The dimensional variation analysis of complex mechanical systems

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By

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This thesis is respectfully dedicated to the memory of

Charlotte Edna and Leslie Thomas Sleath

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ABSTRACT

Dimensional variation analysis (DVA) is a computer based simulation process used to identify potential assembly process issues due the effects of component part and assembly variation during manufacture.

The sponsoring company has over a number of years developed a DVA process to simulate the variation behaviour of a wide range of static mechanical systems. This project considers whether the current DVA process used by the sponsoring company is suitable for the simulation of complex kinematic systems. The project, which consists of three case studies, identifies several issues that became apparent with the current DVA process when applied to three types of complex kinematic systems. The project goes on to develop solutions to the issues raised in the case studies in the form of new or enhanced methods of information acquisition, simulation modelling and the interpretation and presentation of the simulation output

Development of these methods has enabled the sponsoring company to expand the range of system types that can be successfully simulated and significantly enhances the information flow between the DVA process and the wider product development process

KEY WORDS

Concurrent engineering, Dimensional management, Dimensional variation analysis, Kinematic constraint map, New product development, Three dimensional visualisation of variation distributions,

PREFACE

The Engineering Doctorate (EngD) programme at Loughborough University was instigated to address challenging and significant industrial problems from an academic viewpoint. The aim of the programme is to provide industrially relevant solutions backed by the full rigor of scientific research.

The research was undertaken in conjunction with i-dmsolutions Ltd one of the most experienced Dimensional Variation Analysis providers in the UK. The research is primarily intended to review and revise the methods used by the company when undertaking the dimensional variation analysis of complex mechanical assemblies. The five chapters of the thesis comprise;

Chapter One	The background to the research, the aims and objectives of the research and the justification for the project
Chapter Two	A review of related work
Chapter Three	The methodology adopted
Chapter Four	The research undertaken, preliminary and case study findings
Chapter Five	The contribution to existing practice and the implications for the sponsoring company and the wider industry

The thesis is supported by four peer reviewed papers. The papers are attached as appendices to the thesis and form an integral part of the overall work

ACRONYMS / ABBREVIATIONS

3D	Three dimensional
3DCS	Dimensional Control System's 3DCS V5 software
ATDC	After top dead centre
CAD	Computer aided design
CAE	Computer aided engineering
CAM	Computer aided manufacture
CE	Concurrent engineering
Cetol	Sigmetrix's Cetol 6 Sigma V7.2 software
DFC	Datum flow chain
DFX	Design for X
DLM	Direct linearised method
DM	Dimensional management
DOHC	Double overhead cam
DOF	Degree of freedom
DVA	Dimensional variation analysis
EDM	Engineering data management
FEA	Finite element analysis
FEAD	Front end accessory drive
GASAP	Geometric As Soon As Possible
KCM	Kinematic constraint map
MCS	Monte Carlo simulation
MSM	Method of system moments
NPD	New product design
PDM	Product data management
PPM	Parts per million
QC	Quality control
RSS	Root sum square
TDC	Top dead centre
VisVSA	Teamcenter's VisVSA 2005 SR1 software
WC	Worst case

GLOSSARY

Base component	The one component in an assembly which is not located from another component and from which all the other components are located either directly or indirectly
Base engine	An assembly comprising engine block, crankshaft, connecting rods, gudgeon pins and pistons.
Castor angle	The angle between the steering axis and the vertical plane when viewed from the side of the vehicle
Complex kinematic system	An assembly system containing one or more continuous movement ranges and one or more key characteristics that requires to be measured over all or part the system movement range(s).
Constraint propagation chain	A graphical representation of the degrees of freedom constrained by the assembly of a series of component parts.
Default orientation	The alignment of the component parts of a system in the CAD model

Dimensional variation analysis

A process which simulates the effects of component part and assembly variation on a system allowing the probable dimensional variation behaviour of the entire system to be determined.

Dimensional variation behaviour

The cumulative effect of the propagation of individual component part and assembly variations within a system.

Key characteristic

A measurable system attribute that is critical to the assembly operation or performance of the system.

Mechanism

A system that has a limited number of different configurations. The means by which the system changes from one configuration to another is not considered

Movement range

The continuous distance over which a kinematic system, or part thereof, may occupy any position between the two extremities

Static system

A system which has one and only one configuration which kinematically constrains all six degrees of freedom of every component part

Variation propagation chain

The series of component parts and interfaces between the base component and any other component part that constrain a given degree of freedom.

CONTENTS

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LIST OF PAPERS

The following papers, included in the appendices, have been produced in partial fulfilment of the award requirements of the Engineering Doctorate during the course of the research and form an integral part of the overall work.

PAPER 1 (SEE APPENDIX A)

The Use of a kinematic constraint map to prepare the structure for a dimensional variation analysis model

PAPER 2 (SEE APPENDIX B)

The use of virtual fixtures, jigs and gauges in dimensional variation analysis, simulation models.

PAPER 3 (SEE APPENDIX C)

A method to visualise the 3D dimensional variation behaviour of kinematic systems

PAPER 4 (SEE APPENDIX D)

The use of a two stage dimensional variation analysis model to simulate the action of a hydraulic tappet adjustor in a car engine valve train system

1 INTRODUCTION

This chapter outlines the background to the research and sets forth the aims and objectives of the research project. The issues facing the sponsoring company are also discussed.

1.1 BACKGROUND TO THE RESEARCH

The first 3D computer based variation simulations appeared, as the result of software developed within the American automotive industry, on the industrial scene in the late 1970's. General Motors being one of the early pioneers in the field. These early simulations often employed Monte Carlo simulation and a point based user interface and were restricted to simple mechanical assemblies of no more than a dozen component parts. The limited capability was a result of, the infancy of the software, the need to manually encode the model and the limited computer power available at the time. In the mid 1990's a kinematic method based on vector loop analysis was introduced (Sigmetrix, 2006) which significantly increased DVA capability. The dramatic increase in the power of personal computers over the last two decades has seen a radical change in the methods engineers use to predict, analyse and more importantly manipulate the effects of minor variations in component size, shape and location have on the assembly, operation and performance of the complete mechanical assembly. The traditional tolerance stack methods used in the past have, to a large extent, been superseded by the variation simulation methods used today. The simulation software now commonly available, known in industry as dimensional variation analysis (DVA) software, gives the engineer the capability to;

- Create 3D feature based models of the dimensional relationships that govern the topology, assembly, operation and performance of the components within a mechanical device.

- Simulate the creation and propagation of minor component variations throughout the complete device.

- Analyse the likely overall variation behaviour of a mechanical assembly consisting of hundreds of individual components.

- Predict the range, distribution, root causes and subsequent variation in the key overall dimensions.

- Identify and resolve potential variation problems during the design and development stage while there is still time to design out the problems.

This increase in analytical capability has enabled engineering companies to design more dimensionally robust, and as a result, more competitive products and production processes. The more proactive engineering companies have also evolved a parallel, in house, operational process, known as dimensional management (DM), within the existing product development activity. The DM process provides the necessary structure, organisation and communication to obtain the best from the analysis work. The function of a DM process is to protect the dimensional integrity of a new product and/or production process from concept through design and development and on into production. An effective DM process also enables the bi-directional flow of information between the various areas of expertise within the product development process and strongly promotes cohesive teamwork between the different engineering, technical and commercial groups involved in bringing a new product to market.

This allows decisions regarding variation problems to be made early in the product development process where any action is most cost effective. The use of DVA has also spread from the automotive sector into the aerospace (Jeffreys & Leaney, 2000) and shipbuilding

(Spicknell & Kumar, 1999) sectors and is now beginning to penetrate the packaging and pharmaceutical industries. Each sector presents its own unique challenges which will require the development of new and imaginative methods before they can be resolved

1.2   ISSUES FACING THE SPONSORING COMPANY

This research was initiated by i-dmsolutions Ltd, a company that specialises in providing the initial or additional DVA and technical support that engineering companies need to develop dimensionally robust new products, robust new production processes and to implement an effective DM process within the companies' existing product development process. As DVA capability has increased, so to has the demand for increasingly complicated analysis work and for the simulation of increasingly complex mechanical systems and devices. Kinematic systems such as automotive suspensions, base engines and valve trains are good examples of such complex mechanical systems. The components move through a range of positions during the operational cycle. Such systems are more complex to model as it is necessary to account for the kinematic relationships over a range of component positions and more complex to analyse as substantially more results are produced that have to be processed, interpreted and communicated. To remain ahead of the competition the culture within i-dmsolutions Ltd is to push the available DVA software to the limit and to constantly find new ways to stretch the boundaries of the analytical capability. The first major challenge facing the company is how best to adapt, modify or develop current practice for each new application. Each new system or device, each new analysis, raises new and often unique issues that must be overcome to deliver realistic and reliable analysis results. The second major challenge is to efficiently collect and document the data necessary to construct the DVA model and to disseminate the

analysis results and the interpretation of those results so as to maximise the benefit of the DVA. To do so the company must develop techniques at two levels. The modelling methods are developed at the technical level where precision is the key factor while the methods for collection and dissemination of data are developed at a more general level to enable the flow of information without the need for a detailed knowledge of DVA.

Over the years many imaginative solutions have been found, to meet the many challenges in terms of model building, analysis, communication and how to use the analysis work to enhance the product and process design. Some were one off solutions to answer a specific problem, other solutions were more general and have become "tricks of the trade" to deal with recurring issues. The challenge to the company is to identify and extract the more generic aspects/elements of these ad hoc/previous solutions and thereby capture the lessons learnt and the experience gained in each case by developing more effective modelling, analysis or operational methods for use in future projects. Any such methods must also take into account the fact that DVA is often introduced to a customer by means of a retrospective study of an existing product or as a fire fighting exercise when problems arise during production. In such cases DVA is often used as a stand alone process rather than as part of a DM process. Yet the methods and techniques used should ideally be equally applicable to either process to facilitate the introduction of a fully integrated DM process at a later date.

1.3   AIMS

The primary aim of the project is to improve the sponsoring company's existing DVA process and to enhance the capability to simulate and analyse the effects of variation on complex kinematic systems. The secondary aim is to enhance the information transfer process to improve the data flow and reduce the possibility of data loss and misinterpretation.

1.4   OBJECTIVES

- To model three examples of three different types of complex kinematic systems using the existing company DVA process and to note any issues arising.

- Where necessary, to adapt or modify the DVA process to overcome any issues encountered.

- To devise new methods to resolve issues that cannot be overcome by modifying the existing DVA process

- To develop new methods that improve the data flow between the DVA process and the overall DM process

1.5   JUSTIFICATION OF THE OBJECTIVES

The sponsoring company's DVA process and its links to the wider DM process were formulated when the bulk of the work carried out was on static systems. The objectives of this project address the question as to whether the company's DVA process is still valid when applied to complex kinematic systems and what, if any, modifications are required. Where the

process cannot be modified the development of new methods will restore the capability and functionality of the DVA process. The trend from static to kinematic systems has changed the nature of both the information required by the DVA process and the output it produces. The output from the case studies will be used to test the company process for transferring information from the DVA process via the DM process to the wider CE process to determine if it is capable of disseminating both static and kinematic based output. It is not just the nature of the DVA output that has changed. The nature of the input to the DVA process has also changed; kinematic systems may require multiple model configurations with differing constraint schema to achieve the desired output. It is not sufficient just to revise the DVA process, the information transfer mechanisms between the DVA process and the wider DM and CE processes must also be competent as information transfer is known to be key element of both processes. The presence of a competent information transfer system will also allow the sponsoring company to progress the customer from a simple DVA installation to an efficient DM process. This will give the customer a better return on their DVA investment and strengthen any existing concurrent engineering process used by the customer.

2 LITERATURE REVIEW

This chapter consists of an overview of relevant literature concerning DVA, DM and the overall new product development process and introduces the concepts that underpin the research project

2.1 NEW PRODUCT DEVELOPMENT (NPD)

The development of a new product is a highly complex and difficult process that entails considerable risk. As with most processes, NPD has evolved in response to changing market conditions. Traditionally NPD was divided into a number of functional activities performed sequentially. Often known as the "over the wall" approach (Figure 2-1) the sequential process resulted in long lead times and because the information flow between stages was limited quality problems often arose due to a lack of understanding of the different design, manufacturing and customer requirements.
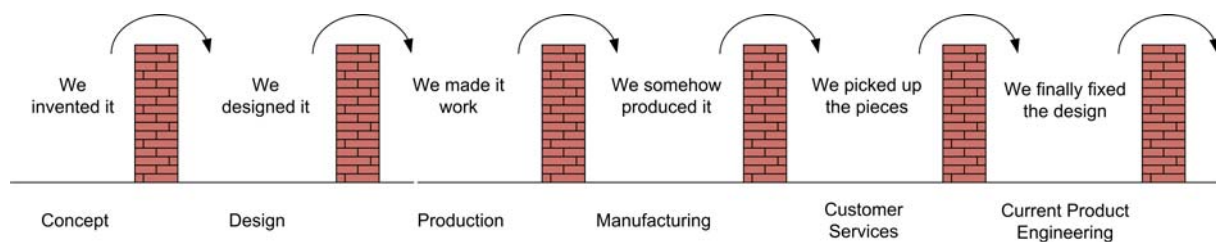
Figure 2-1 Sequential or over the wall engineering process (Baião et al., 2011)
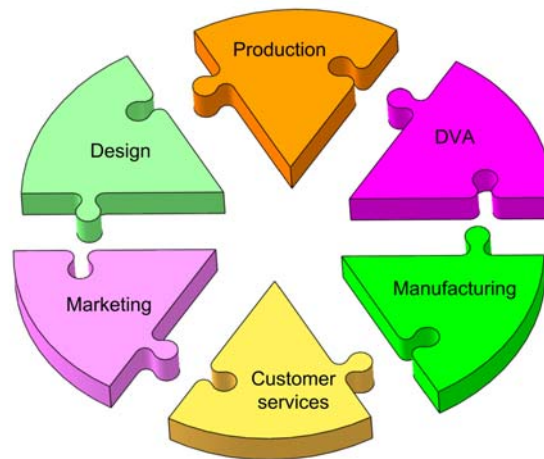
Figure 2-2 Concurrent engineering process

By the 1980's the system was unable to respond effectively to the increasing market and business uncertainties. Customers demanded an increasingly wide product range and the more frequent introduction of new products. To respond to these customer demands the concept of concurrent engineering was developed. The aim of concurrent engineering is to break down the barriers between stages in the traditional sequential NPD process and reduce the lead time for new products while improving quality and productivity (Figure 2-2).

2.2   CONCURRENT ENGINEERING

The Institute for Defence Analyses (Winner et. al., 1988) defines concurrent engineering as

"A systematic approach to the integrated, concurrent design of products and the related processes, including manufacture and support".

Keys et. al. (1992) note that while the specific implementation of a concurrent engineering process can vary significantly there are three generic elements found in concurrent engineering;

- The integration of product design, manufacturing and support processes by the use of multi-functional teams

- The use of CAD/CAE/CAM to support design integration through shared product and process models

- The use of formal evaluation methods to optimise product design, manufacture and support processes. Eg. FMEA, QFD, DFA, DFM and DVA.

Norell (1998) concludes that the concurrent engineering approach to product development requires a high level of co-operation between the functional domains with the workload distributed between multiple parallel processes.
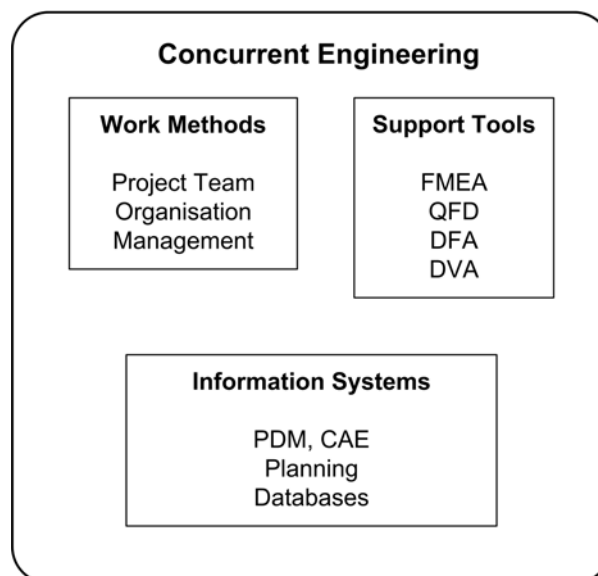
Figure 2-3 Concurrent engineering functional domains (Norell 1996)

Norell (1998) identified the three main functional requirements for concurrent engineering as being;

- Organisation and management supporting an integrated method of working

- The use of efficient support tools and methods in product development

- The use of relevant information transfer systems and tools

Curtis (2002) states that the concept of early and ongoing cooperation and information sharing between functional groups is central to the concurrent engineering process

2.3   DESIGN FOR X (DFX)

DFX is a generic term for a range of design tools that aim to optimise a particular facet of a product design as part of a concurrent engineering approach (Huang, 1996). The range of DFX tools includes amongst others;

DFA	Design for assembly
DFM	Design for manufacture
DFR	Design for reliability
DFS	Design for serviceability
DDC	Design for dimensional control

Care must be exercised when employing these tools to ensure that a balanced approach is used. Failure to do so may result in one facet of the overall design being over optimised to the detriment of the remaining design facets and the overall development of the product. This is one area where cooperation and information exchange within a concurrent engineering approach is essential. Leaney (1996) states that DDC has an important role in robust design as a cornerstone for linking related design tools (e.g. SPC and DFA) and that many commercially available DDC tools address the analysis or simulation of dimensional variation in the assembly process. The managerial and organisational features of DDC have largely been absorbed into the dimensional management process while the analysis and simulation aspects have evolved into the DVA process.

2.4 HISTORY OF DIMENSIONAL MANAGEMENT

The introduction of the Whitworth, later to become British Standard Whitworth, thread form (BSI, 2007) in 1841 realised the concept of standardised, universally interchangeable parts. This had a significant effect on the manufacturing process not least because it introduced new standards of measurement and accuracy. Despite this moves to control dimensional variation in the manufacturing process only came into being with the quality control movement of the 1920's-1930's. Work by Walter Shewhart (1925, 1926, 1927 1931) established the concept of modern quality control as applied to manufacturing. Yet despite these advances, during the 1930's most British engine manufacturers still used fully trained skilled labour to perform machining operations and engine fitting. The engine fitting was carried out by small teams of fitters responsible for each engine rather than on an engine production line (Robinson, 1979). Quality control was usually accomplished by testing and fixing each engine prior to dispatch

and was the responsibility of the respective charge hand or foreman. The Second World War had a profound effect on manufacturing particularly in the United Kingdom. High scrap rates due to poor assembly were no longer acceptable. The problem was attributed, in part, to the lack of full and complete information on the engineering drawings and fundamental weaknesses in the plus-minus system of co-ordinate tolerancing (Krulikowski, 2007). A solution was developed by Stanley Parker of the Royal Torpedo Factory who created the concept of true position tolerancing (HMSO, 1948).

In the post war period the work of W. Edwards Deming a former apprentice of Shewhart revolutionised the management of quality control and played a significant part in the Japanese industrial renaissance. Deming's work supplied the philosophy and management structures necessary to maximise the benefits of quality control (Deming, 1975). The resulting high quality of Japanese products posed a threat to American manufacturers (Deming, 1985) and lead directly to the development and launch by Motorola of their "Six Sigma Quality Program" in 1987 (Smith, 1993). The dramatic improvements in product quality achieved by Motorola stimulated interest in the quality control movement and particularly statistical techniques. One of these statistical techniques was dimensional variation analysis, developed by Ford and General Motors in the late 1970's DVA software became available in the early 1980's. Thus by the time Motorola introduced their Six Sigma process most of the modern concepts of quality control were in place. These concepts, DVA, GD&T, SPC , Six Sigma and others have evolved into a new engineering process known as dimensional management (DM) (Craig, 1997). The aim of DM is to safeguard the dimensional integrity of a product from initial concept, through design and product development and into production (Sleath, 1998). A good dimensional management system provides the structure and organisation necessary to ensure a competitive and robust product. Curtis (2002) defines a robust product as one which

can tolerate significant production and assembly variation and still meet functional requirements. With regard to the product development process Leaney (1996) states that;

"The aim of DM is to produce a robust product and process design by identifying parameter values that make a product or process insensitive to the inherent variation encountered during the manufacturing process".

Inherent in the DM process is the systematic implementation of DM tools. These tools systematically define the design, production and inspection of a product and monitor the process so that the predetermined dimensional quality goals are met (Nickolaisen 1999). The purpose of DM, also known as dimensional variation management or dimensional engineering, is to improve first time quality, control costs and improve product performance DM also serves to raise awareness, preventing variation from being overlooked during the early design stages.

There are a wide range of DM tools available today that cover nearly every aspect of the manufacturing process. However, DM relies on the capability to simulate, analyse and predict the effects of component and assembly variation on the complete system. This capability is DVA and it is a core element of DM.

2.5 DIMENSIONAL VARIATION ANALYSIS

Dimensional variation analysis (DVA) software first became available during the late 1970's and early 1980's. The DVA software was capable of simulating the accumulation of minor variations in the size, shape and location of the component parts throughout the assembly

process in order to predict the subsequent overall variation in the dimensions of the complete product.

Engineers at Ford and General Motors made good use of DVA during the development of new vehicle systems to analyse the probable variation behaviour of the system. As a core element of an overall DM process, DVA offered considerable advantages over the tolerance stack methods used previously. DVA provides a systematic and objective framework to minimise the risk that minor variations in the manufacture and assembly of the components could combine to produce a significantly greater overall variation that compromises the assembly process or the product quality. It became possible to identify potential variation problems well in advance, during the design stage, when it is still time and cost effective to modify the design or devise effective control measures and thereby deliver a far more robust system design and production process.

Since the late 1980's there has been a continuous drive to improve product quality by developments in manufacturing methods and the materials used in the manufactured products. Modern manufacturing techniques allow higher quality products to be manufactured with lower failure rates. However, Linares et. al (2007) show that while the failure rate is reduced the failure mode, in some manufacturing sectors, has changed significantly.
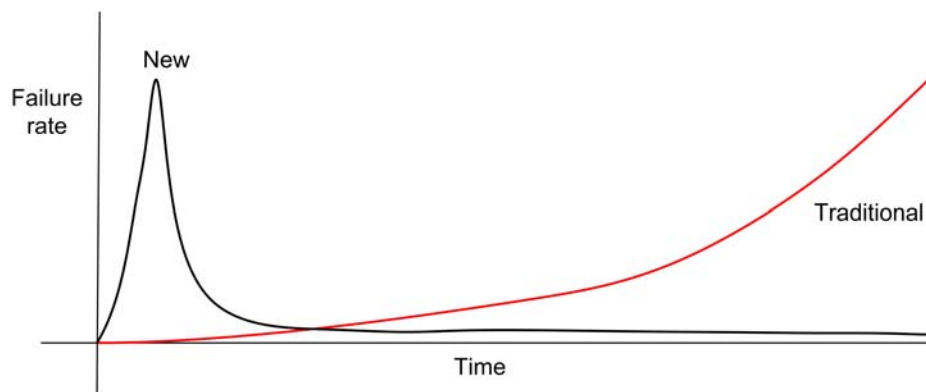
Figure 2-4 Change in failure mode according to Linares et al (2007)

In modern manufacturing, the reduction in working clearances decreases the amount of macro geometrical compensation available and the use of more wear resistant materials reduces or eliminates entirely any micro geometrical accommodation. Where such conditions apply components are at the greatest risk of failure early in their service life. The probability of failure then decreases dramatically and remains low for the remainder of the service life. This is the opposite of the traditional failure mode where the risk of failure increased with time, peaking near the end of the service life. The perceived quality of a product that fails after several years of service albeit prematurely will always be higher than one that fails quickly during the warranty period even if the product, when repaired, goes on to give a much longer service life. Thus where such a change in failure mode has occurred robust product design becomes even more important as a means of limiting warranty claims and maintaining customer perceived quality and satisfaction.

