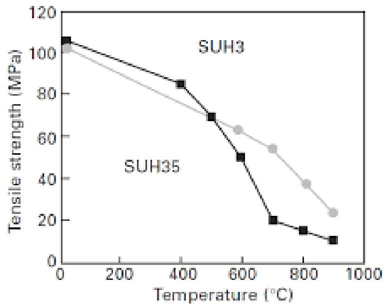

Fig.36: A comparison of SUH3 and SUH35 tensile strength vs. temperature plots

While SUH3 offers good tensile strength up to 800 deg. Celsius, SUH35 goes much beyond that limit to retain its properties. SUH35 also offers better oxidation resistance, and excellent sulfidation corrosion resistance, compared to SUH3.

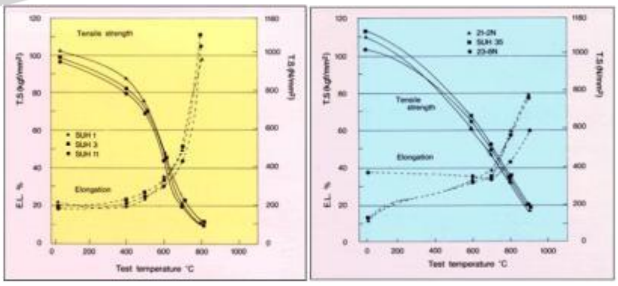

Fig.37: A comparison of SUH3 and SUH35 tensile strength/elongation percentage vs. test temperature plots

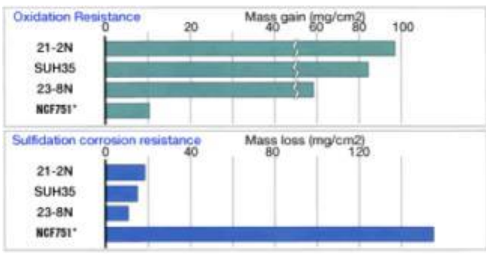

Fig.38: SUH35 oxidation and sulfidation resistance comparison

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With SUH3 and SUH35 similarly priced at $1.1 - $3.5 per kg, SUH35 is therefore the better suited material for the intended application - heat resisting engine valve steel.

IX. Fatigue Analysis Comparison

Fig.39: Fatigue analysis results of the incumbent valve model

Fig.40: Fatigue analysis results of the alternative #2 model (chosen as replacement)

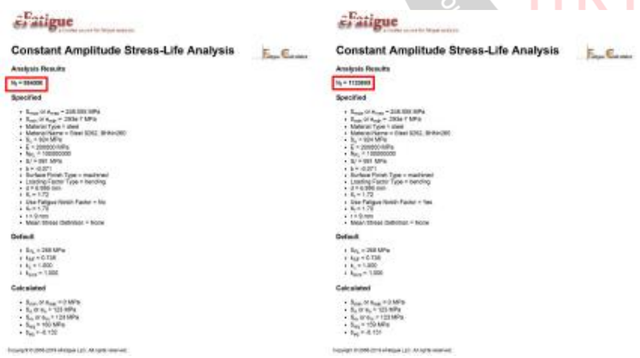

Fig.41: Fatigue analysis of the alternative #1 model (not chosen as replacement)

The current configuration has an **Nf** (number of cycles before fatigue failure) of just 728000 without notch factor, and 816000 with notch factor. Similarly, alternative #2 has 1460000 and 1640000 respectively, while alternative #1 has 994000 and 1120000 respectively. This shows that the alternative #2 can sustain the loads for a longer period of time, i.e., it has a greater endurance limit. This makes it the better design among the available configurations.

X. Mistake-Proofing (Poka-Yoke)

The necessity of mistake-proofing a component or an entire product itself is to ensure its sustainable quality and performance. The purpose of poka-yoke is to eliminate defects by preventing, correcting, or drawing attention to inadvertent human errors as they occur. The poka-yoke solutions obtained for the vital issues, which make up ~80% of the problem, are as follows:

Identified Problem	Solution (POKA-YOKE)
Premature fracture at the profile-stem intersection (neck) region of similar non-automotive engine valves.	Check and replace worn valve guide, as it fails to assist the valve to close squarely on the seat, and creates bending of valve stem. This reduces threat to transition region of valve. Also check valve-guide clearance (mm) with respect to stem diameter (mm).
Source of Problem	Address badly worn valve tips, if any, as these increase side loading.
High Stem Side Loading.	
Type of Defect / Error	After overhaul, correct valve material volume symmetry needs to be checked, viz-a-viz valve seat and face, due to operational material recession.
Fatigue Failure; Permanent.	

Table 15: Poka-Yoke worksheet for high stem side laoding

Identified Problem	Solution (POKA-YOKE)
Premature fracture at the profile-stem intersection (neck) region of similar non-automotive engine valves.	Spring to be kept in almost coil-bound condition during fully open valve position, to return the coil spring to a uniform, stable shape on every closing cycle.
	Less aggressive camshaft to regulate the speed at which valve is propelled off the valve seat. The lifter needs to stay in contact with the cam lobe to retain valve spring longevity.
Source of Problem	
High Spring Pressure.	
Type of Defect / Error	
Fatigue Failure; Permanent.	

Table 16: Poka-Yoke worksheet for high spring pressure

Identified Problem	Solution (POKA-YOKE)
Premature fracture at the profile-stem intersection (neck) region of similar non-automotive engine valves.	Prevent over-speeding of engine by adding speed governor and have a parallel connection via ECU to valve train sensors.
	Check lack of valve rotation during running conditions, which creates excess forces on the same region of the seat.
Source of Problem	
High Seating Forces.	Excessive circumferential seal force can cause accelerated wear. As temperatures increase, the pressure between seat and ball increase, leading to greater friction. Ensure that the coefficient of friction between materials is ideal and select appropriate finish for both seat and insert.
Type of Defect / Error	
Fatigue Failure; Permanent.	

Table 17: Poka-Yoke worksheet for high seating forces

Identified Problem	Solution (POKA-YOKE)
Premature fracture at the profile-stem intersection (neck) region of similar non-automotive engine valves.	Eliminate steep (harsh) opening and closing ramps of cam profile to eliminate high valve lash.
	Adjust valve lash for each maintenance and service cycle, only when engine is warmed up and running, using the same feeler gauge, to avoid simulative errors and premature valve failure.
Source of Problem	
Excessive Valve Lash.	
Type of Defect / Error	Prevent over-speeding of engine by adding speed governor and have a parallel connection via ECU to valve train sensors.
Fatigue Failure; Permanent.	

Table 18: Poka-Yoke worksheet for excessive valve lash

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XI. Conclusion

Physical analysis of the failed valve(s) was conducted to ascertain the root cause of the problem, and, from the fracture surface characteristics, it was inferred that repeated stem stress was the cause for this fracture, as the fracture had occurred in the high cycle fatigue mode. The non-automotive valve in consideration was designed using PTC Creo software for subsequent usage in FEA. More models were created by altering only the profile radius and analyses were carried out to find out the optimal dimension. Further, models were created to accommodate each of the iterations considered for DoE with suitable dimensional increments.

Appropriately loaded (2D/3D) structural and thermal analyses (transient FEA) was carried out for each of the designed models, for both open and closed conditions, using ANSYS APDL software. The DoE templates were set based on the obtained results – Principal Stress, von Mises Stress and Elastic Strain.

Additional damage-based analyses of the valve was conducted for three distinct conditions – damage at stem region, at profile region and due to unidirectional bending. Experiments were conducted using DoE and the interactions between factors across levels were observed. Two configurations were selected for final optimization study.

The valve design was optimized based on the analyses for peak operating and loading conditions. This was done using statistical t-tests to determine if the hypotheses could be accepted or rejected based on the constructed experimental design. It was found that outputs having more stresses, despite having better factor interactions, were not suitable for the intended purpose. Manufacturing reliability was checked for sample populations of 10 each for the past 6 months of mass production and the changes in dimensions were noted using the Profile Projector apparatus. It was observed that the machines could absorb the suggested changes and there need not be any expensive alterations made.

The optimized valve (alternative #2) was validated using ANSYS APDL software and compared with the initial configuration model valve. It was found out that stresses on the valve were significantly reduced and the strain due to fatigue had halved. The optimal material was found to be SUH35 instead of SUH3 in terms of similarity. The fatigue analysis results of the valves were obtained using Altair e-Fatigue Calculator, to validate the claims made by the ANSYS outputs. The fatigue life of the products were calculated for the 10^8 lifecycle period, and the comparative results were taken both with and without considering the notch factor. The mistake-proofing (poka-yoke) sustainability worksheets were generated for the project. For extrapolation of the project to other valve models – automotive or non-automotive – SOP creation was suggested to include phase-wise FEA and DoE in design optimizations.