Today, DVA is widely practised in the UK automotive industry and to a lesser extent in the aerospace and other manufacturing industries. At Ford and Jaguar/Landrover, DVA is now a prerequisite of all new vehicle programmes. The automotive industry has exerted a powerful influence on the development of DVA. As a result methods have been developed to allow the analysis of non-rigid systems (Mortensen, 2002. Lee et. al., 2007), such as sheet metal assemblies or multistage manufacturing processes (Shi, 2007). These methods, however, fall outside the sponsoring company's sphere of interest and are thus beyond the scope of the current project.

2.6 METHODS TO DEFINE, EXPRESS AND COMMUNICATE COMPONENT VARIATION

Prior to the advent of modern mass production methods there was little need to consider component variation. Component parts were individually fitted to the assembly and adjusted where necessary. Mass production required interchangeable component parts and thus means of defining and communicating the extent of component variation were required

2.6.1 TRADITIONAL METHOD

The traditional method of communicating dimensional variation information by means of co ordinate dimensional tolerances made its first appearance in 1927 when the British standard for the fit of holes and shafts was introduced. The use of dimensional tolerances to represent the acceptable variation of most machined surfaces was widespread by the late 1930's. The amount of information conveyed by these dimensions is, however, limited and open to interpretation.
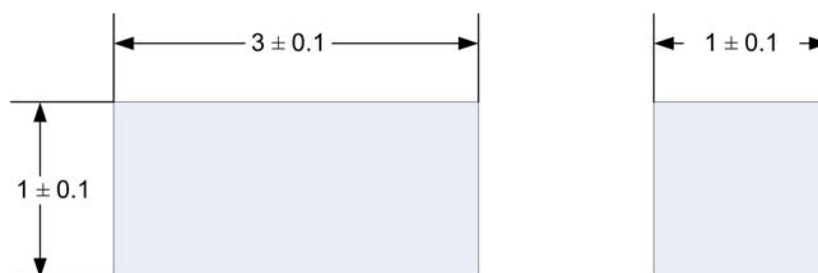
Figure 2-5 Dimensional tolerances

Figure 2-5 shows a block and its associated dimensional tolerances; the design intent is that adjoining faces should be perpendicular and opposing faces parallel. However, much of this detail is reliant on the interpretation of the drawings by the person manufacturing the block.

The dimensional tolerances give little information on how the block should be measured to check dimensional compliance. One major draw back of this method is that it produces square or rectangular tolerance zones (Figure 2-6). These allow more variation along the diagonal of the tolerance zone than parallel to the sides of the zone. It was to address some of these deficiencies that Stanley Parker (HMSO, 1948) developed the concept of true position tolerances. This work lead ultimately to the development of geometric dimensions and tolerances.
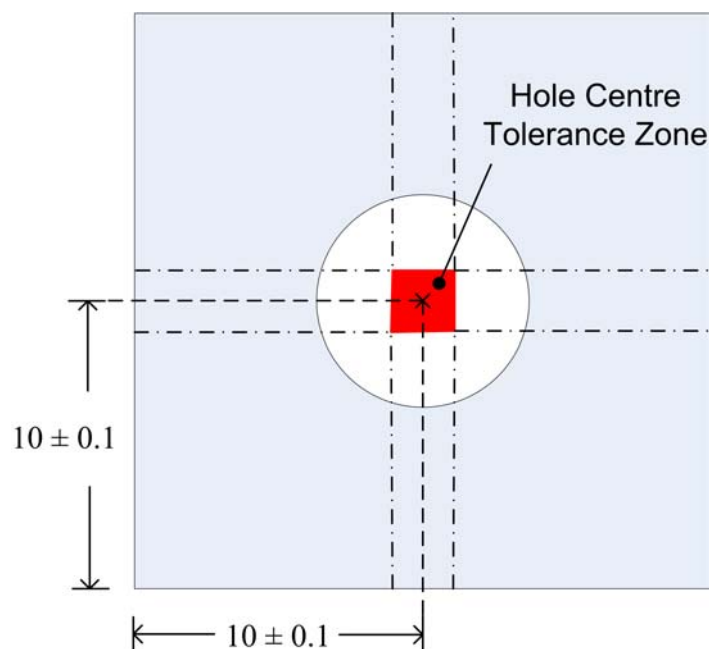
Figure 2-6 Dimensional tolerance zone

2.6.2 GEOMETRIC DIMENSIONS AND TOLERANCES (GD&T)

GD&T is conceptually a development of the traditional Go, No-Go gauges (Whitney, 2004) in that it defines two 3D surfaces one of which represents the minimum acceptable size of the component. The second represents the maximum acceptable size. When the geometric centres

of the two surfaces are co-incident any component, in this instance a cube, which falls entirely within the zone between the two surfaces (Figure 2-7), is deemed to be of acceptable size and shape.
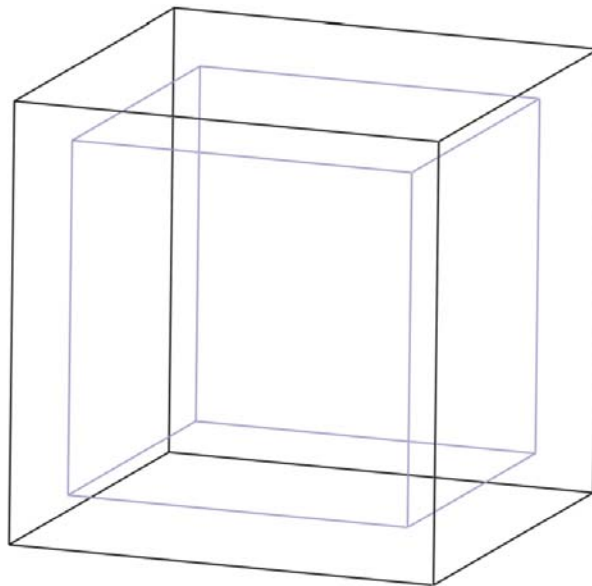
Figure 2-7 Minimum and maximum size surfaces

The size, shape and orientation of the two surfaces will vary depending on the particular tolerance being defined and is specified in several national and international standards such as ASME Y14.5M, BSI 8888 and ISO 1101. There is a continuing international effort (ASME, 2009, Krulikowski & DeRaad, 1999) to harmonise these standards towards a single international standard for GD&T. At present the standards are very similar in content but subtle differences do exist between them. To accurately interpret GD&T information, and in particular legacy information, it is still necessary to know which standard, and which version of that standard, were used to create the specifications.

Most DVA software originates from the USA and uses the American Y14.5M standard. Originally an American military standard ASME Y14.5M, defines geometric dimensions and

tolerances in mathematical terms (Nasson, 1999). It conveys both the nominal dimensions (ideal geometry) and the acceptable variation (tolerance) from the nominal with regard to the size, shape and location of the part (Stites & Drake 1999). The latest version of ASME Y 14.5M contains a dimension origin symbol to identify the feature from which the measurement originates (ASME, 2009) this may prove useful to indicate a specific measurement plan. Equally many tolerances but by no means all have associated datum features. These datum features and the order in which they are specified should allow the metrologist to specify an appropriate measurement scheme for the tolerance in question. The Y14.5-M standard identifies which degrees of freedom are constrained by different types of primary datum feature but there is no mechanism for communicating any constraint information via the feature control frame. The feature control frame (Figure 2-8) is one of the great strengths of GD&T. It is capable of communicating a considerable amount of complex information in a simple unambiguous manner.
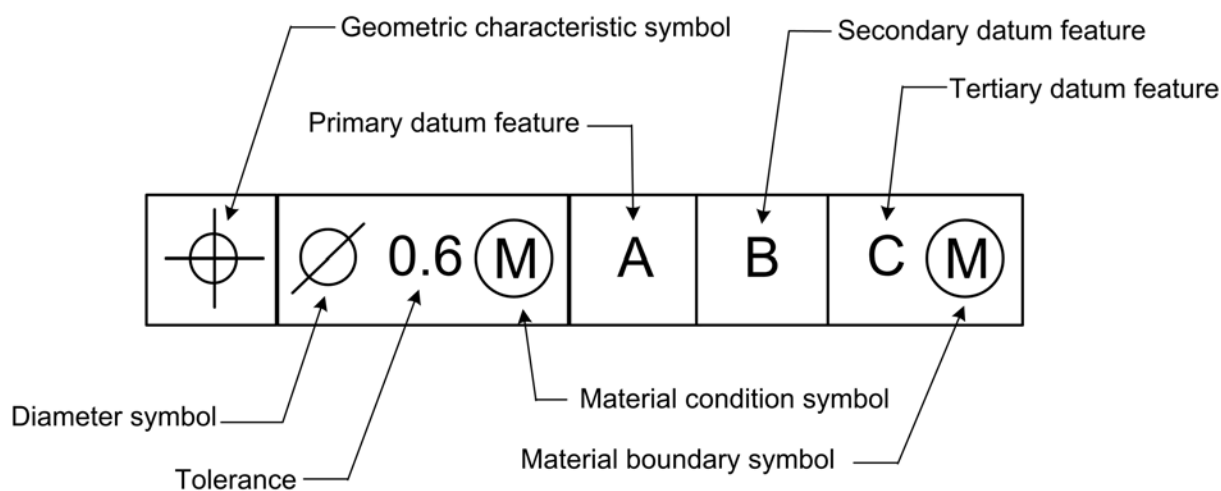
Figure 2-8 Feature control frame

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2.7 STATISTICAL PROCESS CONTROL (SPC)

There are a myriad of SPC methods in use. One common feature is that they all utilise production data, either real or simulated. Some of the more widely used methods are;

2.7.1 VARIATION DISTRIBUTIONS

The process output is sampled and the results plotted as a histogram, to which a distribution curve is often fitted to allow for the discrete nature of the sample data. From either of these distributions a variety of statistical parameters can be derived as can be seen in Figure 2-9.

Nominal	160.000	Lower Spec Limit	159.000	Cp	0.7812
Mean	160.026	Upper Spec Limit	161.000	Cpk	0.7606
Std Deviation	0.4267				

	Sample	Estimate		Sample	Estimate
% < Low Limit	0.7467	0.8077	Low	158.1348	158.7463
% > High Limit	1.0933	1.1250	High	161.8878	161.3055
% Out of Spec	1.8400	1.9327	Range	3.7530	2.5602
	± 3 Sigma Range	99.7300%			
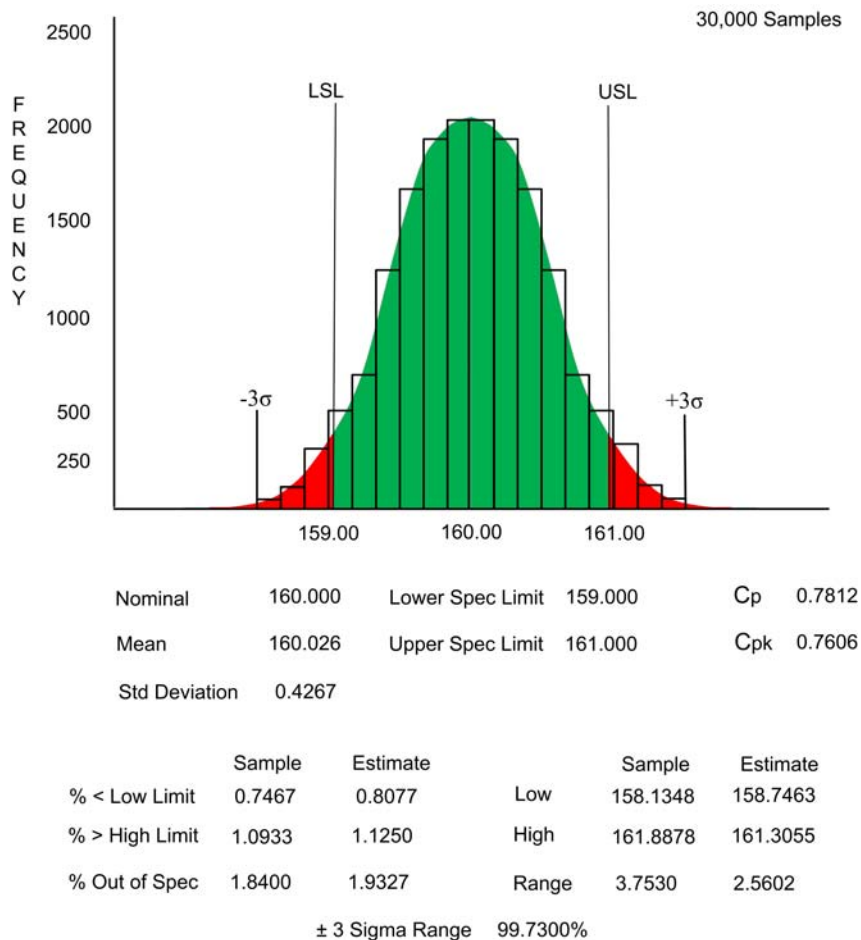
Figure 2-9 Variation distribution and typical derived data (VisVSA, 2001)

2.7.2 PROCESS CAPABILITY

The process capability index is a comparison of the inherent variation in a dimension against the tolerance limits applied to the dimension. It shows how well the variation range fits within the tolerance limits. It is defined as;

$$C_p = \frac{U - L}{6\sigma}$$
Equation 2-1

Where U is the upper specification limit

L is the lower specification limit

σ is the standard deviation of the dimension

This does not however take into account that the dimension variation distribution may not be centred on the nominal dimension. A second capability index, Cpk, is often used which allows for this. Cpk is defined as the smaller of the two indices Cpl and Cpu where;

$$C_{pl} = \frac{\mu - L}{3\sigma}$$
Equation 2-2

$$C_{pu} = \frac{U - \mu}{3\sigma}$$
Equation 2-3

Where μ is the distribution mean.

Cpk is therefore a measure of both the spread and location of the distribution. The capability of the process is indicated by the value of Cpk;

- Cpk < 1, The process is not capable, non-conforming output is inevitable

- Cpk = 1, The process is minimally capable but any changes may result in non-conforming output.

- Cpk > 1, The process is capable.

2.7.3 SHEWHART CONTROL CHARTS

Murdoch (1979) describes a variety of Shewhart control charts two of which are shown below (Figure 2-10). In process control, small test samples are taken at intervals from the production run. The mean and range of these samples will vary about the underlying mean value of the production run.
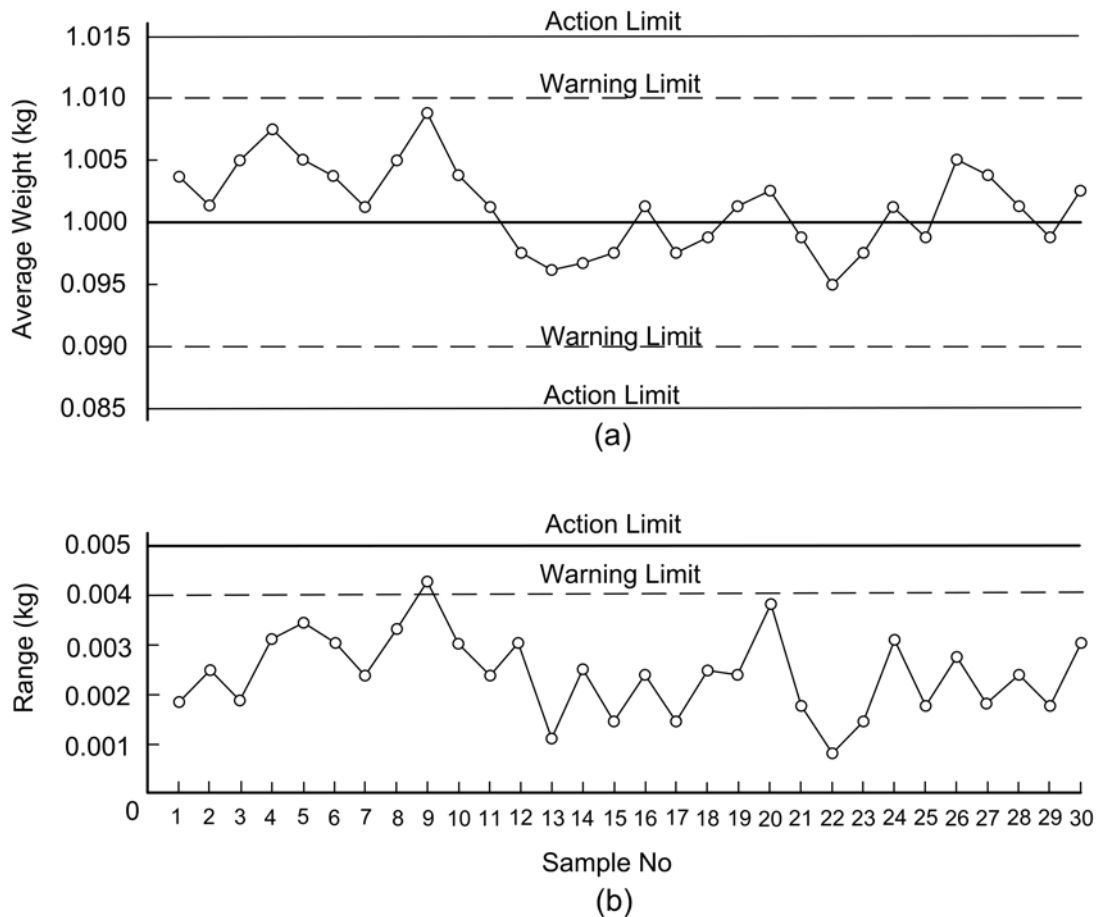
Figure 2-10 Shewhart control charts (a) Process average chart (b) Range chart.

(after Murdoch, 1979)

The question is, is this underlying mean value stable or does the variation indicate changes in the process, which should be corrected. While the sample mean and range remain within the warning limits on the charts, it is probable that the underlying mean value is stable. In a stable system the probability of exceeding the warning limit is low; if it occurs it may indicate the system may be becoming unstable, particularly so if successive sample means breach the limit. If the action limit is exceeded there is a high probability that the process has become unstable and the underlying mean value has changed.

2.8 METHODS USED TO GRAPHICALLY DEFINE, EXPRESS AND COMMUNICATE ASSEMBLY INFORMATION

The assembly process is an integral part of DVA. Assembly provides the means by which component part variation can propagate throughout the system. It also introduces a further source of variation into the system as location of a part in the assembly can vary. This variation can take the form of changes in the position or orientation of a component part relative to other parts in the assembly. The sequence in which the assembly operations are performed will have a direct bearing on the manner in which variation propagates through the assembly. An understanding of the various methods used to document the assembly process is therefore essential to understand how variation propagates in the assembly process.

2.8.1 **ASSEMBLY SEQUENCE ANALYSIS**

The overall aim of assembly sequence analysis is to generate one or more feasible assembly sequences for a product. Whitney (2004) assigns five stages to the process (Figure 2-11).
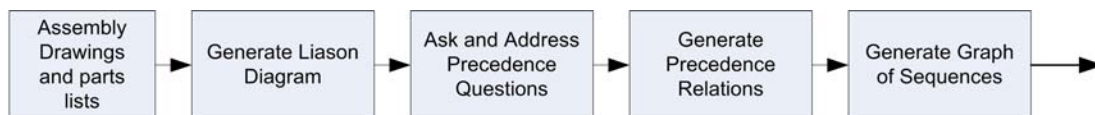
Figure 2-11 Flowchart for generating feasible assembly sequences. (Whitney, 2004)

2.8.1.1 Assembly drawings and parts lists

These provide the basic information to construct the liaison diagram. The detail need only be sufficient to establish pairs of mating parts and any assembly operation precedence issues. The assembly sequences generated can be re-assessed as more design information becomes available. It does, however allow the process to commence early in the design process when it is still cost effective to design out any assembly sequence problems.

2.8.1.2 Generate liaison diagram

The liaison diagram establishes which parts mate and which do not. Consider the roller towel in Figure 2-12 the liaison diagram can be seen in Figure 2-13. In constructing the diagram it is assumed that all the parts are rigid and that once a liaison is made it remains made. To insure efficient operation of the various algorithms applied to liaison diagrams, they must comply with the loop closure rule. This rule first demonstrated by Bourjault (1984), states that for any loop in a liaison diagram if at some point in the assembly process a loop of n liaisons

stands with n - 2 liaisons already made, then the next step applied to that loop shall close both the remaining open liaisons. Consider components A, B and C in Figure 2-13, if part B is already assembled to part A (liaison 1) when part C is added to the assembly it must close liaisons 2 and 4.
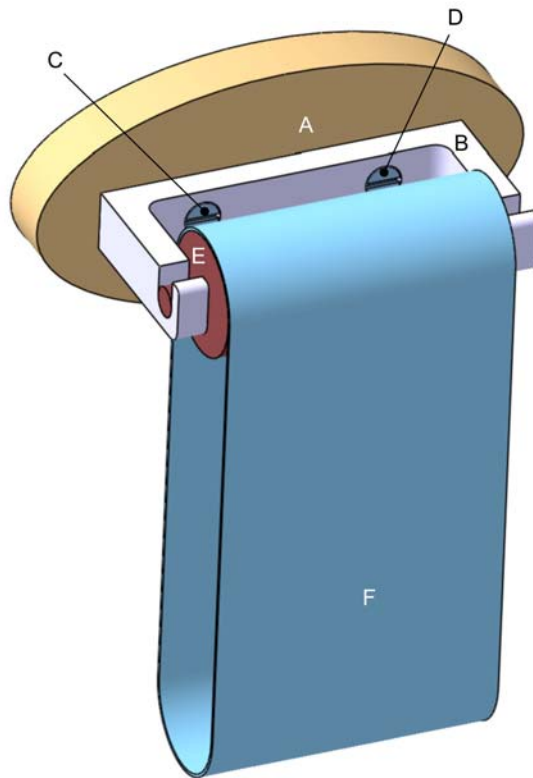
Figure 2-12 Roller towel assembly
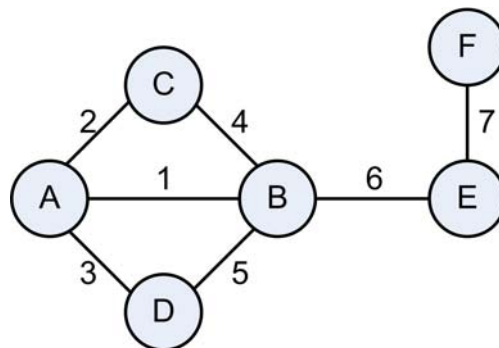
Figure 2-13 Roller towel liaison diagram

Each liaison in the diagram represents the sum of all the constraints between the two parts thus liaison 4 may represent both a face to face and a pin in hole contact between parts B and C. This attribute may limit the functionality of liaison diagrams in other applications.