Appendix

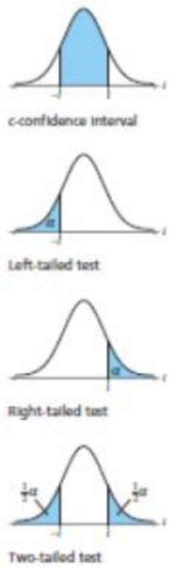
Common loading conditions used on the model for FEA:

Structural Analyses	Thermal Analyses
Symmetry: Axisymmetric Displacement Constraints: Seat insert top face and outer curved surface. For bending analysis – one outer half of stem chrome flashed region. Contact Pair: Valve seat face and seat insert mating surface. Mechanical Loads: Combustion gas pressure (max) = 13 MPa Combustion gas pressure (min) = 5 MPa Spring force (max) = 500 N Spring force (min) = 250 N Seating force = 500 N	Symmetry: Axisymmetric Displacement Constraints: Seat insert top face and outer curved surface. Contact Pair: Valve seat face and seat insert mating surface. Thermal Loads: Temp. at valve face and dimple = 650°C Temp. at seat insert top and outer curved surface = 350°C Temp. at profile fillet = 650°C Temp. at oil cooled region of valve stem (tip) = 100°C

Note:
For fatigue analysis and N_f calculation, both analyses were considered as a couple-field analyses for open and closed conditions of valve respectively. Similarly, both damage-proof and damage-based analyses were carried out without and with the suitable notch factor (K_f) of 1.70, respectively.

t-Test Table for various levels of confidence:

d.f.	Level of confidence, c	0.80	0.90	0.95	0.98	0.99
	One tail, α	0.10	0.05	0.025	0.01	0.005
	Two tails, α	0.20	0.10	0.05	0.02	0.01
1		3.078	6.314	12.706	31.821	63.657
2		1.886	2.920	4.303	6.965	9.925
3		1.638	2.353	3.182	4.541	5.841
4		1.533	2.132	2.776	3.747	4.604
5		1.476	2.015	2.571	3.365	4.032
6		1.440	1.943	2.447	3.143	3.707
7		1.415	1.895	2.365	2.998	3.499
8		1.397	1.860	2.306	2.896	3.355
9		1.383	1.833	2.262	2.821	3.250
10		1.372	1.812	2.228	2.764	3.169
11		1.363	1.796	2.201	2.718	3.106
12		1.356	1.782	2.179	2.681	3.055
13		1.350	1.771	2.160	2.650	3.012
14		1.345	1.761	2.145	2.624	2.977
15		1.341	1.753	2.131	2.602	2.947
16		1.337	1.746	2.120	2.583	2.921
17		1.333	1.740	2.110	2.567	2.898
18		1.330	1.734	2.101	2.552	2.878
19		1.328	1.729	2.093	2.539	2.861
20		1.325	1.725	2.086	2.528	2.845
21		1.323	1.721	2.080	2.518	2.831
22		1.321	1.717	2.074	2.508	2.819
23		1.319	1.714	2.069	2.500	2.807
24		1.318	1.711	2.064	2.492	2.797
25		1.316	1.708	2.060	2.485	2.787
26		1.315	1.706	2.056	2.479	2.779
27		1.314	1.703	2.052	2.473	2.771
28		1.313	1.701	2.048	2.467	2.763
29		1.311	1.699	2.045	2.462	2.756
30		1.310	1.697	2.042	2.457	2.750
31		1.309	1.696	2.040	2.453	2.744
32		1.309	1.694	2.037	2.449	2.738
33		1.308	1.692	2.035	2.445	2.733
34		1.307	1.691	2.032	2.441	2.728
35		1.306	1.690	2.030	2.438	2.724
36		1.306	1.688	2.028	2.434	2.719
37		1.305	1.687	2.026	2.431	2.715
38		1.304	1.686	2.024	2.429	2.712
39		1.304	1.685	2.023	2.426	2.708
40		1.303	1.684	2.021	2.423	2.704
45		1.301	1.679	2.014	2.412	2.690
50		1.299	1.676	2.009	2.403	2.678
60		1.296	1.671	2.000	2.390	2.660
70		1.294	1.667	1.994	2.381	2.648
80		1.292	1.664	1.990	2.374	2.639
90		1.291	1.662	1.987	2.368	2.632
100		1.290	1.660	1.984	2.364	2.626
500		1.283	1.648	1.965	2.334	2.586
1000		1.282	1.646	1.962	2.330	2.581
∞		1.282	1.645	1.960	2.326	2.576



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