2.8.1.3 Ask and address precedence questions

The precedence questions are designed to determine if there are any local precedence requirements in the assembly process. In the case of the roller towel example bracket B must be placed on the mounting plate A (liaison 1) before the two screws (C & D) are inserted (liaisons 2, 3, 4, 5). The towel (F) is placed on roller E (liaison 7) at anytime prior to its assembly to the bracket (liaison 6). This can only take place after the screws have been inserted as the roller blocks access to the screw holes.

2.8.1.4 Generate precedence relations

The precedence relations can be expressed graphically (Figure 2-14). From the above description it is clear that liaison 6 must be preceded by liaisons 2, 3, 4, 5 and 7. Liaisons 1 and 7 are unprecedented, and therefore, the assembly could begin with either of these liaisons.
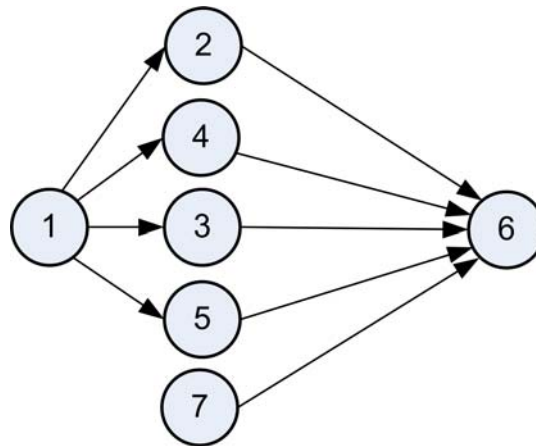
Figure 2-14 Roller towel precedence relations

2.8.1.5 Generate graph of sequences

All the possible assembly sequences are determined by considering the liaison diagram with respect to any precedence requirements. In the case of the roller towel example the loop closure rule requires that liaisons 2 and 4 be closed in the same step, similarly liaisons 3 and 5. Construction of the liaison sequence graph (Figure 2-15) commences at the top of the graph with the blank squares. These represent the liaisons which still have to be made in the assembly. The individual liaisons are identified by the liaison key. As each liaison is made the representative square is filled in.
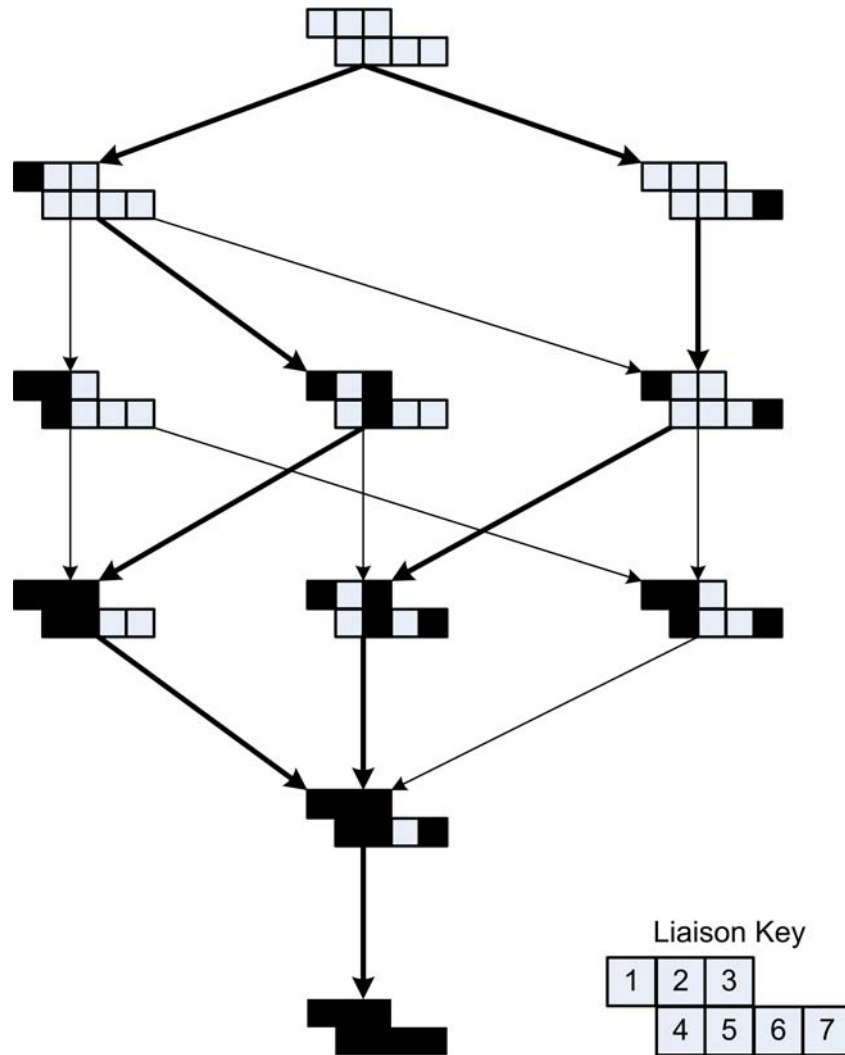
Figure 2-15 Roller towel liaison sequence graph; the bold lines represent two possible assembly sequences. The filled in squares represent completed liaisons (after Bourjault, 1984)

In the example the only unprecedented liaisons are numbers one and seven; these are represented by the two steps in the second level of the graph. The next level in the graph shows the second liaison to be made in the assembly sequences. These are liaisons two and four, three and five, which must be completed as a pair in the same assembly step and liaison one. At this stage there are four assembly sequences. The next level represents the third liaison to be made. The number of assembly sequences has now increased to seven. All the sequences converge in the next level where liaison six is the only liaison that has not been

made. This is because it must be preceded by all the other liaisons in the assembly. The final level in the graph represents the complete assembly with all liaisons made.

The broad lines in Figure 2-15 represent two of the possible, if not necessarily practical, assembly sequences. Each unique route from the top to the bottom of the diagram represents a possible assembly sequence. This technique is useful in that it can generate all the possible assembly sequences sufficiently early in the product development process that the assembly sequence of choice can be subjected to DVA. Should the initial sequence prove unsuitable due to variation propagation or other issues, it is still early enough in the product development process that an alternative sequences can be selected. Assembly sequence analysis has been computerised (Baldwin et al, 1991) but still requires a certain amount of human input.

2.8.2 ANNOTATED LIAISON DIAGRAMS

The concepts of liaison diagrams and interface control have been expanded upon in more recent work by Falgarone and Chevassus (2006) and Ballu et al (2006). Their technique GASAP (Geometric As Soon As Possible) utilises the principles of liaison diagrams, key characteristics and datum flow chains. Lee and Thornton (1996) define a key characteristic as a product characteristic for which reasonably anticipated variation could significantly affect the products safety, or customer satisfaction with the product. A datum flow chain (Mantripragada & Whitney, 1998) is the chain of component part features which links the two ends of a key characteristic. They allow the relationship between a product key characteristic and the component part features that constrain it to be established. The technique consists of mapping the relationship (datum flow chain) between assembly feature interfaces and key characteristics of the assembly.
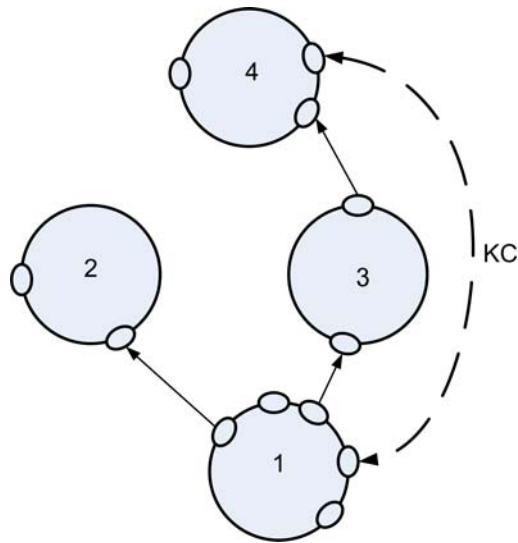
Figure 2-16 Annotated liaison diagram; the numbered circles represent component parts, the elliptical ears represent assembly features. (Mathieu & Marguet, 2001)
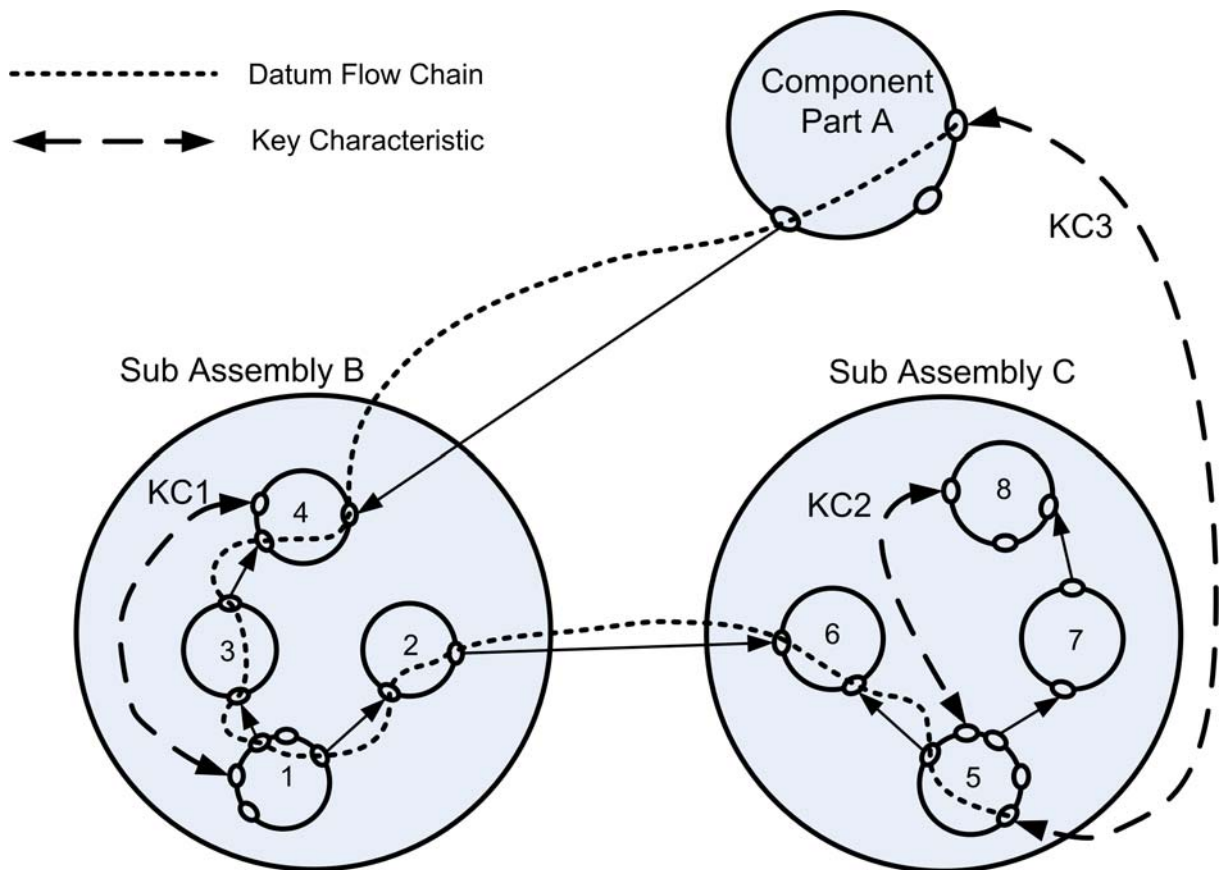
Figure 2-17 Assembly nested liaison diagram showing multiple key characteristics,

sub assemblies and a datum flow chain (after Falgarone & Chevassus, 2006)

At its most basic level this is achieved by means of an annotated liaison diagram (Figure 2-16) which incorporates part features, key characteristics and to a certain extent liaison precedence requirements. In the diagram the parts are represented by the large circles, the part features by the small elliptical ears and the precedence requirements by the direction of the arrows on each liaison. The key characteristic (KC) is represented by the dotted line between two part features. The diagram can be expanded to incorporate sub-assemblies and datum flow chains (Figure 2-17). The dotted line indicates the datum flow chain for key characteristic three (KC3). The precedence requirements are retained and the diagram shows that the assembly order is part A, sub-assembly B and sub-assembly C.

The GASAP technique, used in conjunction with the modelling tool GAIA (Falgarone & Chevassus, 2006), also allows the modelling of functional and kinematic attributes of the product which can be broken down to constraints and parameters in later stages of the process. This provides a link, based on functional analysis, between the CAD system and the conceptual design (Ballu et al 2006). The interesting feature is that the computer design tool GAIA, which supports the technique of Falgarone, Chevassus and Ballu can be used in conjunction with Catia CAD software, as can Cetol 6 Sigma. It may therefore be possible to link or integrate the two systems and produce more open system architectures.

2.9   METHODS OF SIMULATING VARIATION

Some of the more commonly used techniques to simulate the propagation of variation include;

2.9.1 TOLERANCE STACK UP

Tolerance stack ups are a long established method of simulating variation accumulation. Traditionally performed as a manual, paper based simulation, in 1D and to a lesser extent 2D. They have largely been superseded by 3D computer simulation techniques. The basic technique consists of defining a tolerance chain for the dimension of interest (Figure 2-18). Once the tolerance chain has been identified it can then be analysed using one of the various tolerance accumulation models such as worst case or root sum square.

2.9.2 SPREAD SHEETS

Computerised spreadsheets provide a convenient interface for the calculations necessary to analyse 1D tolerance stack-ups. Typically they would provide worst case, statistical or root sum square and six sigma analyses. In the case of a 1D analysis the typical input would be a mean and a three sigma variation for each dimension. The higher order dimension analyses are more difficult to perform as it is necessary to include a sensitivity index into the calculation to allow for the individual variations having a greater or lesser effect in each of the three dimensions. 3D tolerance stack-ups have largely been superseded by 3D computer simulation which automatically calculates the sensitivity indices.
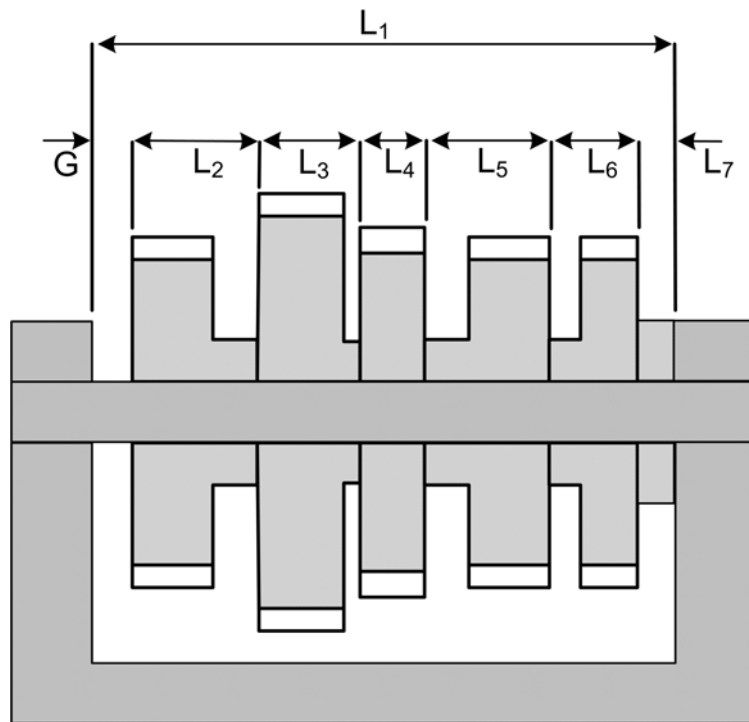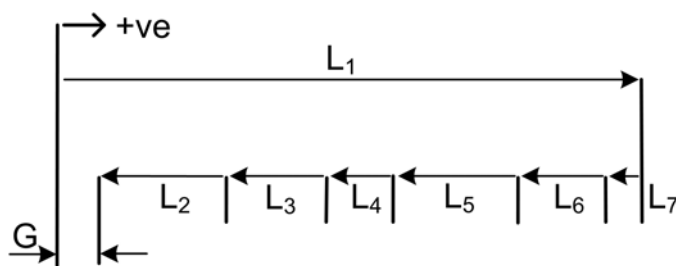
Figure 2-18 One dimensional tolerance stack up (Scholz, 1995)

2.9.3 COMPUTER SIMULATION (3D)

The basis of computer simulation is the DVA model. This consists of the nominal CAD geometry, which is overlaid with selected assembly features (Sigmetrix, 2006, VisVSA,

2001) to which variation is applied in the form of tolerances. The assembly operation between adjacent assembly features is analogous to a single link in a DFC. The overall effect of the DVA model is to provide a physical link through adjacent component parts between the two ends of each key characteristic or other system attribute under investigation. The DVA model thus defines the assembly function graphically and the extent of the variation in that function. It has the advantage of simplifying the assembly function as only those features that affect the key assembly characteristics under consideration are included in the feature overlay. This combined with an appropriate tolerance accumulation model allows the propagation of variation and its effect on the assembly to be simulated.

In a rigid mechanical system variation propagation is an additive process. There are three sources of variation that have to be taken into account, size, shape and assembly process variation. Assembly variation consists of the small kinematic adjustments to the location and orientation of the component parts which are dependent on size and form variation (Chase et al 1994). Figure 2-19a shows how the location U of the circle centre, an assembly variable, adjusts to accommodate the dimensional variation in the radius of the circle R. In this particular example the relationship is explicit and can be expressed as;

$$U = \frac{R\left(1 + \dfrac{1}{Cos\,\vartheta}\right) - A}{Tan\,\vartheta} \qquad \textbf{Equation} \qquad \textbf{2-4}$$

It will, however be noted that if θ varies then the resultant variation in U will be non-linear despite the simplicity of the assembly.

a) Kinematic adjustment in U due to
   dimensional variation in R

b) Kinematic adjustment due to
   form variation
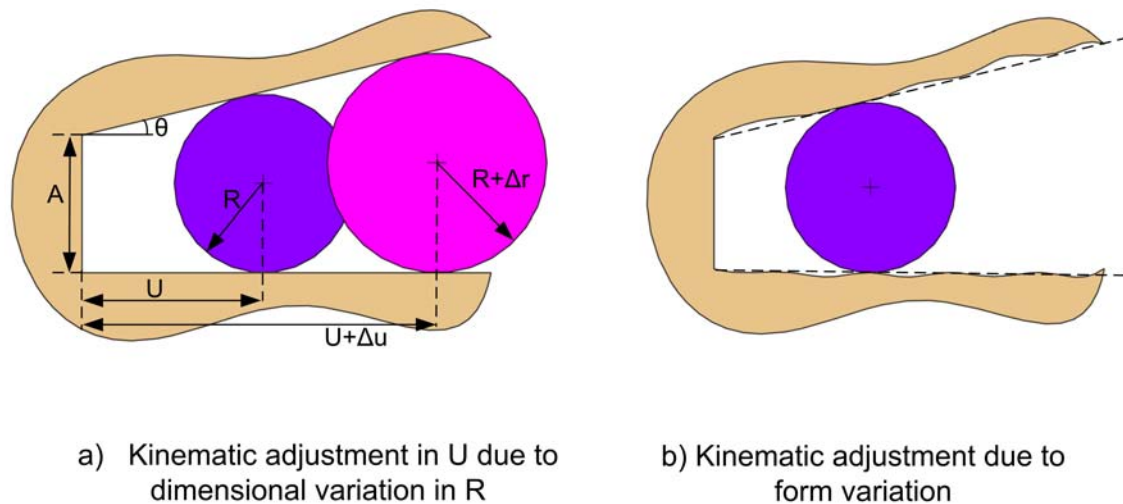
Figure 2-19 Kinematic adjustment due to variation; illustrated by a circle in groove system
(after Chase et al 1994)

A similar effect can be produced by form variations (Figure 2-19b); in this instance the exact

position of the circle centre will depend on the pattern of the surface waviness of the groove

which will vary from one part to the next. This is significant in that the relationship between

the variables is now implicit. Not all modelling techniques can accommodate the implicit

equations which may be required to resolve small kinematic adjustments.

2.9.3.1 Vector loop simulation

When using vector loop simulation, the DVA model consists of kinematic joints at the mating

interfaces of the assembly linked by vectors which represent the component dimension

between the mating interfaces. The vector loop can either be closed (Figure 2-20b), in which

case it finishes at exactly the same point from which it originated, or open in which case the

loop starts at one side of a gap and finishes at the other side. The variation in a product key

characteristic is calculated using an open loop, with the vector matrix of the key characteristic closing the loop, while closed loops can be used to determine kinematic variation. It is, however, a necessary condition for analysis that all the vector loops in the DVA model are closed and the overall system kinematically constrained. In the case of Figure 2-20a it would be necessary to constrain the rotation of the crank before analysis was possible.
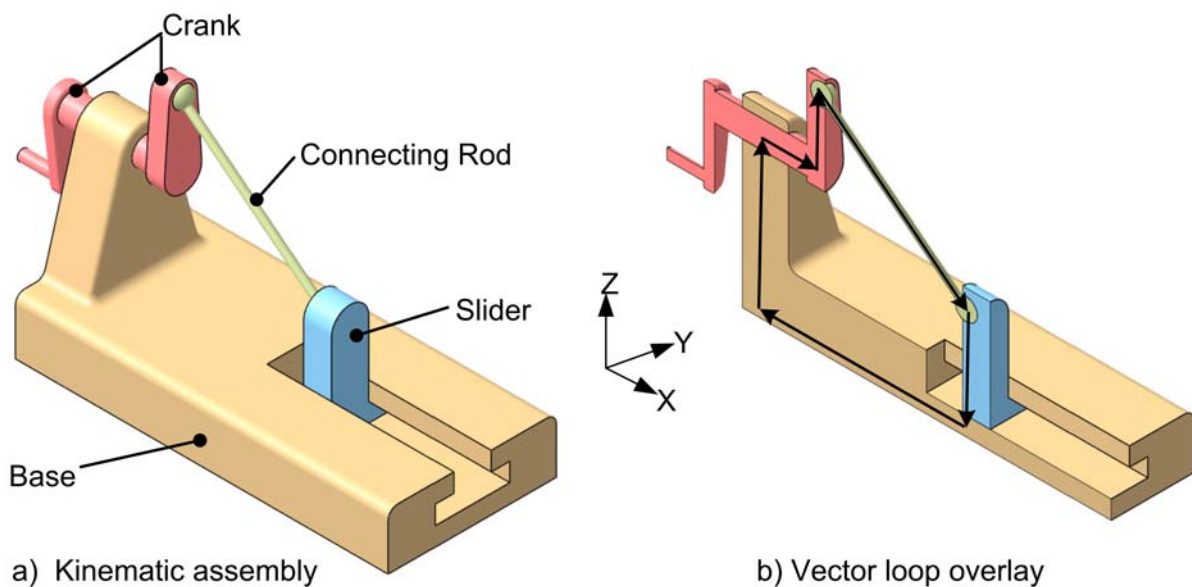
Figure 2-20 Kinematic assembly and vector loop overlay (after Goa et al 1998)

The advantage of vector loop analysis is that if arrangements were made to constrain the crank in three or four different positions then the remaining components in the assembly would automatically adjust to each different crank position. This is because when two component parts are mated they remain free to move relative to each other in any unconstrained degree of freedom. It is also the reason the overall assembly must be kinematically constrained as otherwise the position of at least one of the component parts would be indeterminate preventing analysis.

2.9.3.2 Monte Carlo simulation

Monte Carlo simulation (MCS) provides a powerful tool for the analysis of mechanical assemblies. It has the advantage that it can be used for non-linear assembly functions and with non-normal input or output distributions. MCS is a sampling technique. A large array of sample parts are created by assigning, at random, a tolerance value of each nominal dimension which falls within the variation distribution of the dimension to simulate the effects of manufacturing variation on the component parts. The random parts are inserted in to the DVA model, which graphically defines the assembly function. The required output data is then measured directly from the model and stored. The process is repeated with different sets of random tolerance values until sufficient data has been generated (Figure 2-21). The output data is usually plotted as a series of histograms, one for each output measurement, (Figure 2-22). One advantage of MCS is that the number of assemblies which fall outside the assembly specification can be counted directly from the output sample or estimated from the distribution curve which is usually fitted to the sample histogram. Standard statistical techniques can be applied to the output distribution to determine parameters such as mean, standard deviation, range, Cp, Cpk and percentage of rejects. MCS is usually applied to an explicit function of random variables; however, kinematic adjustments due to geometric form variations are implicit. Gao et. al. (1995) proposed a modified form of MCS (McCATS) which would take into account the implicit assembly variations. In the modified simulation the random parts are sent to the assembly function which solves the non linear vector loop equations iteratively for the dependent assembly variations (Fig 2-23). The results are stored and the process repeated until a sample of suitable size has been created to produce the assembly histogram.
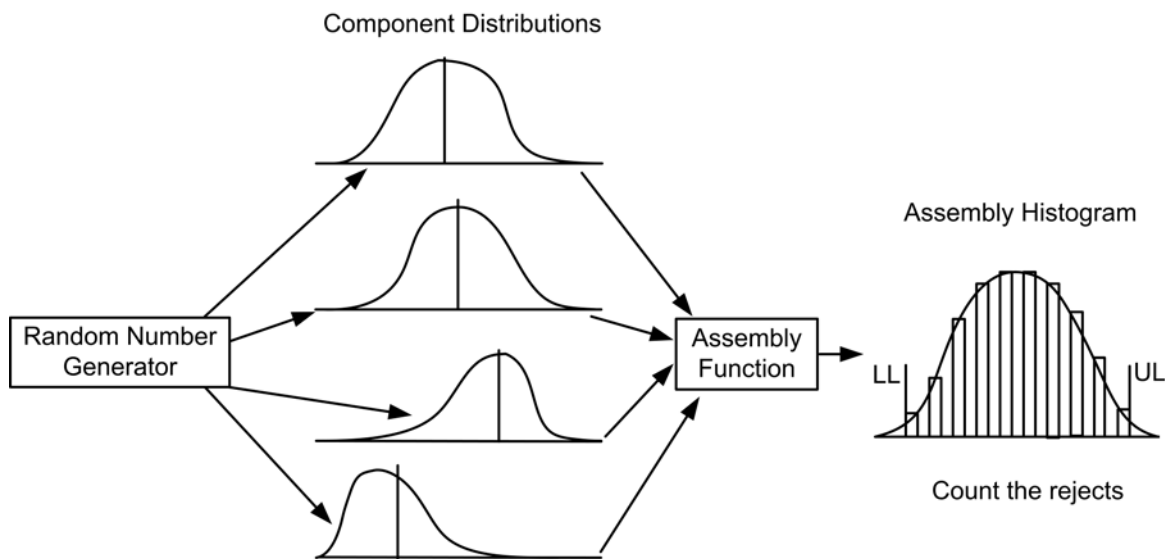
Figure 2-21 Tolerance analysis by Monte Carlo simulation (Chase & Parkinson 1991)

Nominal	160.000	Lower Spec Limit	159.000	Cp	0.7812
Mean	160.026	Upper Spec Limit	161.000	Cpk	0.7606
Std Deviation	0.4267	Distribution	Tested Normal		

	Sample	Estimate		Sample	Estimate
% < Low Limit	0.7467	0.8077	Low	158.1348	158.7463
% > High Limit	1.0933	1.1250	High	161.8878	161.3055
% Out of Spec	1.8400	1.9327	Range	3.7530	2.5602
			± 3 Sigma Range		99.7300%
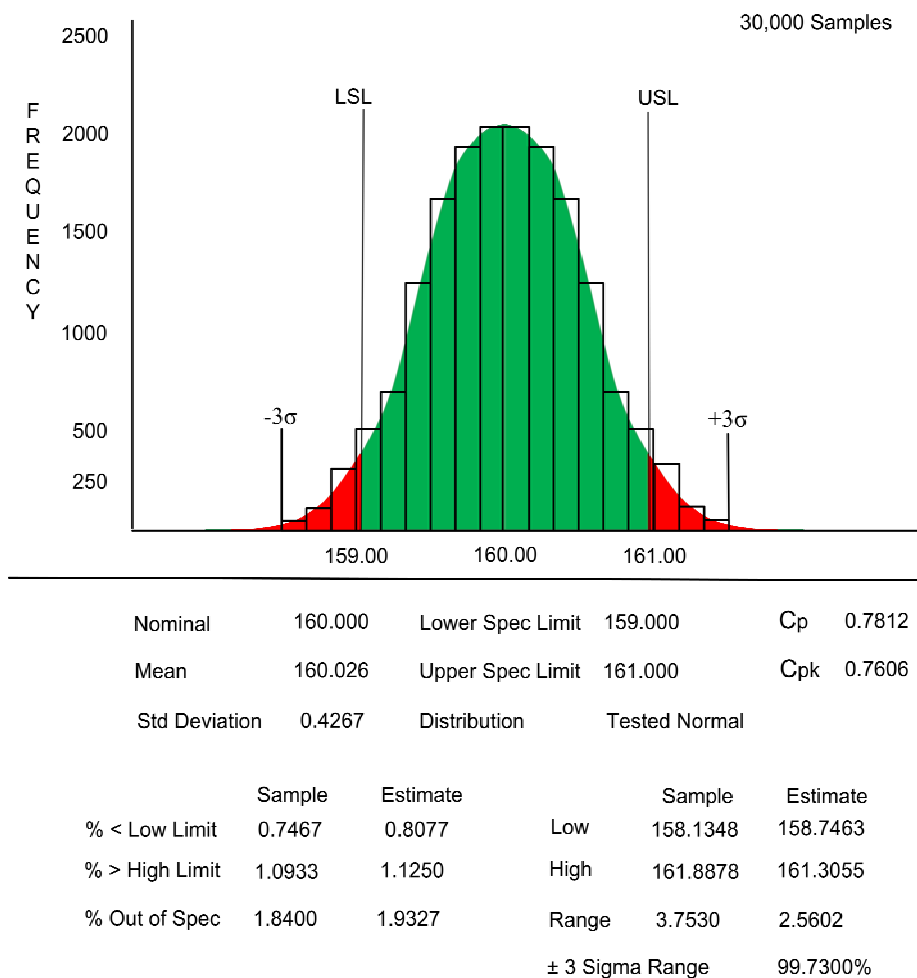
Figure 2-22 Monte Carlo simulation output (VisVSA, 2001)
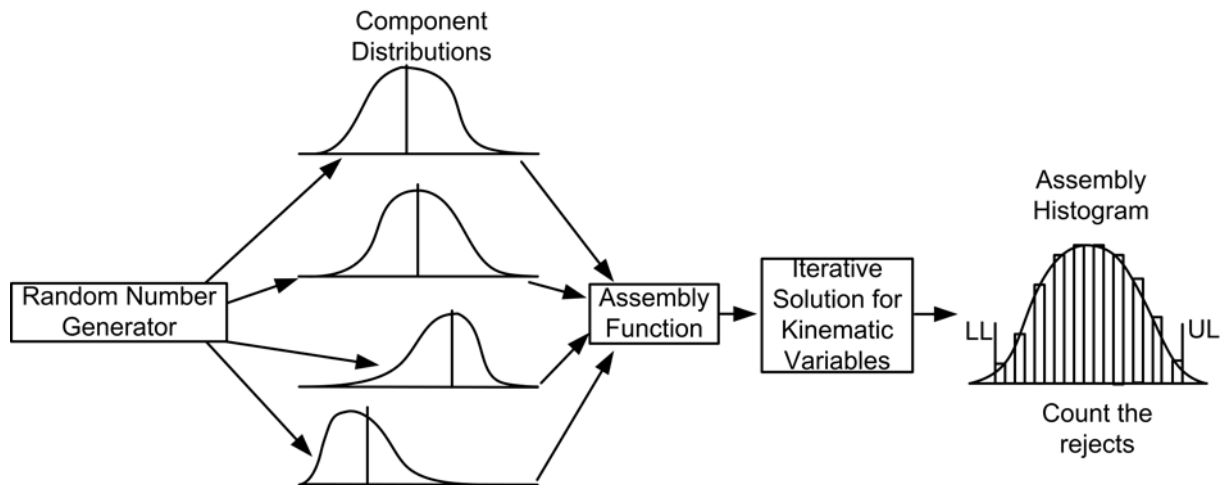
Figure 2-23 Monte Carlo simulation for implicit assembly constraints (Gao et al, 1995)

A major disadvantage of Monte Carlo analysis is that very large sample sizes are required to obtain an accurate result. Chase and Parkinson (1991) suggested that a sample size of 100,000 to 400,000 is necessary to accurately predict the small number of rejects produced by modern manufacturing techniques. This makes accurate MC analysis computationally expensive, especially as the entire simulation has to be repeated if any input variable is changed. In the case of the McCats simulation the computational burden is increased by a factor of approximately 3.4 due to the iterative process (Glancy & Chase, 1999). Carlson (2000) proposed the direct second order method (DSO) as a technique that eliminates the requirement for iterative solutions by providing the second order sensitivities in closed form. The DSO method obtains the second order sensitivities by exploiting the analogy between variation and kinematics. The vector loop assembly model is subjected to kinematic acceleration analysis and by combining various terms of the analysis the second order sensitivities can be obtained in closed form without iteration.

2.10 METHODS TO ANALYSE VARIATION BEHAVIOUR

Every product or assembly has a design intent or function that it should meet if it is to be viable. This intent or function is frequently expressed in terms of assembly level dimensions or key characteristics that are both important to product function and subject to the effects of variation. These assembly level dimensions or key characteristics form the basis of the dimensional variation analysis. It is the effect of component part variation on these measurements that is simulated in the dimensional variation analysis. Since the measurements play a central role in the analysis it is important to establish;

- Between which two component parts the measurement is to be made

- Between which features on the two component parts the measurement is to be made

- The direction in which the measurement is to be made

- That a competent datum flow chain (tolerance chain) exists between the two features at either end of the key characteristic

- The type of measurement to be made

2.10.1 SENSITIVITY ANALYSIS

The sensitivity analysis determines the sensitivity of an assembly level measurement to variation in each component part dimension in the assembly. The analysis consists of setting all the assembly variables to their nominal values except one. A unit variation is applied to this variable. The system is then assembled according to the assembly function and the measurement made on the assembly. The result is recorded and the process repeated for the remaining variables (Figure2-24). The sensitivity values produced can then be used in root cause analysis (see 2.10.2) or tolerance accumulation models (see 2.10.4)
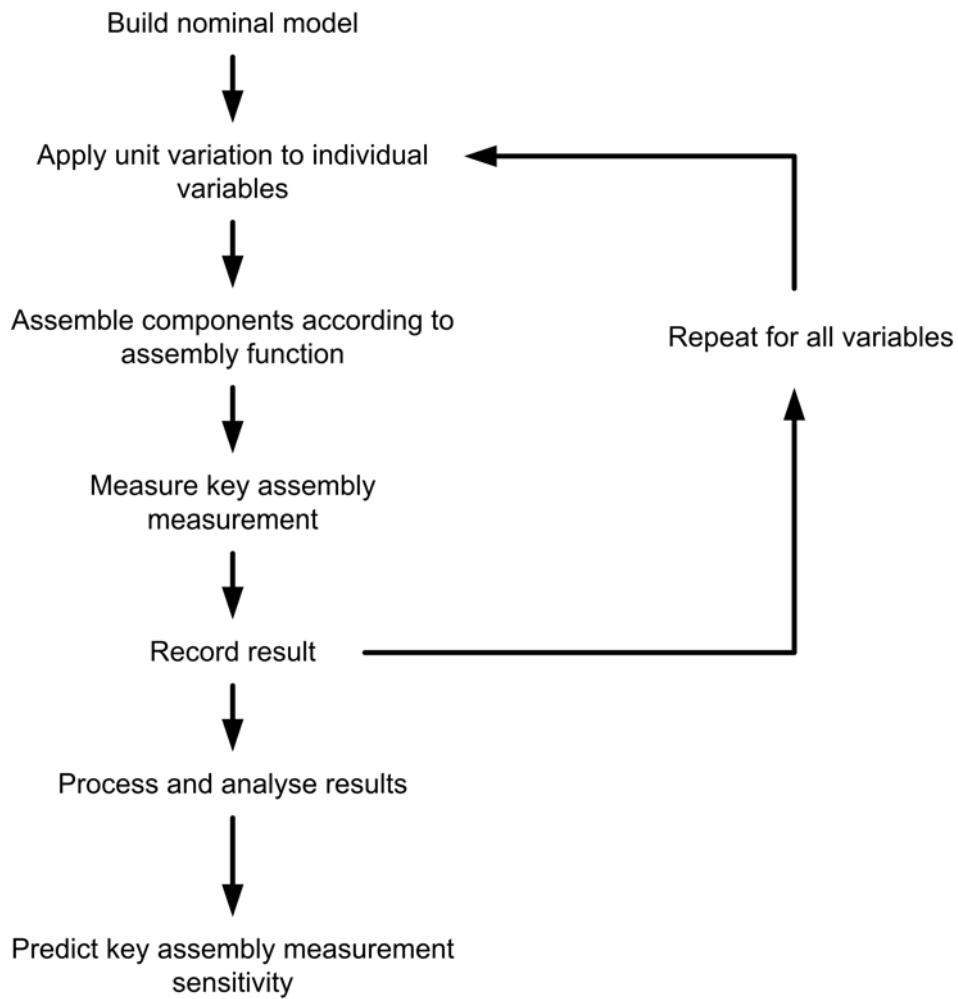
Figure 2-24 Sensitivity analysis flow chart (i-dmsolutons, 2008)

2.10.2 ROOT CAUSE ANALYSIS

Root cause analysis is also known as HLM or contributor analysis. The analysis consists of applying the high, low and median values to each of the assembly variables in a similar manner to that used in the sensitivity analysis. The variation contributed by an individual dimension to the overall assembly variation is defined (Chase, 2004) as;

$$\% \text{ Contribution} = \frac{\left(\frac{\partial U}{\partial Xi}\sigma_{Xi}\right)^2}{\sigma_U^2} \text{ x } 100 \qquad \text{Equation 2-5}$$

where $\frac{\partial U}{\partial X_i}$ is the sensitivity of the assembly to dimension X_i, σ_{Xi} is the standard deviation of

dimension X_i and σ_U is the standard deviation of the assembly variation.

The ability to identify the major contributor to the assembly variation is useful in design optimisation as it allows the effort to be concentrated on those dimensions which contribute the most to overall product variation. The analysis takes into account that a dimension may have a wide distribution but if the sensitivity is low the overall effect may be small. Equally a dimension with a very narrow distribution but a high sensitivity may produce a significant product variation.

2.10.3 VARIATION DISTRIBUTION

The variation distribution of a key product measurement is often used as the basis for analysing the extent and character of the variation by means of the various statistical process control methods (see section 2.1.7). The distribution curve parameters such as mean and standard deviation, in the case of a normal distribution, or mean, standard deviation, skew and kurtosis in the case of a generalised lambda distribution can also give an insight into the nature of the variation.

2.10.4 TOLERANCE ACCUMULATION MODELS

Commonly used models to analyse tolerance stack-up include worst case, root sum square and six sigma or mean shift. The models calculate the total cumulative effect of variation on the assembly level dimensions of a mechanical assembly. The equations for these models, for a 3D system, as described by Chase & Parkinson (1991) are given in equations 2.17 and 2.18

2.10.5 WORST CASE (WC)

The worst case model is a non-statistical method defined as;

$$dU = \sum \left[\frac{\partial f}{\partial X_i} \right] T_i \leq T_{ASM}$$ Equation 2-6

Where dU is the predicted assembly variation and T_{ASM} its specified tolerance. X_i is the nominal component dimension, T_i the component tolerance and f(X) the assembly function. The sensitivity of the assembly to variation is represented by the partial derivative $\frac{\partial f}{\partial X_i}$.

.

The weakness of the WC model is that as the total number of component parts increases the probability of all the individual tolerances being at their worst value simultaneously, decreases. The advantage is that it ensures that the assembly will be within specification every time, regardless of how low the probability of a particular assembly occurring is. While this is suitable for critical systems, particularly in military applications, which must work first time, every time. This means that as the number of component parts increases the tolerances must decrease. This method predicts overly cautious tolerances, which increase production costs, to guard against an event which may never happen during the production life of the product. One

area where such an approach may be justified is military missile launch systems which must work first time every time and the consequences of any failure could be devastating.

2.10.6 ROOT SUM SQUARE (RSS)

The RSS model is a statistical model which assumes that the component part variation is normally distributed about a nominal value. It is defined as;

$$dU = \left(\sum \left[\frac{\partial f}{\partial X_i} \right]^2 T_i^2 \right)^{\frac{1}{2}} \leq T_{ASM}$$

Equation 2-7

Where dU is the predicted assembly variation and T_{ASM} its specified tolerance. X_i is the component dimension, T_i the component tolerance and f(X) the assembly function. The sensitivity of the assembly to component dimension variation is represented by the partial derivative $\frac{\partial f}{\partial X_i}$.

By using a statistical model the probable proportion of assemblies (yield) within the tolerance, normally at $\pm 3\sigma$, can be predicted. This allows larger component part tolerances to be used to reduce production costs at the expense of a small proportion, typically 0.3%, of the assemblies failing to meet specification.

2.10.7 SIX SIGMA

This is a modified form of the RSS model which takes into account process mean drift. It can account for both the long and short term process capability of a system and is defined as;

$$dU = Z\left(\sum\left[\frac{\partial f}{\partial X_i}\right]^2\left(\frac{T_i}{3C_p(1-M_i)}\right)^2\right)^{\frac{1}{2}} \leq T_{asm}$$

Equation 2-8

Where C_p is the process capability ratio and M_i is the mean shift factor.

This model is used where high quality levels are expected and a mean shift is anticipated but should not compromise product quality. A typical value for the mean shift is 1.5σ which gives a reject rate of 3.4 ppm.

2.10.8 ASSEMBLY VARIATION

To determine the variation for an assembly level measurement the sensitivity indices are substituted into a tolerance accumulation model. The tolerance accumulation model chosen will depend on the proportion of rejects considered acceptable and the presence or otherwise of any mean shift.

2.11 APPLICATION OF DVA SOFTWARE

There is a considerable body of work from authors such as Chase (1991), Chase and Parkinson (1994), Goa et al. (1998), Cvetko (1998), Glancy and Chase (1999), Laperrière and ElMaraghy (2000), Desrochers et al. (2003), Whitney (2004) and Ghie et al. (2009) on the theoretical aspects of DVA and the mathematical theory behind the various methods employed. There have also been several papers which review or give an overview of DVA from a less mathematical viewpoint. Prisco and Giorleo (2002) reviewed the theory behind and compared the functionality of several CAT or DVA methods. Spicknall and Kumar

(1999) defined 18 criteria for the selection of DVA software for use in the shipbuilding industry

The amount of published work on the application of DVA software is limited. Milberg et. al. (2002) described the application of DVA in the construction of steel framed concrete tunnels. Taylor (1996) demonstrates, albeit using mathematical means rather than DVA software, how variation analysis fits into the concepts of six sigma and robust design which in turn form part of the DM process. In more recent work by Baião et. al. (2011) DVA was used to asses the impact of variation on the FEAD of an automotive engine. The paper also demonstrates some of the auxiliary techniques used in connection with DVA. McFadden (2005) notes that at one major but unspecified automotive manufacturer DVA has been in use for a considerable period but remains misunderstood, underexploited and generally regarded as a post mortem tool. Despite this McFadden suggests that the primary advantage of DVA is the method of approach it offers. He also states that the most critical recommendations for the improved use of DVA relate to the culture surrounding and communication with the DVA analyst. This is the province of DM which when properly implemented provides the structure and organisation to exploit DVA to the full.

3 METHODOLOGY

This chapter introduces the research, outlines the choice of DVA software, the research methodology used and the three case studies undertaken.

3.1 INTRODUCTION TO THE RESEARCH

This project concentrates on three areas of interest connected with the use of this software. The three areas are;

- Pre processing – The collection of necessary information from various sources within the product development process and documentation of that information.
- Processing – The construction of the DVA model and the measurement and analysis of system key characteristics.
- Post processing – The interpretation and dissemination of the analysis output.

The processing is the primary area of interest with a view to the development of new methods or the modification of existing methods that will extend the range of systems that can be analysed and improve the accuracy and realism of the simulation used in the analysis particularly when applied to complex kinematic systems. This area will also include the development of new methods or modification of existing methods used to define and measure key characteristics in complex kinematic systems. The pre and post processing form part of the "culture" that surrounds DVA and which McFadden (2005) considered to be the main advantage of the DVA process. They are also the vital links between DVA and the wider DM process and ultimately to the overall concurrent engineering process. Unless the necessary information can be drawn into the DVA process analysed in an accurate and realistic manner

and the output transferred to those who need to know, then the full benefit of DVA cannot be realised.

3.2 METHODOLOGICAL CONSIDERATIONS

A case study approach was adopted for this particular project. Actual industrial projects and commercial work could not be investigated or shown for reasons of confidentiality. However, the generalised case studies undertaken were chosen to reflect, and relate to, real commercial interests. Studies were performed on three different classes of complex kinematic assemblies: base engine, valve train and suspension system. For each class of assembly three common examples where chosen for investigation. The strong automotive bias reflects the commercial interests of the sponsoring company. The initial approach was determined by the sponsoring company's current process for conducting DVA analysis as part of the wider DM process.

3.2.1 CHOICE OF CAD SOFTWARE

The choice of CAD software was largely dictated by that in use by the sponsoring company. The particular software in question was Dassult's Catia V5R16. This has the advantage that two of the three DVA platforms, Cetol and 3DCS utilised by the sponsoring company are capable of reading the native Catia file format and thus no file translation is required. In the case of the third DVA platform, VisVSA, CAD models are imported using the step file format or translated to the independent jt file format, used by VisVSA by means of a file translator available at Loughborough University.

3.2.2 CHOICE OF DVA SOFTWARE

The choice of DVA software was as with the CAD software largely dictated by the sponsoring company. Three commercial platforms were available to the project.

3.2.2.1 Sigmetrix's Cetol 6 Sigma V7.2

This is the DVA software of choice. It is a kinematic system based on vector loop analysis. As each component is fitted it is not fully fixed but only held according to the degrees of freedom constrained. Once fitted, components remain free to move in any unconstrained degrees of freedom. The component parts once fitted are still free to reorietate should the circumstance change This allows the configuration of the DVA model to be changed simply by altering the defining parameter the component parts then automatically reorientate to the new configuration. While the analysis process is typically slower than Monte Carlo simulation, due to the need to create a sensitivity matrix, it has the advantage that the entire analysis need not be repeated if a single parameter tolerance is changed. Cetol is also capable of reading native Catia V5 file formats so the question of file translation does not arise.

3.2.2.2 Teamcenter's VisVSA 2005 SR1

VisVSA as it is commonly known is widely used and well known DVA software that is quick and simple to use. The analysis is by means of Monte Carlo simulation which allows a wide range of input data to be used with few restrictions on the type of variation distribution or the continuity of the data. The VisVSA software will tolerate a certain amount of under and/or over constraint in the DVA model, this is of considerable assistance in the construction and

debugging of the DVA model. However, the Monte Carlo simulation method does require the entire analysis to be rerun if a single parameter is changed. VisVSA uses the neutral JT file format; consequently all the CAD models will require translation into this format. While such a translator is available there is the possibility that the translation process may introduce errors into the model. A common feature of most Monte Carlo simulation based DVA software including VisVSA is that once a component has been fitted it will not reorientate should the circumstances change. Given the large number of model configurations required when modelling kinematic systems the increased modelling time was considered to be prohibitive.

3.2.2.3 Dimensional Control Systems 3DCS V5

3DCS like Cetol is capable of reading native Catia V5 file formats and thus the need for translation and the possibility of translation errors do not arise. 3DCS like VisVSA uses the Monte Carlo simulation method to perform the analysis and thus has similar capabilities with regard to the construction of large numbers of model configurations. In this particular version of 3DCS available to the project the user interface employed a mixture of point and feature based methods. Prior experience had shown that even with simple static systems the mixed user interface requires considerable expertise in the application of GD&T to the DVA model. For these reasons the Cetol software was considered more suitable for the modelling of complex kinematic systems.

3.3   APPROACH

The sponsoring company's current approach to the DVA process (Figure 3-1) consists of four stages namely information acquisition, model building analysis and output.
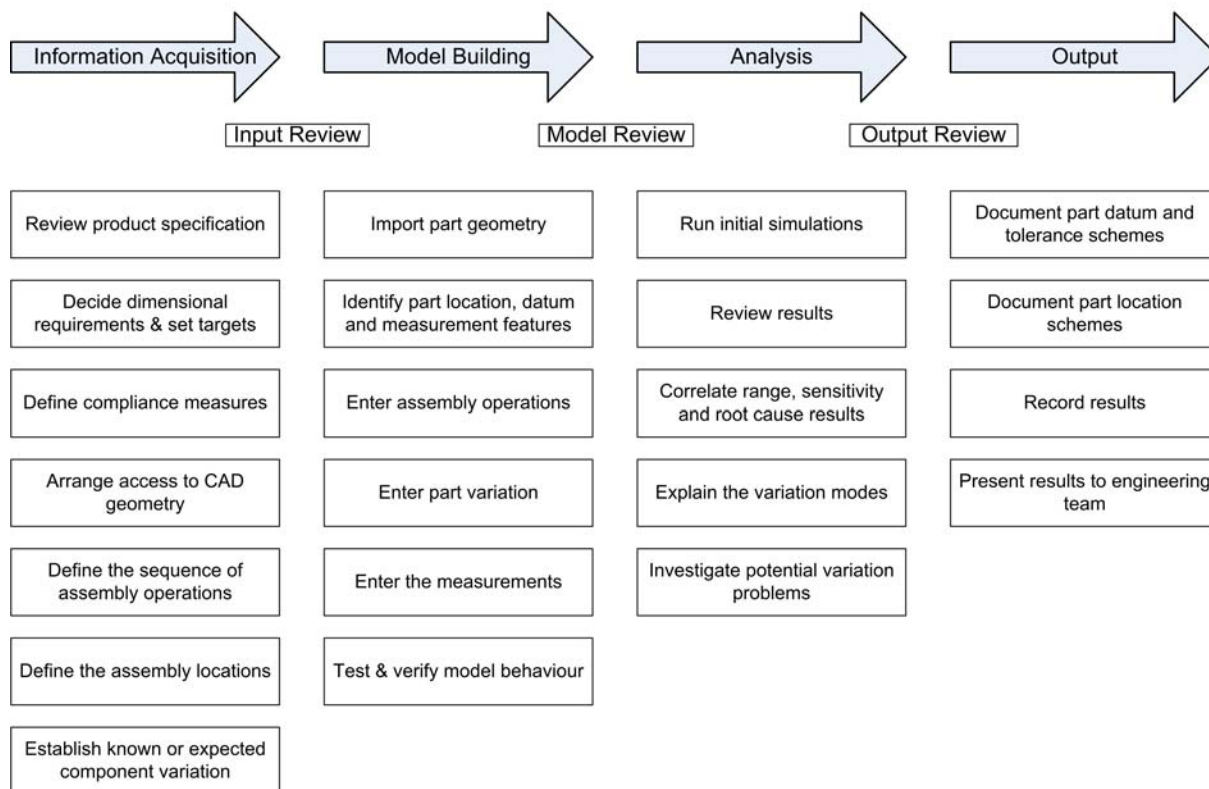
Figure 3-1 DVA process chart (i-dmsolutions, 2008)

3.3.1 INFORMATION ACQUISITION

The first task is to determine what dimensional requirements are to be investigated, how these requirements can be represented in the DVA model and measured to ensure compliance. For example the specification for a suspension system may call for straight line stability. The dimensional requirements to satisfy this requirement may be met by a front wheel toe in of 3.1 mm ± 2.2 mm which increases under heavy braking. If the toe in measured with the suspension fully extended, at the nominal ride height and fully compressed lies within this

range and the toe in when fully compressed is greater than that at the nominal ride height then the system may be deemed to comply with the specification.

The second task is to acquire the information necessary to construct the DVA model. The component geometry must be in, or translated into, a suitable format. The order in which the component parts are assembled, the features on adjacent parts that make contact and the extent of any component variation must be defined.

3.3.2 MODEL BUILDING

The component part geometry is imported into the DVA model and the location, datum and measurement features identified and the appropriate variation to be assigned to the features. Once the various features are incorporated in the DVA model the assembly operations are specified. Location features on mating component parts are constrained together to simulate the assembly process. Once the assembly process is complete the assembly level measurements or key characteristics that constitute the compliance measures are added. The final stage in the model building is to verify the model. This is achieved by the use of functions within Cetol and through the inclusion of check measurements in the DVA model to test the model behaviour. The first of these allows the state of constraint and model closure to be assessed. Model closure and kinematic constraint of the assembly are necessary conditions for analysis. The second function places the component parts in the assembled position which allows a visual check of the model behaviour. The final stage of verification is to run a preliminary analysis of the model to ensure that it produces the expected results. In complex systems check measurements may be included in the simulation model. These are assembly level measurements the sole purpose of which is to ascertain that the model behaves in the

expected manner. They are often useful when visual examination alone is insufficient to verify the model behaviour.

3.3.3 ANALYSIS

It is not the actual running of the analysis, which is a largely automatic process, but the correct interpretation of the results that is vital. A kinematic system may contain numerous model configurations in which case it may be helpful to tabulate or otherwise process the raw analysis output before it can be reviewed. Review of the output will establish if the results are consistent with the expected system behaviour, if there are any general trends and if the individual contributor sensitivity and variation range analyses are consistent with each other. Unexpected system behaviour may be due to model error or previously unknown system behaviour and appropriate action should be taken to either correct the model or explain the behaviour.

3.3.4 OUTPUT

This section of the DVA process consists of documentation and information transfer. For the product design and development to benefit from the DVA work it is essential to record and communicate the results. The information acquired, the DVA model and the analysis results are fully documented so that they can be reproduced should the system be revisited at a later date. The information transfer is an essential function of the wider CE process, it is at this stage that the necessary knowledge is transferred to the people who need to know and in a form that they can readily understand thus realising the full benefit of the DVA process.

3.3.5 MODELLING OF KINEMATIC SYSTEMS

Kinematic systems involve gross system movements through one or more predefined ranges or cyclic operations. However, the DVA software performs a static analysis. One method of dealing with movement ranges in kinematic systems is to divide the movement range into a series of incremental steps, each of which is analysed separately. In the case of vector loop based DVA software each incremental step in the movement range becomes a separate configuration of the simulation model and it is only necessary to realign the simulation model to each configuration to permit analysis. This raises the question as to what method will be used to align the component parts of the simulation model into the various assembly configurations required.

Where the assembly is particularly large it may be advantageous not to construct a DVA model of the entire assembly system. To conserve resources and reduce analysis times the DVA model may consist of only part of the overall assembly. In such circumstances, a method must be devised of locating the DVA model relative to the parent assembly and of mimicking any relevant dimensional variation behaviour of those component parts excluded from the DVA model. To be effective a DVA model must be capable of measuring and evaluating the effects of variation on the chosen system attributes or key characteristics. If no appropriate CAD geometry exists to support the necessary measurement features a means of incorporating suitable fixtures, gauges or jigs into the DVA model must be devised. The method chosen must also be capable of accommodating any changes in size or orientation of the required measurement feature as the system progresses through the movement ranges.

3.3.6 CASE STUDY 1: BASE ENGINES

A base engine consists of the engine block, crankshaft, connecting rod, gudgeon pin, piston and any associated bearings (Figure 3-2 to 3-4). Two factors known to strongly influence engine performance are compression ratio, which is largely dependant upon maximum piston height, and crank timing. The aim of this case study was to model three common engine arrangements flat, inline and vee in order to analyse how component variation is likely to affect the overall engine performance in terms of piston height and crank timing. The objective of this case study is to simulate the variation in the maximum piston height and the variation in the crank position at maximum piston height. Variation in the maximum piston height may affect the compression ratio. In multi-cylinder engines such as those chosen for this case study this could lead to a difference in compression ratios across the four or six cylinders of the engine and affect the overall performance of the engine.
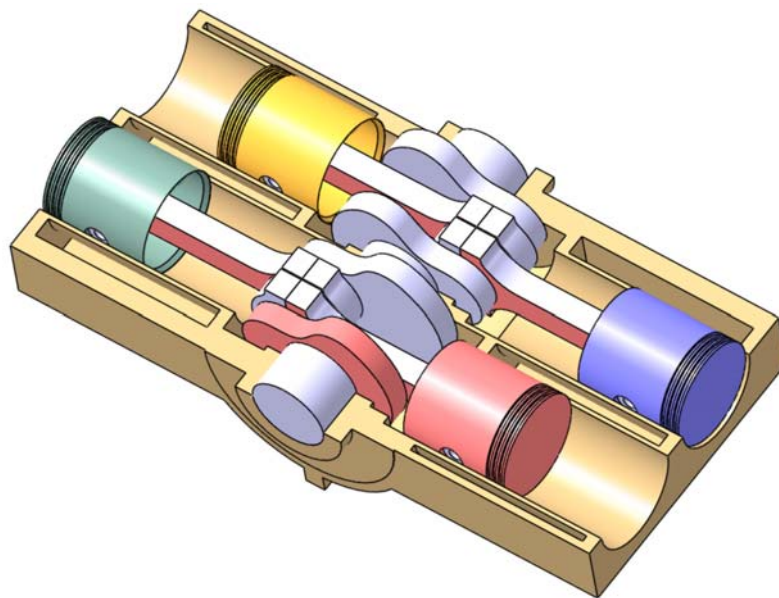
Figure 3-2 Flat four base engine
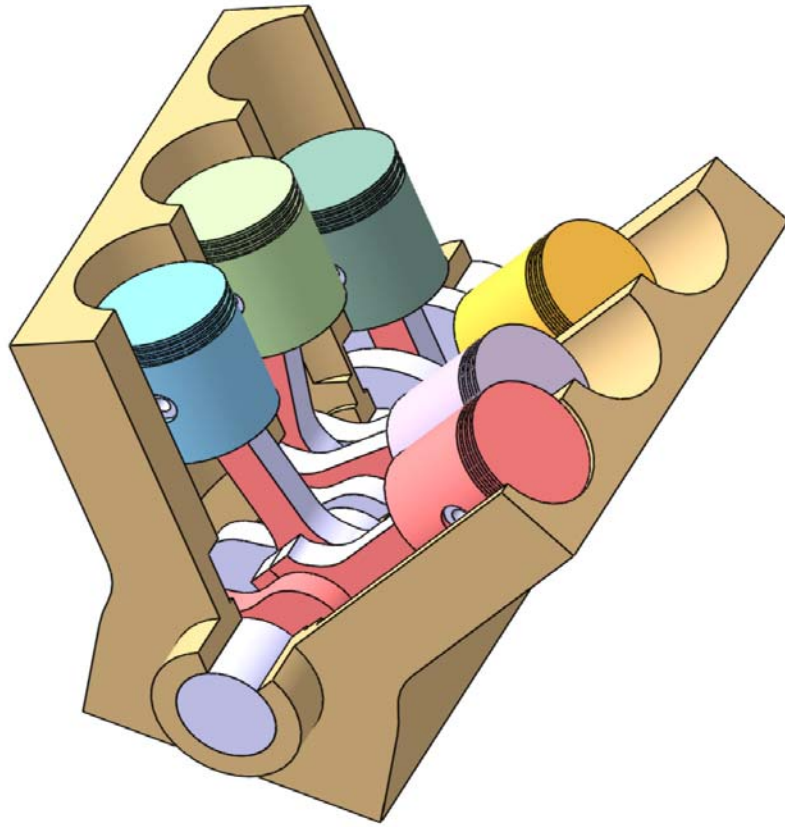
Figure 3-3 Vee six base engine
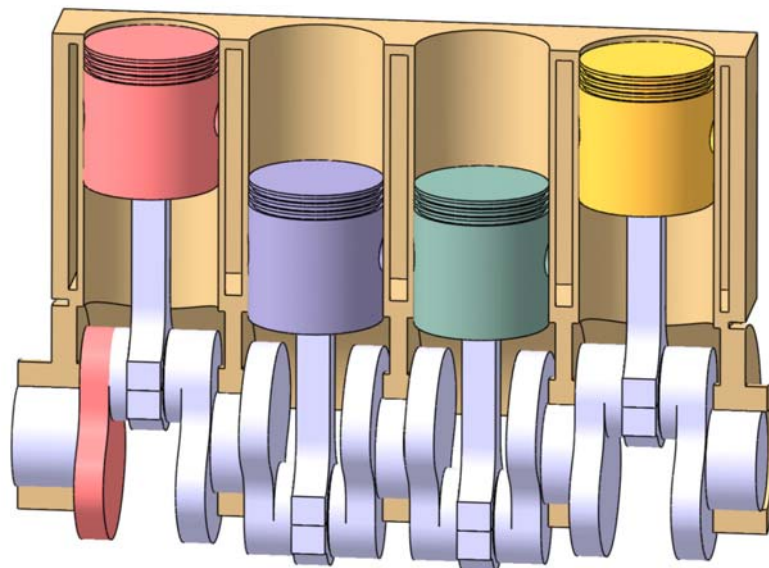
Figure 3-4 Inline four base engine

Variation in the crank position at maximum piston height can be significant as often many of the other engine systems, such as ignition or valve timing, are set relative to the crankshaft position, but are directly affected by the position, or relative position of the piston.

3.3.7 CASE STUDY 2: VALVE TRAINS

The point at which the valve opens or closes (valve timing) and the maximum valve lift are three factors known to strongly influence engine performance. The aim of this case study is to model three common valve operating systems, bucket and shim, under slung rocker arm and over slung rocker arm in order to analyse how dimensional variation is likely to affect the overall performance of the valve system in terms of valve timing and maximum valve lift. The objective of this case study is to determine the effects of component variation on the valve lift for a given cam angle and the variation in the cam angle at the point where the valve is just opening and closing. The three types of valve train used in the case study are shown in Figures 3-5 to 3-7. The operation of three valve systems differs significantly. In the bucket and shim system (Figure 3-5) the cam bears directly on the hydraulic bucket adjuster which in turn bears directly onto the valve stem. The motion imparted by the cam is almost identical to the valve lift there being no rocker arm present.
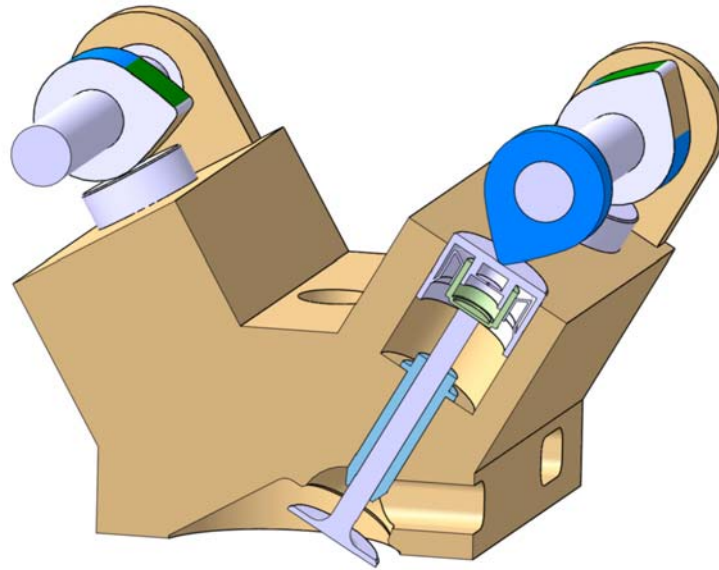
Figure 3-5 Cross section of a bucket and shim valve train
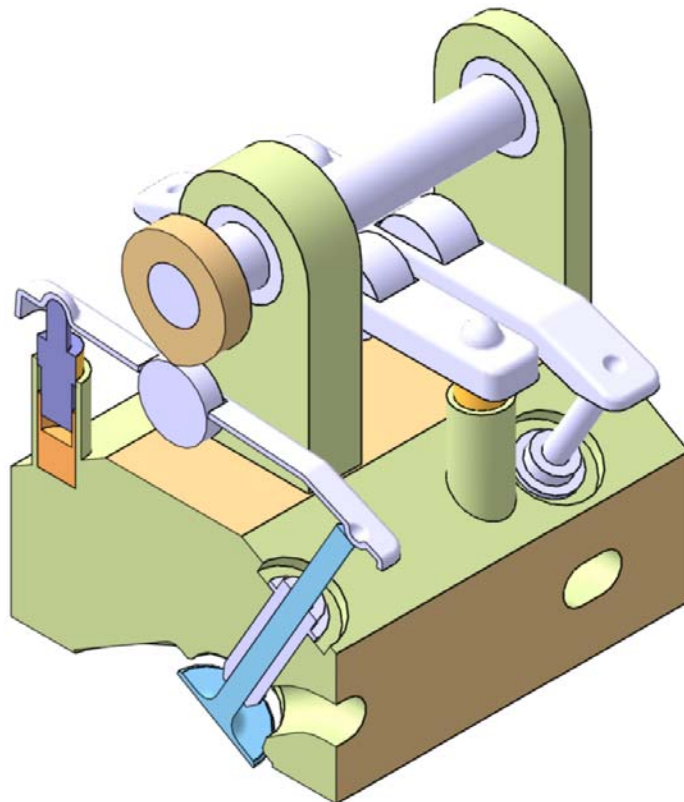
Figure 3-6 Cross section of an under slung rocker arm valve train
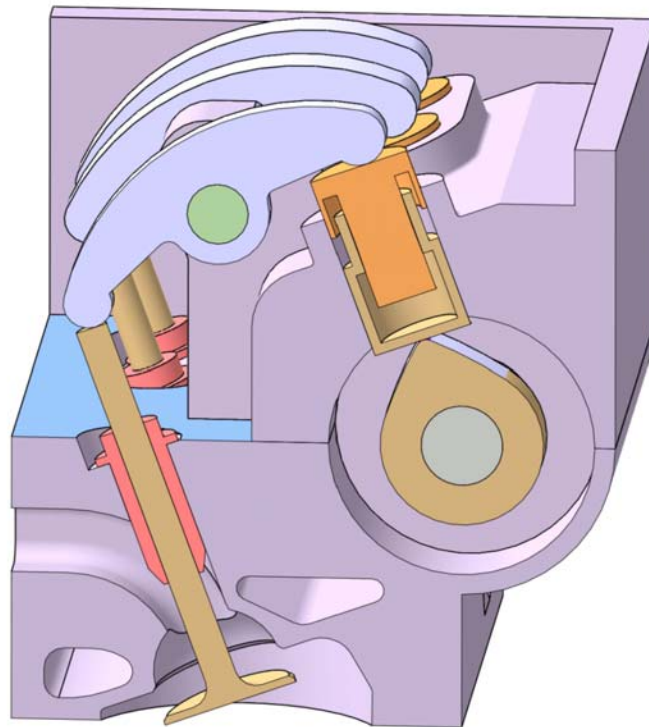
Figure 3-7 Cross section of an over slung rocker arm valve train

The under slung rocker arm system (Figure 3-6) contains a rocker arm that pivots about the hydraulic tappet adjuster which is located at the opposite end of the rocker arm to the valve. The cam contacts the rocker arm near the midpoint of the rocker arm and in consequence the valve lift is considerably greater than the motion imparted to the rocker arm by the cam. In the over slung rocker arm system the rocker arm rotates about a fixed central shaft which is common to all the rocker arms in the system. The valve lift is of similar magnitude to the motion imparted by the cam on the hydraulic tappet adjuster

3.3.8 CASE STUDY 3: SUSPENSION SYSTEMS

The three common suspension systems were modelled and the performance evaluated by means of three parameters, castor angle, camber angle and toe in. The suspension systems

chosen consist of a McPherson strut and a double wish bone system, both of which are modelled as front suspension systems and a twist beam system modelled as a rear suspension system. The McPherson strut and double wish bone systems contain two movement ranges of interest namely the ride height (up-down) and the steering lock (side to side). In the case of the McPherson strut and double wish bone systems the toe-in is defined as the difference in the distance between the leading and trailing edges of the wheel and the vehicle centreline as only a quarter car CAD model is used. As the twist beam system is a half car model the toe-in is measured as half the difference between the distance between the leading and trailing edges of the wheels. The twist beam system also only has a single (up-down) movement range as it is a non steering system.
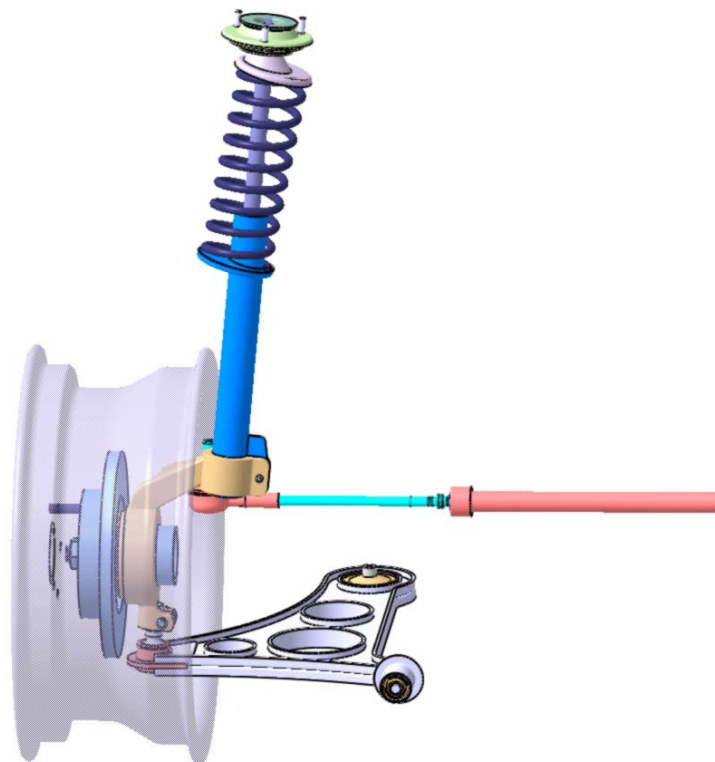
Figure 3-8 McPherson strut suspension system viewed from the front of the car
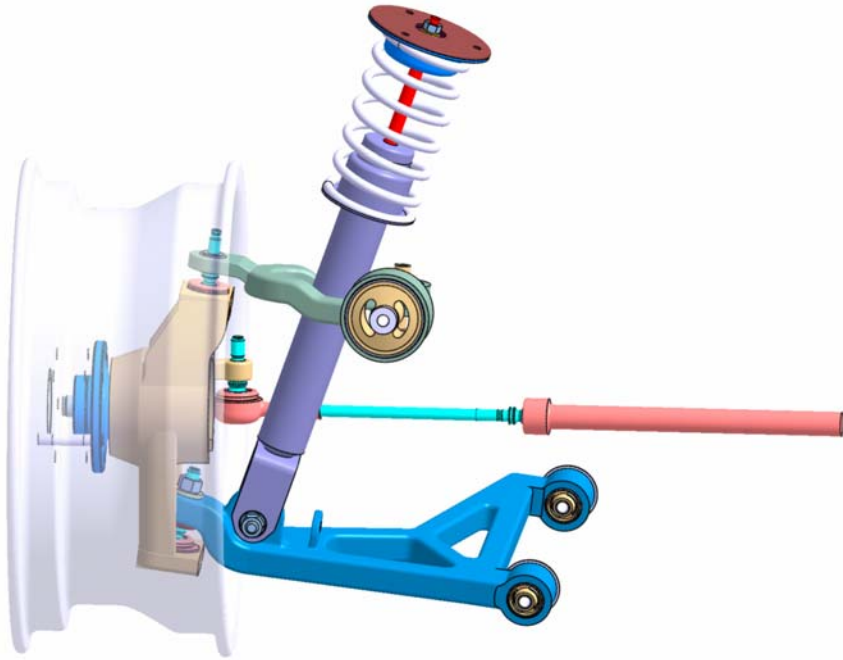
Figure 3-9 Double wish bone suspension system viewed from the front of the car
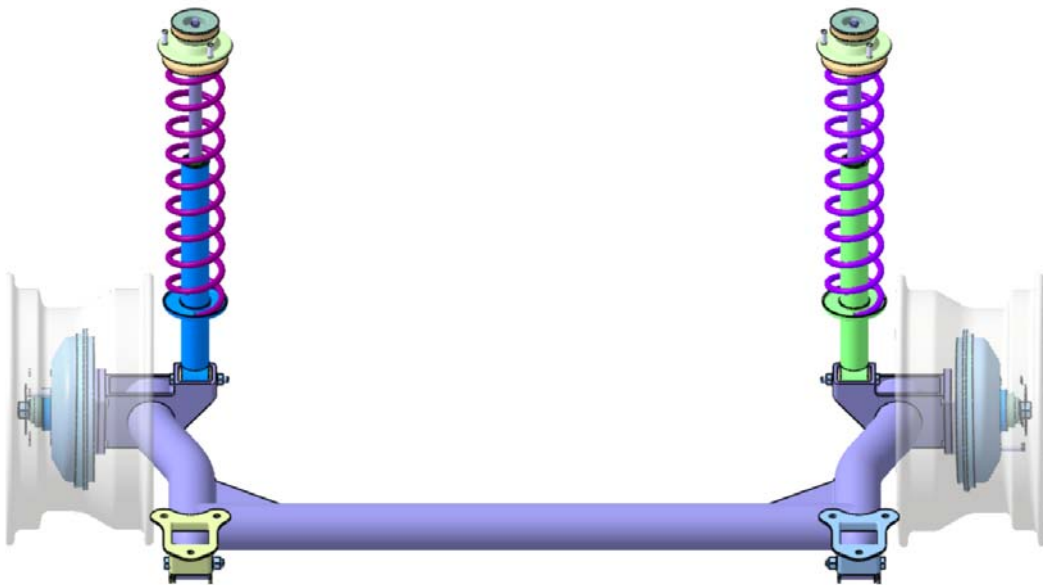
Figure 3-10 Twist beam suspension system viewed from the front of the car

4   THE RESEARCH UNDERTAKEN

The research is divided into two distinct sections the first of these involves modelling each example of the three different classes of complex kinematic systems using the current DVA process developed by the sponsoring company for static systems and mechanisms and noting any issues that arise. These issues form the preliminary findings (see section 4.1). The second section of the research, the case study findings (see section 4.2), describes the remedies used to addresses those issues raised in the preliminary findings either by modifying the existing DVA process or by developing new methods.

4.1   PRELIMINARY FINDINGS

This section details the issues that arose during the construction of the various DVA models. The issues raised in this section constitute areas in which the current DVA process used by the sponsoring company requires enhancement for use with complex kinematic systems. The issues are grouped according to the three basic system types.

4.1.1 BASE ENGINES

During the construction of the base engine DVA model two significant issues arose;

4.1.1.1 Definition of maximum piston height (TDC)

One of the objectives of this case study is to measure the maximum piston height. A frequently used definition for maximum piston height is; the point at which the big end journal is directly above the crankshaft. This definition, however, is not valid if the cylinder bore and crankshaft are offset and is thus unsuitable for a DVA model. Neither can the piston

be fixed at its nominal maximum height as this would not allow the piston to react to any variation in the system A definition of maximum piston height that aligns the base engine to the configuration but does not restrict the movement of the piston is thus required to allow measurement of the variation in piston height.

4.1.1.2 Aligning DVA model to maximum piston height

Having defined the maximum piston height, how is the DVA model to be aligned to the configuration. In the real world an engine is often set to TDC by aligning a mark on the crankshaft to a similar mark on the crankcase. However, this method does not take into consideration the effects of component variation and is thus not particularly accurate. A second and highly accurate real world solution is to measure the vertical height of the piston with a dial gauge direct onto the piston. This method can determine the exactly when the piston reaches maximum height. The process also takes into account any and all variation in the system which may affect the piston height. Unfortunately neither of these methods is feasible when constructing a DVA model. The first method lacks the necessary accuracy. The second, given the slight difference between real world practice in which the piston height is measured continuously and the DVA model where the piston height is measured for each discrete model configuration may require a considerable number of iterations (Figure 4-1) and still prove insufficient to align the system exactly to the maximum piston height.
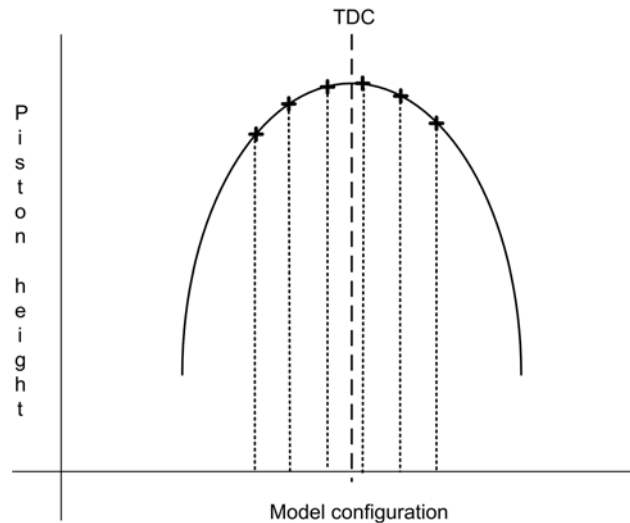
Figure 4-1 Piston height against model configuration

4.1.2 VALVE TRAINS

Of the three case studies the valve train study provides the greatest challenge to the current DVA process. Some of the issues encountered, such as the definition of particular system states are similar to issues raised by the base engine study. Other issues provide a challenge to the basic concepts and methods of the current DVA process. The major issues encountered include;

4.1.2.1 Modelling hydraulic tappet adjusters

While it is possible to define the axial size and location of the hydraulic tappet adjuster sub assembly by adopting a specific configuration for the DVA model the definition is only valid for this one configuration. When the DVA model is reconfigured to analyse the effects of variation on the valve lift or cam angle the axial length of the hydraulic tappet adjuster became indeterminate which precludes analysis. The current DVA process requires enhancement to allow the modelling of systems with dependant configurations where

information derived from one model configuration is incorporated into the remaining model configurations.

4.1.2.2 When is an engine valve just opening or just closing.

A definition of this state is required so that the variation in the cam angle in such a condition can be measured. However, at the point of valve opening or closing several other significant changes of state occur which complicate the issue. The location in the DVA model between the cam and the cam follower changes. When the valve is closed the cam follower is in contact with the base circle of the cam. When the valve is open the cam follower makes contact with the cam flank (or later the cam toe). Theoretically, assuming there is no variation or clearance gaps in the system, the valve will begin to open as the cam follower transits the boundary between the cam base circle and the cam flank. It is, however, at this point that the nature of the hydraulic tappet adjuster changes and the previously variable sub assembly is assumed to become effectively rigid. The situation is still further complicated by the fact that the profiles of the cam base circle and flank, in this instance, meet tangentially at the transition boundary between the two. In real life the two profile sections merge seamlessly into one another with no discernable boundary between them. However, in the CAD model, due to the manner in which it is constructed, the two features are separated by a distinct boundary line which can be selected as an assembly feature in the DVA model. It is theoretically possible for the cam follower to be in contact with both the cam base circle, cam flank and the boundary line simultaneously. As all three features can be selected as assembly features in the DVA model the question arises as to which of the three cam features the cam follower should be located on.

4.1.2.3 Aberrant model behaviour

When configuring the simulation model if any component with a rotational range of movement such as a camshaft were rotated by more than 90° from the default position then aberrant behaviour of the DVA model was observed. Components rotated in a retrograde manner or detached from the simulation model and assumed unexpected positions

Default position   Plus 89 degrees   Plus 91 degrees
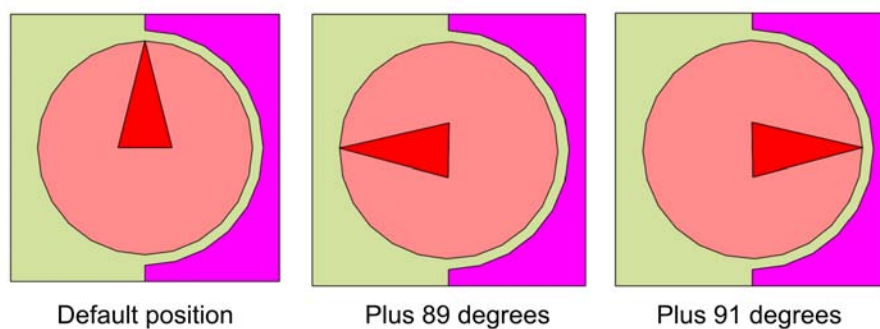
Figure 4-2 Aberrant behaviour of components rotated in excess of 90 degrees

A second instance of aberrant behaviour was noted in those configurations of the DVA model where the contact between the cam and cam follower approached or receded from the cam flank and cam toe transition boundary. The aberrant behaviour appeared as a visible gap between the cam and cam follower (Figure 4-3) despite the fact that the inbuilt testing functions of the DVA software verified the DVA model. The visible gap is not merely a computer graphics error but actually influences the valve lift (Table 4-1). The fact that the visible gap also influences the standard deviation of the valve lift suggests that the DVA software "sees" the model differently.
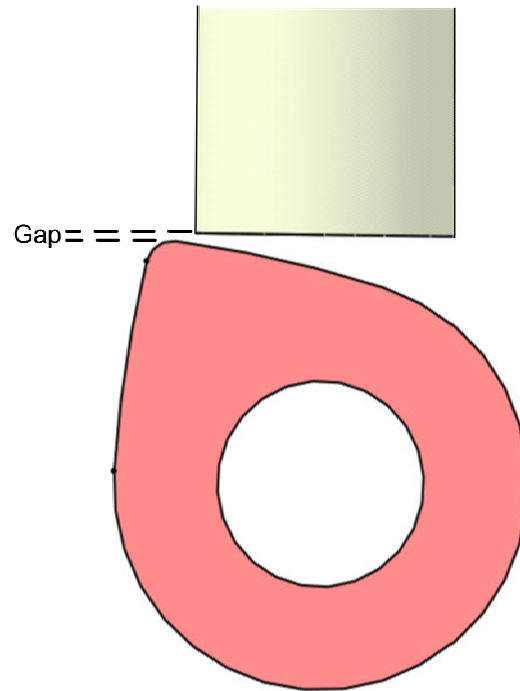
Figure 4-3 Aberrant behaviour in the form of a visible gap between cam and cam follower.

Table 4-1 Effect of visible gap on valve lift

Cam Angle (° ATDC)	Joint Configuration	Valve Lift (mm)	Standard Deviation (mm)
15	Initial	2.0851900	0.0292845
15	Revised	1.8513100	0.0209455
20	Initial	4.1580500	0.0407581
20	Revised	3.1303800	0.0239868

4.1.2.4 Correct part location

In those DVA model configurations where the cam angle was fixed and the valve lift measured the exact contact between the cam and the cam follower could on occasions be difficult to determine especially when the contact occurred in close proximity to a transition boundary between two facets of the cam profile. A visual inspection of the simulation model often proved inadequate to resolve the issue even when the transition point was known.

Neither could it necessarily be determined from the analysis output if an error had been made in selecting the correct cam facet.

4.1.3 SUSPENSION SYSTEMS

The suspension system case studies raised a number of issues including the following;

4.1.3.1 Exclusion of CAD geometry

The suspension systems locate from either a cross beam, sub frame or the body in white. It may not be feasible, particularly in the case of the body in white to include the entire locating structure in the DVA model.

4.1.3.2 Measurement features

In the DVA model of the McPherson strut it was discovered that there was no suitable geometry to act as a measurement feature for measurement of the castor angle. This is the angle between the steering axis and the vertical plane when viewed from the side of the vehicle. In the McPherson strut system the steering axis links the upper and lower pivot points and lies largely outside the component part geometry. The size and orientation of the steering axis are also affected by the differences between the DVA model configurations. Attaching measurement features to a component part in the CAD model to measure system attributes such as the castor angle is not feasible as the features either, tended to move during the construction of the CAD model and are thus unreliable or the DVA model became over constrained precluding analysis.

4.1.3.3 Modelling of rubber bushes

Each of the three suspension systems contain rubber bushes. These are non rigid components which may allow relative motion between two adjacent components even if the rubber bush is rigidly attached to both components. This raises the question as to how the behaviour of a non rigid component is to be simulated when a basic assumption of the DVA software is that all the components are rigid.

4.1.3.4 Documentation of design intent

  In modelling the McPherson strut suspension system a question arises when modelling the behaviour of the two rubber bushes at the inboard end of the lower arm (Figure 3-8). In the real world the original design intent is available during the product development process. In this particular case study the CAD geometry was based on the McPherson strut system used in a Ford Fiesta, the original design intent being inferred from the inspection of an actual example. This raised the question as to how the original design intent can be documented in sufficient detail to construct a DVA model. The current method used by the sponsoring company is used in conjunction with an assembly sequence diagram (Figure 4-4) and consists of tabulating the information for each assembly operation (Table 4-2)
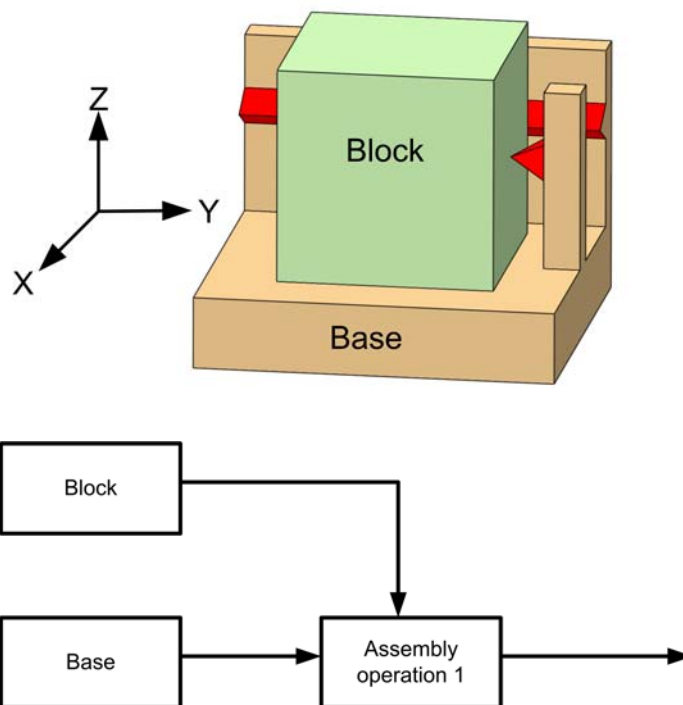
Figure 4-4 A simple assembly and associated assembly sequence diagram

Table 4-2 Current method of documenting assembly operation constraints

Assembly operation 1						
Location	Constraint					
	Tx	Ty	Tz	Rx	Ry	Rz
Surface to surface contact			●	●	●	
Edge to surface contact	●					●
Point to surface contact		●				

The use of this method when modelling the McPherson strut suspension system proved challenging and prone to errors which were difficult to locate. If a datum flow chain or variation propagation chain were required then it is necessary to plot a separate diagram which further complicates the data management process. When employed on kinematic

systems such as suspension systems there is no simple test to check if the component is kinematically, under or over constrained this makes constraint errors, which may prevent model closure and subsequent analysis, difficult to detect and does not aid the modelling process.

4.2 CASE STUDY FINDINGS

This section outlines the modifications to the existing methods, the new methods developed and the conceptual changes adopted to resolve the issues raised in the preliminary findings (see section 4.1)

4.2.1 BASE ENGINES

4.2.1.1 Definition of maximum piston height

The definition of maximum piston used in this case study is;

"The piston is at its maximum height when the centre lines of the crankshaft main bearing, big end journal and gudgeon pin can be intersected, in that order, simultaneously by a single straight line."

This fixes the position of the piston in the cylinder bore without restricting the unconstrained movement of the piston and thus allows the measurement of any variation in the piston height.

4.2.1.2 Aligning DVA model to maximum piston height

A virtual jig (Paper 2, Appendix B) is used to align the DVA model to maximum piston height. A point to note is that the definition of the maximum piston height also effectively defines the nature of the virtual jig used to align the DVA model into the correct configuration. The initial virtual jig was designed as if it were a real world component (Figure 4-5). However, just like a real world artefact the possibility of variation was present. The jig relied on the square cut-outs being a tight fit on the gudgeon pin and big end journal. If these varied in size the alignment might suffer. Conversely the initial alignment jig was assembled to the centre line of the crankshaft using a virtual point which was immune to the effects of component variation. In light of this, the final version of the virtual jig consisted of a single virtual line that was assembled to the axes of the crankshaft, big end journal and the gudgeon pin. In this form the alignment jig was immune to the effects of component part variation but had a tendency to be mistaken for a construction line.
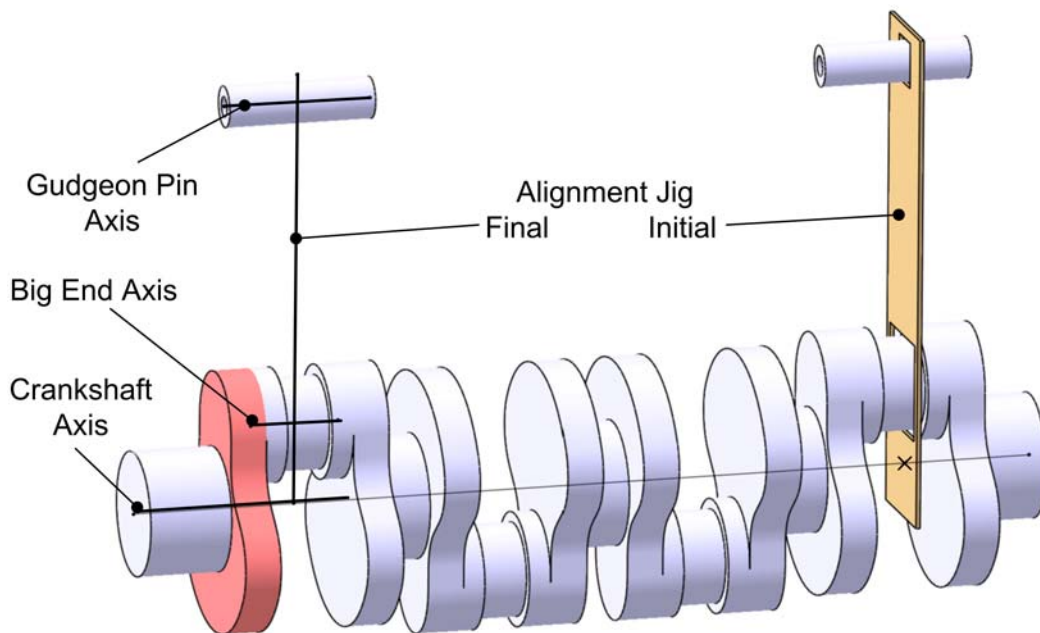
Figure 4-5 Base engine alignment jigs

This is overcome by incorporating sufficient solid geometry into the virtual jig to allow easy identification. However, the solid geometry does not necessarily play any part in the alignment process

4.2.2 VALVE TRAINS

4.2.2.1 Modelling hydraulic tappet adjusters

To simulate the behaviour of the hydraulic tappet adjusters it was necessary to develop new concepts and a new method of modelling dependent configurations in a DVA model. This method is explained in detail in paper 4, (Appendix D).

4.2.2.2 When is an engine valve just opening or just closing.

The point at which a valve is just opening or closing was defined as that point at which the valve lift was equal to 0.01 mm. This definition was chosen as it defines two and only two points in the operational cycle, whereas the valve is in contact with the valve seat for a considerable period. Secondly if the valve has lifted from its seat the cam follower must be in contact with the flank of the cam thus resolving any issue as to which part features are involved in the assembly of the cam follower to the cam. Theoretically, assuming no clearance or slack in the system, the valve opens as it transits the boundary between the cam base circle and flank. It is possible to model this configuration in the simulation model but the analysis results it produces give rise to concern. Table 4-3 shows the valve lift for a cam angle of 3 degrees, the point at which the valve theoretically opens.

Table 4-3 Valve Lift at a cam angle of 3 degrees

Feature in contact with cam follower	Valve Lift (mm)	Valve Lift Standard Deviation (mm)
Cam Heel	0.00018499	0.00422983
Transition boundary	0.00018499	0.00069058
Cam Flank	0.00018840	0.00422983

The most notable feature is that the valve lift when the cam follower is in contact with the base circle of the cam is not zero. This is due to slight discrepancies in the CAD model. As it has been said no model is perfect but some are useful. The interesting feature is that the valve lift when the cam follower is in contact with the cam flank is greater than that of the other two configurations. A significant feature is that the standard deviation of the cam follower to transitional boundary configuration is different to the other two. This suggests that, since the contributing variation sources are identical in all three configurations, the transitional configuration may be unreliable. Of the two remaining configurations the cam follower cam flank was chosen as this is consistent with the valve having started to open. The value of 0.01 mm for the valve lift was chosen as this is small enough to prevent any significant gas flow past the valve but big enough so than it will not be masked by model error and can be measured comparatively easily in the real world if necessary. This particular example illustrates the point that the numerical value of a result may not be nearly as important as the trend it illustrates, particularly so in kinematic systems.

4.2.2.3 Aberrant model behaviour

With regard to the behaviour of the DVA model no solution was found to the issue of the aberrant behaviour of a component part when rotated more than 90° from the default CAD position. The problem is thought to be inherent in the DVA software. However, it is possible to almost entirely avoid this issue by a simple modification to the CAD geometry. Where a system contains one or more movement ranges the default position of the CAD geometry should be set to the mid point(s) of the movement range(s). Provided the angular movement ranges of the system are less than 180° the DVA model can be configured across the entire movement range as the rotational range in the DVA model will be less than ± 90° and the issue will not arise. If the angular movement range of the system exceeds 180° then it will be necessary to use two versions of the CAD geometry the two default positions being 180° apart.
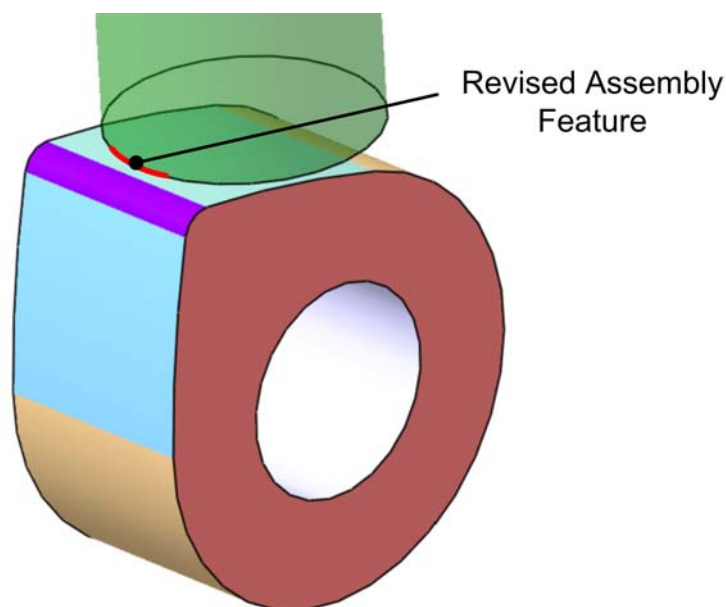
Figure 4-6 Cam follower revised assembly feature

The second instance of aberrant behaviour is connected to the nature of the contact between the cam flank and cam follower. Similar cam/cam follower interfaces have been used in the

Skoda 136X and the Ford Endura-E engines. However, in this instance the major diameter of the cam is significantly larger than the diameter of the cam follower. This allows the contact between the cam and cam follower to change from a line contact to a point contact as the cam rotates rather than maintaining a line contact as is normally the case. This particular instance is not a particularly strong or robust design. However, a conscious decision was made to use this design in the DVA model for two reasons. Firstly it provided an additional challenge to the modelling process and secondly it is those designs that are weak and non robust that benefit most from DVA. If DVA were only applied to strong robust designs it would largely defeat the object of the exercise. The solution to this particular issue was to change the nature of the assembly feature on the cam follower in those model configurations where aberrant behaviour was apparent. Instead of assembling the bottom face of the cam follower to the flank of the cam a short arc on the leading edge of the cam follower was assembled to the cam flank (Figure 4-6). To ensure that the short arc followed any changes in the topography of the cam follower the short arc was constrained to the bottom face and cylindrical surface of the cam follower. These modifications maintain the point contact between the cam and cam follower and do not detract from the realism of the assembly operation.

4.2.3 SUSPENSION SYSTEMS

4.2.3.1 Exclusion of CAD geometry

Paper 2 (Appendix B) explains how a virtual fixture can be used to mimic the effect of a component, such as the body in white, which has been excluded from the DVA model to reduce the model complexity.

4.2.3.2 Measurement features

The lack of suitable measurement features in the CAD geometry and the variation in size and orientation of the parameter being measured in each configuration of the simulation model was overcome by the use of virtual gauges. The nature and use of such devices are explained in detail in paper 2.

4.2.3.3 Modelling of rubber bushes

The modelling of non rigid components such as rubber bushes is possible using a combined FEA/DVA approach (Mortensen, 2002. Imani & Pour 2009) but is time consuming. The alternative is to mimic the real world behaviour of the rubber bush but apply that behaviour to an essentially rigid component to conform to the inherent software assumption that all components are rigid. A common form of rubber bush consists of two coaxial metal tubes permanently bonded together by an elastomeric compound. If the CAD geometry of the rubber bush is accessible then one of the more aesthetically pleasing methods of mimicking the behaviour of a rubber bush is to divide the rubber element, in this instance, into three articulated annular sections (Figure 4-7). The spherical boundary between sections one and two acts as a ball joint while the cylindrical boundary between sections two and three allows for axial displacement either boundary could be used to simulate rotation about the vertical axis. Figure 4-7 shows the rear bush of the McPherson strut system (Figure 3-8) to which a 10° rotation about the Y axis has been applied combined with a 3mm deflection along the Z axis.
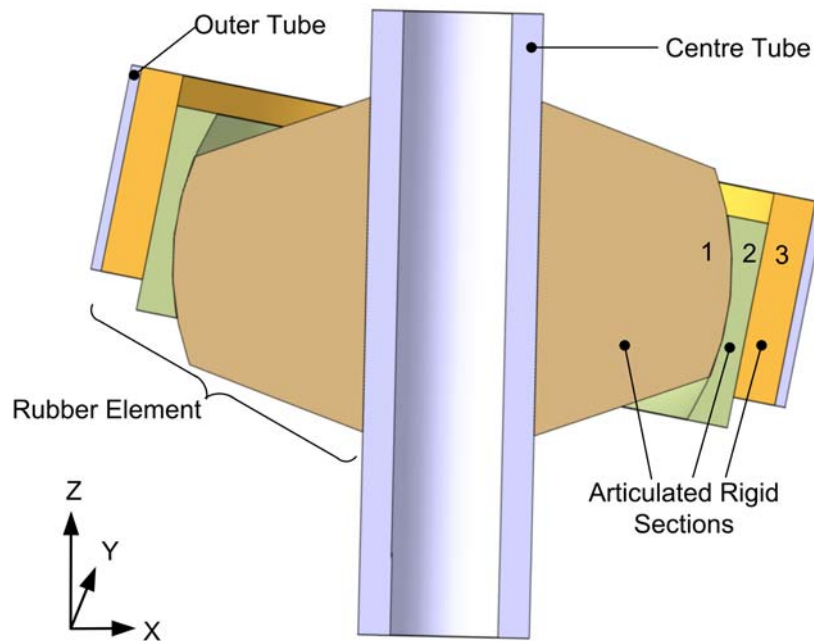
Figure 4-7 Mimicking the behaviour of a rubber bush by segmentation of the rubber element into three articulated rigid sections
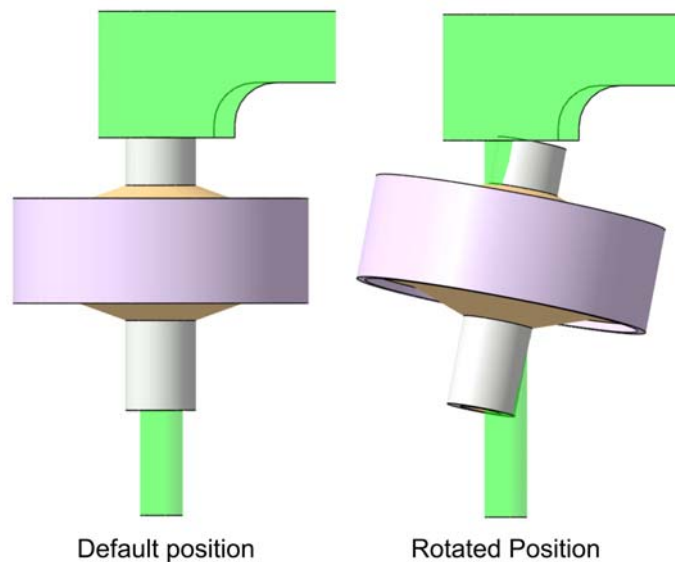
Figure 4-8 Simplified method of modelling rubber bushes which transfers the interaction to the interface with the adjoining component

Many companies, particularly those employing an engineering or product data management

system (EDM, PDM) restrict access to the master CAD geometry. Where the CAD geometry

has been transferred or copied using an IGES or STP file format then much if not all of the CAD history will have been lost. Equally the rubber bush may have been constructed as a single revolved solid in the CAD model. In such circumstances it may be necessary to mimic the behaviour of a rubber bush using a different method. If the behaviour of the rubber bush cannot be mimicked by interactions between the rubber elements as in Figure 4-7 it is necessary to transfer those interactions to the contact points between the rubber bush and the adjacent component parts. Consider the 10° rotation described in the previous method. This is transferred, in this instance, to the contact between the centre tube and the virtual fixture. In the real world this joint would be a deep pin in hole joint. However, to mimic the behaviour of the rubber bush this is modified to a ball joint. This allows the outer tube of the rubber bush to be rotated 10° relative to the pin on the virtual fixture (Figure 4-8) to give the same overall effect as in the previous method although the appearance of the DVA model is far less aesthetically pleasing.

4.3 CHECK MEASUREMENTS

A check measurement is one which is added to the DVA model for the sole purpose of assessing the correct operation of the DVA model. In most static systems and simple mechanisms there is little call for the use of check measurements as the component locations are clearly defined. However, check measurements become more significant in kinematic systems and the associated multiple configurations. In moving from one configuration to the next the relative positions of adjoining parts may change sufficiently to influence which component parts are involved in any given assembly joint. The situation is further complicated if one or both of the component parts has multiple alternate assembly features Consider the cam used in the over slung valve train system (Figure 3-7). This has four

alternative assembly features involved in the assembly joint with the single assembly feature of the cam follower base. Thus as the cam rotates through the movement range the cam base circle, leading flank, cam toe and trailing flank will all be assembled in sequence to the bottom face of the cam follower. It is important that the correct pairing of assembly features is used for each model configuration. An error in the paring of the assembly features may still produce plausible analysis results but the analysis will not be accurate or realistic. The correct pairing of assembly features can usually be determined from a visual examination of the DVA model but this is not always the case. The use of one or more check measurements can resolve the issue by determining the relative position of the boundary between two alternate assembly features on the cam and the base of the cam follower. Returning to the example of the cam and cam follower consider the situation as the cam rotates from the TDC position. Initially the cam follower is in contact with the base circle of the cam. At 3° ATDC the contact moves from the cam base circle to the leading flank of the cam. Visually this change is difficult to detect even when the exact point at which it occurs is known as can be seen in Figure 4-9.
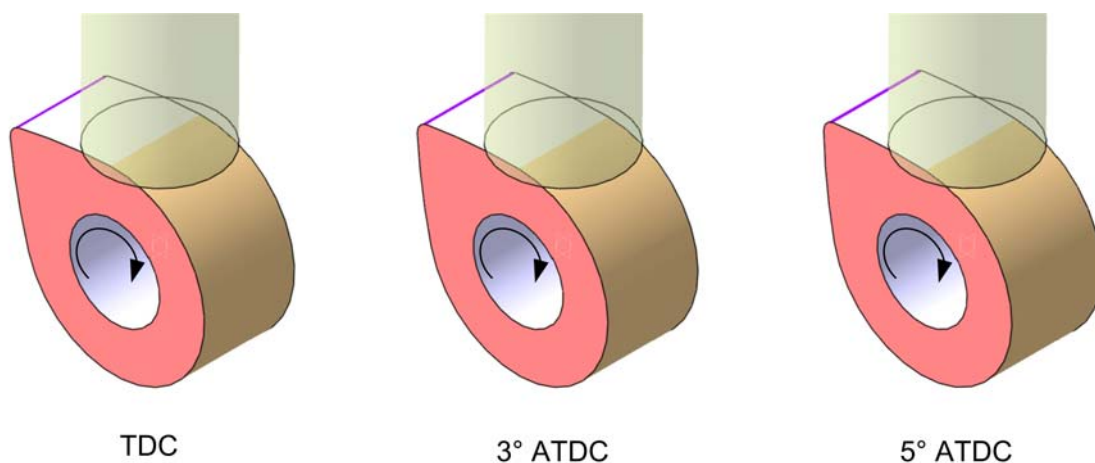
Figure 4-9 Cam follower traversing boundary between cam heel and cam flank assembly features
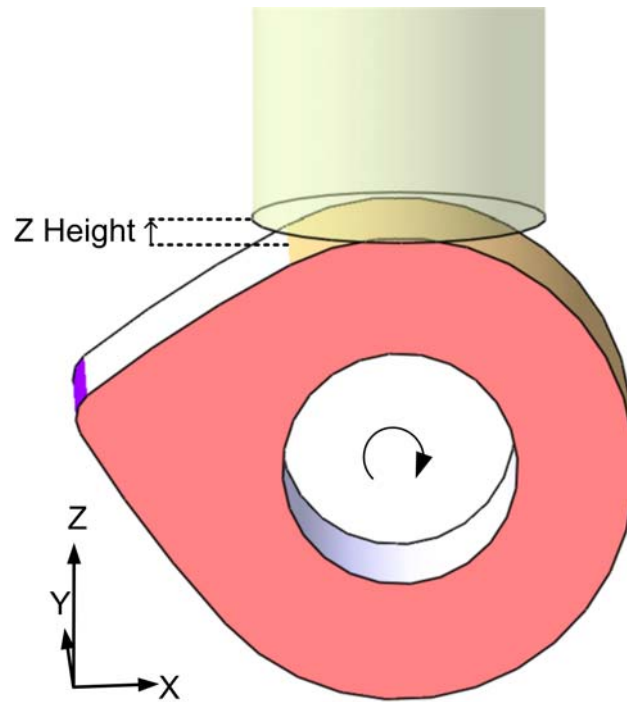
Figure 4-10 Linear measurement to determine Z height
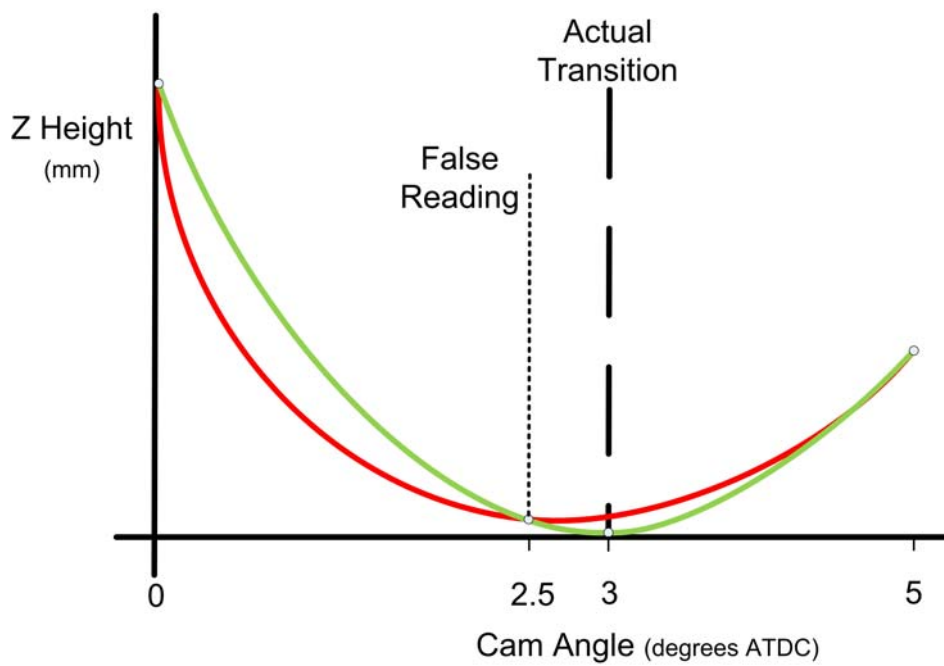
Figure 4-11 Single linear check measurement

A simple directed linear measurement between the base of the cam follower and the boundary between the cam base circle and the leading flank (Figure 4-10) will show how the boundary approaches and recedes from the base of the cam follower but will not definitely identify when the change in contact occurs unless the measurement reduces to zero. The uncertainty is due to the discrete nature of the Z height values. The stationary point of the curve fitted to the data will only give a true indication of when the contact between cam and cam follower changes if one of the Z height values is equal to zero (Figure 4-11). If the minimum Z value is not equal to zero a false reading may result as shown by the red curve in figure 4-11

A more effective method is to attach a virtual gauge to the cam. The virtual gauge, in this instance, consists of a single straight line, one end of which is attached to the boundary between the alternate assembly features. The virtual gauge is, in this instance, aligned tangential to the cam profile at the boundary between the cam base circle and leading flank (Figure 4-12). The check measurement consists of the angle between the base of the cam follower and the virtual gauge. The arrangement of the measurement parameters is important and significantly affects the method capability. In this particular instance the virtual gauge must be tangential to the cam profile at the transition boundary between the two alternate assembly features and the measurement is made from the base of the cam follower to the virtual gauge (Figure 4-12). This gives an angle that is measured in the anticlockwise direction and is thus positive. When the transition boundary makes contact with the cam follower the angle reduces to zero. Once the transition boundary passes the contact with the cam follower the angle is measured in the clockwise direction and is thus negative. This change in sign of the check measurement clearly identifies when the assembly joint features change (Figure 4-13).
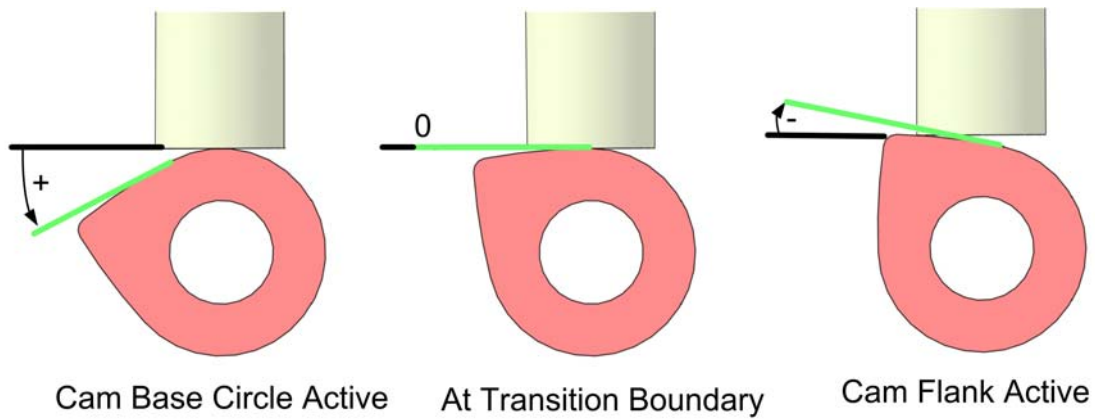
Figure 4-12 Single angular check measurement
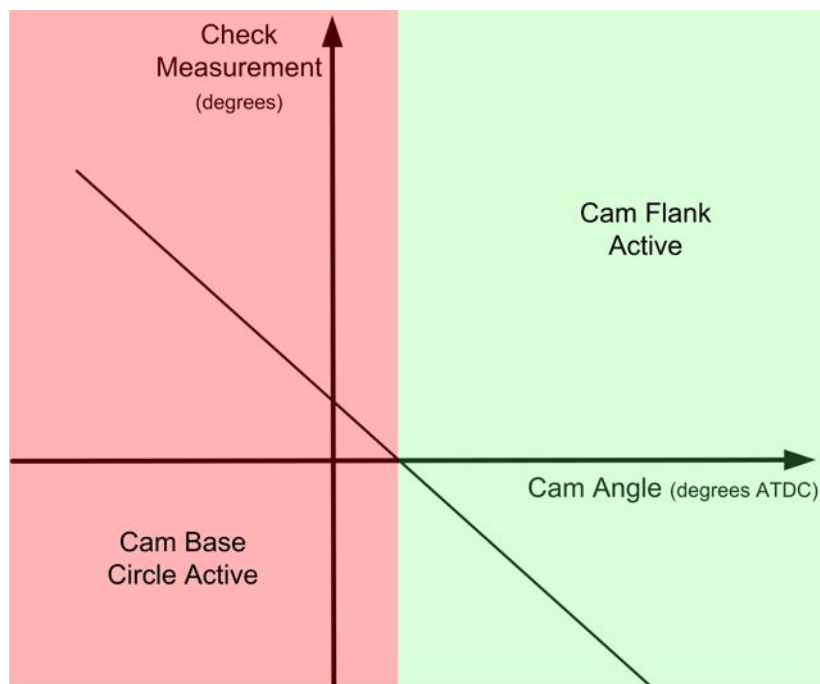
Figure 4-13 Active feature indicated by angular check measurement sign

The advantage of this method is that it is the sign of the angular check measurement rather than the value which identifies which of the alternate assembly features is active. There is none of the uncertainty of the single linear measurement method when evaluating non-zero check measurement values.

The angular check measurement method is an improvement on the single linear measurement method but it is by no means generic. There are circumstances in which the angular check measurement method can be misleading. The cam/cam follower system used previously provides an example of a transition that is beyond the capabilities of the angular check measurement method. Previously the transition between the cam base circle and leading flank was examined. Consider now the transition between the leading flank and the cam toe. Unlike the previous example the transition boundary can now move beyond the footprint of the cam follower base and in limited circumstances move above the cam follower base (Figure 4-14).
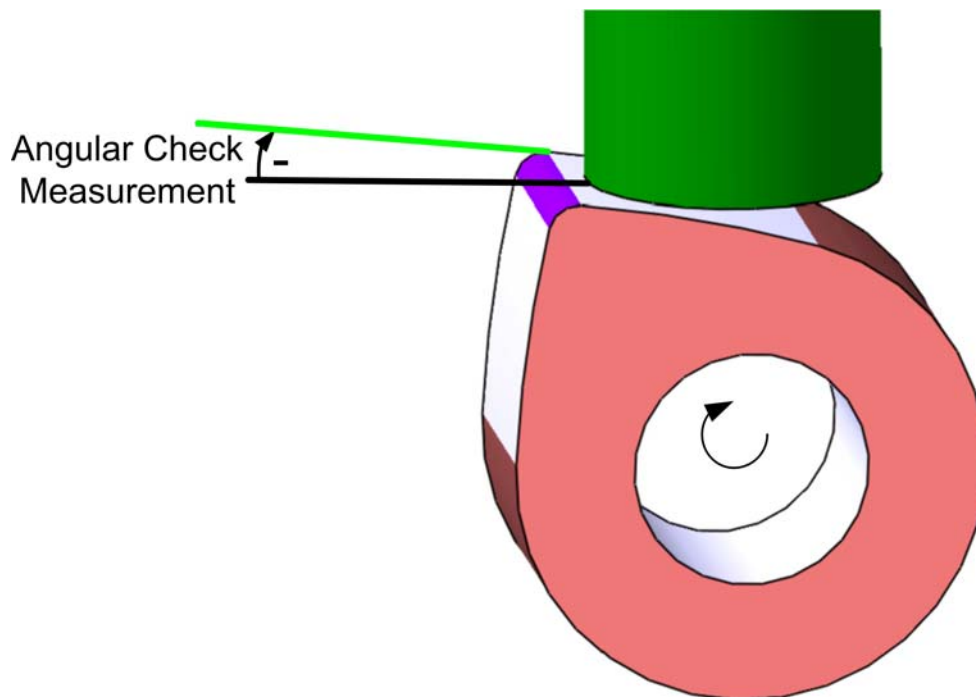
Figure 4-14 Failure of angular check measurement method

This in turn allows the virtual gauge to rotate above the base feature of the cam follower before the cam toe makes contact with the cam follower base feature, giving an inaccurate indication of the transition point.

This issue can be resolved by the use of two directed linear check measurements. The first measure X determines the distance from the leading edge of the cam follower base feature to the cam feature transition boundary parallel to the cam follower base feature (Figure 4-15). The second measure Z determines the distance from the cam follower base feature to the transition boundary, perpendicular to the cam follower base feature.
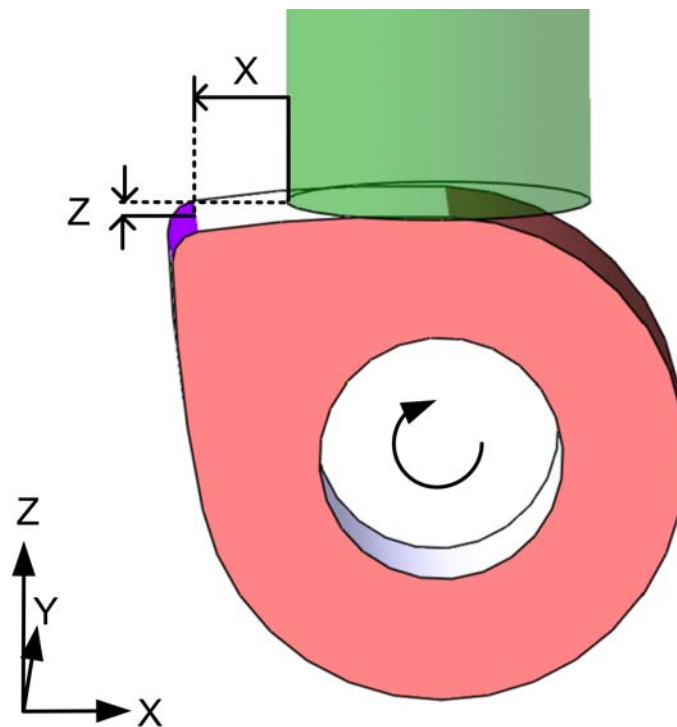
Figure 4-15 Arrangement of dual linear check measurements

When the values for X and Z are plotted for several DVA model configurations the curves shown in Figure 4-16 are obtained. The assembly joint is initially cam follower to cam flank which becomes cam follower to cam toe. This change occurs when both the X and Z curves cut the horizontal axis simultaneously. The two zero values on the Z curve are due to the fact that the boundary between cam flank and cam toe moves outside the footprint of the cam follower base feature for part of the operational cycle. This allows the boundary between cam

flank and cam toe to rise above the level of the cam follower base feature, giving a positive Z value, just before the change in active assembly feature occurs.
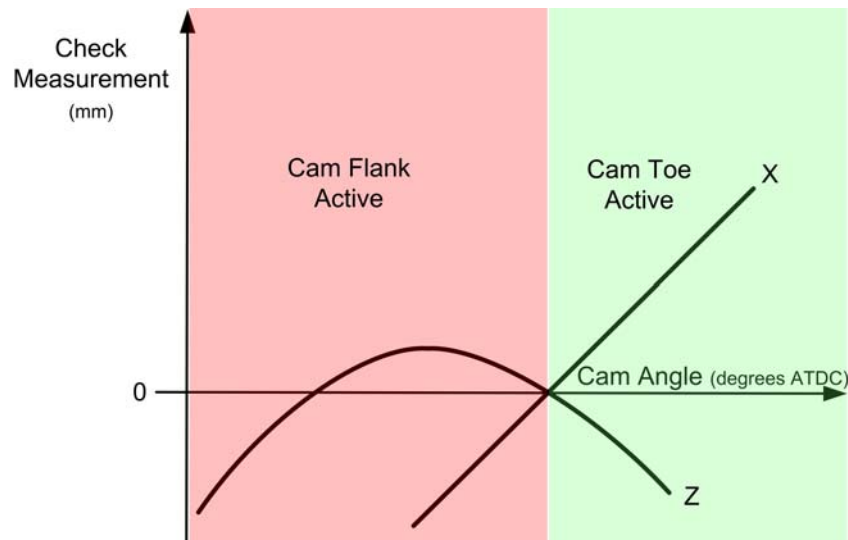
Figure 4-16 Plot of the dual linear check measurements showing the active cam assembly feature and the transition between successive cam assembly features at the point where the X and Z curves simultaneously cut the horizontal axis

4.4 THE KEY FINDINGS OF THE RESEARCH

4.4.1 VIRTUAL GAUGES

One of the conclusions of paper 2 is that many of the virtual gauges used in the analysis of the suspension systems in particular were unnecessarily complex. Figure 4-17 shows two designs for the same virtual gauge. Figure 4-17a shows the minimalist version which is composed entirely of virtual features and as such has no existence in the real world, Figure 4-17b shows the real world version which is composed of actual features to the point that it could be manufactured. Which version of the gauge is used in a DVA model is likely to come down to the personal preference of the analyst and may well be a compromise between the two extremes with the functionality provided by virtual features and sufficient actual features to make the use of the gauge reasonably intuitive. Experience has shown that the compromise

approach has the advantage of making virtual gauges easier to detect and identify as the real features of the virtual gauge can be colour coded. This may be of particular benefit where a previous DVA model is being revisited and the analyst is not familiar with the DVA model.

a) Minimalist version   b) Real world version
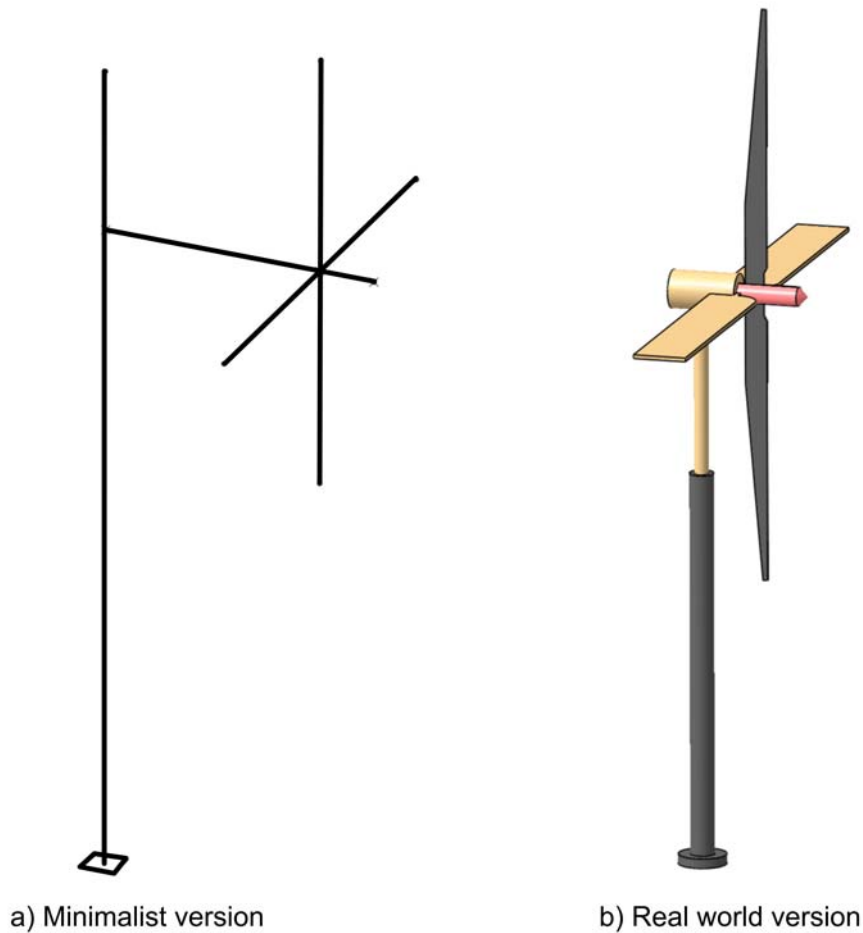
Figure 4-17 Minimalist and real world versions of a virtual gauge

The use of virtual gauges also illustrates another problem. The virtual gauge will almost certainly be either created or selected from an existing toolbox of virtual gauges by the DVA analyst. Virtual gauges by definition enable measurement in some form or other. At some point in the overall product development process the virtual measurements made by the DVA analyst will have to be compared to actual measurements made by a metrologist either to validate the DVA model or for other reasons. It is therefore vital that the two measurement

schemes used are compatible. The virtual gauge shown in Figure 4-17b is multi-component assembly in its own right. The CAD model of the virtual gauge can convey a significant amount of information; the exact amount will depend on whether a minimalist or more intuitive real world design is employed. Similarly the DVA model can convey information regarding the virtual gauge, the type and amount of information being dependent on the specific DVA software in use. However, this requires that both specialists have access to and familiarity with the software which may not be the case. The use of a KCM (Paper 1, Appendix A) avoids most of these problems.

4.4.2 CHECK MEASUREMENTS

The use of check measurements to confirm the correct operation of a DVA model is beneficial when one or more of the assembly joints in the DVA model are not amenable to visual inspection. This is particularly so if one or more of the component parts contain a series of alternate assembly features. Where check measurements are to be deployed in a DVA model it is necessary to consider a number of factors;

- What is the aim of the check measurement

- How will the aim be achieved

- Which check measurement method is to be used

- Is the chosen method compatible with the system geometry

- Will the chosen method produce false or spurious results in addition to the desired result

- Will a single check measurement be sufficient or will multiple measurements be required

A further point that must be taken into account is that each check measurement added to the DVA model will increase the analysis time. Consideration should therefore be given to removing the check measurements from the DVA model once they are no longer necessary. If check measurements are deleted from the final version of the DVA model some external means of documenting the measurement details should be employed, a KCM (Paper 1, Appendix A) may prove suitable for the purpose.

4.4.3 DEPENDANT CONFIGURATIONS

The ability to take a system attribute which is defined in one specific configuration by the relative positions of the adjoining component parts and use that information to analyse the same system across a range of different configurations represents a significant advance in the modelling capability. A two stage modelling process is used for systems containing dependant configurations rather than the more conventional single stage process. As a result it is necessary to consider a number of additional questions when planning the modelling approached to be employed. The additional questions that require answers include;

- Does the system contain any attributes that are indeterminate except in a specific system configuration and do these attributes influence the system characteristics under investigation?

- Can the indeterminate attribute be defined externally across the desired range of configurations in the analysis?

- How does the interdependence manifest itself? Is there a convenient point to isolate the interdependence?

- Which variation sources contribute to the variation of the system attribute?

- Do these variation sources have elements or secondary effects that are not accommodated or negated by the system attribute?

- If such secondary effects exist, are the effects significant and if so how can the effect be incorporated into the second stage of the DVA model?

- Will it be necessary to employ any virtual fixtures, jigs or gauges to determine the extent of the secondary effect and how should these be deployed?

- Will it be necessary to employ a local co ordinate system to aid segregation of the variation source elements. If so will a single local co ordinate system be sufficient?

- What precautions are in place to avoid double counting any variation sources

5 CONTRIBUTION TO EXISTING PRACTICE

5.1 IMPACT ON THE SPONSOR

The primary benefit of this project is that it has allowed the sponsoring company to consolidate and encapsulate the existing know how, experience and tricks of the trade held within the company into recognised new methods that can be adapted for use across a much wider spectrum of industrial applications. The new methods established as a result of this research include;

- The use of virtual fixtures, jigs and gauges to enable components within a DVA model to be located, aligned and measured as required (Paper 2, Appendix B).

- The ability to recognise and account for dependence between different system configurations and thus model components that are set in one configuration but then operate in other configurations of the same system (Paper 4, Appendix D).

- The 3D visualisation of analysis results to condense large numbers of individual results into a single concise display and thus aid the communication and comprehension of those results (Paper 3, Appendix C).

- The mapping and testing of the complexities of the internal and external constraint schema that govern the location of each individual component within the complete assembly (Paper 1, Appendix A).

In the case of the new methods outlined above the latter two also have significant secondary benefits for the sponsoring company. These secondary benefits detailed below have the potential to either, extend the portion of the product development process in which DVA can usefully be deployed, or open up new areas of activity for the sponsoring company.

5.1.1 VISUALISATION OF 3D DIMENSIONAL VARIATION BEHAVIOUR

This method has secondary benefits for the sponsoring company in two areas. The first of these is aiding the flow of information from DVA to the DM process; secondly the use of the method also enhances the type of information that can be made available. Previously most output from the DVA of a system related to the effects of variation on the assembly process. However, with the use of 3D visualisation the possibility exists of showing the effects of variation on the operation and performance of a system and to compare the assembly, operation and performance of different systems. This capability will provide an opportunity for using DVA at the conceptual design stage of the product development process. This is much earlier in the product development process than is currently the case.

5.1.2 KINEMATIC CONSTRAINT MAP

The main function of the KCM is to record and communicate the constraint schema used to construct the DVA model and provide a quick and simple method of checking a system for constraint errors. A secondary benefit of the KCM is that it produces a document that requires little in the way of specialised knowledge to utilise. Such a document can be circulated to all the interested parties in the product development process allowing a consensus to be reached and recorded. Should it be necessary to revisit the DVA model at a later date this document (KCM) will allow the analyst to understand how the original DVA model was constructed

with little or no chance of misinterpretation. These capabilities improve the information flow between DVA and the wider DM process allowing a more cohesive approach to product development.

The remit of the sponsoring company also includes industrial training and it is in this field that the KCM has considerable potential. The construction of an unabridged KCM (Paper 1, Appendix A) is a simple step by step process that gives considerable insight into the constraint of static and kinematic systems. The method is particularly suited to distance learning and/or web based teaching methods as it requires a minimum of equipment, all of which is widely available. The development of a constraint training course based on the KCM would allow the sponsoring company to diversify into web based education and distance learning.

5.2   BENEFIT TO THE WIDER INDUSTRY

The ability to use virtual fixtures, jigs and gauges and to model systems with dependent configurations such as hydraulic tappets will significantly increase the range of mechanical systems that can be successfully analysed and the types of system attribute that can be measured. However, the main benefit to the wider industry will come from those methods that enhance the information flow between the DVA process and the wider DM and CE processes. Methods such as the KCM and the 3D visualisation of analysis data have considerable potential to benefit the wider industry. A major issue facing DVA is one of image. DVA is frequently viewed as a post mortem or fire fighting tool which is only deployed once a problem has occurred. Such problems often occur late in the product development process

when the options for remedial action may be limited and such options as are available are both costly and time consuming. Even the most technically brilliant DVA process, while highly informative, will not be effective if it is applied too late in the product development process. McFadden (2005) noted that the culture that surrounds the DVA process is at least as important as the DVA process itself. Any method or technique that extends the DVA culture and flow of information into other functional areas of the product development process is likely to have a greater general benefit than advanced modelling methods.

For example the KCM provides a document that records the assembly features, constraint schema and key characteristics of a system. This document can form the basis for discussion between the DVA and, for example, quality control (QC) groups to ensure that the system key characteristics can actually be measured in the real world and that the same measurement scheme is used by DVA and QC. Similarly mutually agreeable constraint and datum schema can be established. The KCM assists in the integration of DVA into the wider DM process and, in this instance, extends the DVA culture in the direction of quality control. Enabling an improved information flow between functional groups in the product development process has the potential of changing the image of DVA so that it is no longer seen as a remedial measure but as a preventative measure that should be deployed early in the product development process. This would significantly improve the effectiveness of the entire DVA process and in turn benefit the product development process.

The 3D visualisation of DVA output has the potential to make a significant impact on the way in which DVA is used in the product development process. This method has two areas of influence. The first and most obvious of these is that the combined kinematic and variation

behaviour of a system can be presented in a simple graphical manner. This allows those functional groups within the product development process that are not familiar with DVA to appreciate the affects of variation on the product behaviour and the benefits of DVA. The second area of influence is less obvious but may well have a much greater potential impact on the wider industry. The 3D visualisation method has the potential to compare the effect of variation on the operation and performance of rival products; this would transform DVA from a post mortem utility into a design tool. The use of DVA in the design stages of product development could enable the detection of potential variation based problems before the design freeze when a much greater range of remedial actions are available and any resulting delays or costs are significantly reduced

5.3   RECOMMENDATIONS FOR FURTHER RESEARCH

Several of the methods developed in the course of this research are in a basic form. Further development is required to transform these experimental methods into fully fledged commercial utilities. The method used to visualise analysis data in 3D is a prime example. In its current form the method is very labour intensive. The data entry and the translation of the files into a format compatible with the CAD platform are simple operations but performed manually. Automation of these aspects of the method would significantly improve the usability of the method. Another aspect of the method that requires development is the means used to construct the kinematic and variation behaviour envelopes, at present there are at least two different methods available. The methods need to be evaluated and compared to determine which will result in the most generalised solution and the overall method

standardised on whichever solution gives the most generic method for the 3D visualisation of data.

The method developed to simulate the behaviour of the hydraulic tappet adjuster and the associated dependent configurations has been applied to three different valve train systems. While the valve train systems differ significantly from each other the hydraulic tappet are, except for cosmetic differences, identical. Thus effectively the method is only applied to a single instance of an object with dependent configurations. To evolve the method it will be necessary to apply the same principles to a wider range of systems with dependent configurations where the nature of the dependency differs significantly.

The next stage in the development of the KCM is most likely to be market testing. Several customer preferences need to be established. These include, should the abridged or unabridged version of the KCM be designated as the standard form of the KCM. What should be the standard format of the KCM be, paper or electronic. In the latter case should the KCM be a simple electronic copy of the paper KCM or should it be expanded into a multilevel hierarchical system and how much information should be included in the standard electronic KCM. By defining the standard form, format and content of the KCM the possibility of misinterpretation and the presence of inconsistencies and incompatibilities between KCM's will be reduced.

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Appendix A   **The use of a kinematic constraint map to prepare the structure for a dimensional variation analysis model**

The use of a kinematic constraint map to prepare the structure for a dimensional variation analysis model

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Abstract

Dimensional variation analysis (DVA) models have been used in the manufacturing industry for over 20 years to predict how minor variations in the size, shape and location of the components parts is likely to propagate throughout and affect the overall dimensions, operation and performance of a complete mechanical system. This paper is one of in series of four papers that describe how different techniques can be utilised to aid the creation and application of DVA models. This paper explains the development and use of the kinematic constraint map (KCM) method to prepare in advance the most appropriate structure for a DVA model. The KCM method provides a concise and comprehensive graphical method that, in one document, can identify all the physical constraints that govern the location and (where applicable) the motion of each component within a complete mechanical system. Once complete, the KCM for a mechanical system contains sufficient information to fully define the structure of the subsequent DVA model.

Keywords: Dimensional variation analysis, variation modelling, kinematic constraint mapping.

1. Introduction

Although modern manufacturing methods are becoming increasingly accurate, there is still a small risk that the components produced can vary slightly from the nominal in size and shape. These variations can accumulate and propagate as parts are assembled together into sub assemblies and products, and so maintaining the geometric quality of a product during production and assembly can be problematic. In addition to component part variation, many product variation problems encountered can be traced back to the interface geometry and locating schemes used. The way parts locate relative to each other significantly affects the manner in which variation propagates through the assembly and the effect this variation has on the product key characteristics, (Söderberg et al., 2006).

DVA (dimensional variation analysis) models have been widely used by automotive companies [1] [2] [3] [4] and to a lesser extent by aerospace companies [5], [6] [7] [8] and other manufacturing companies [9] for over 20 years. A DVA model can simulate how minor variations in component size, shape and location are likely to propagate in all six degrees of freedom throughout the assembly and operation of a mechanical system. DVA models have proved very successful in predicting whether or not these minor component variations, when taken collectively, are likely compromise the overall operation, performance or quality of the complete system. The use of a DVA model provides the engineering team with the means to identify potential dimensional variation problems in advance, during the design phase, while there is still time to 'design out' the variation or to devise effective measures to control the variation once in production. As the software used to build DVA models has advanced over the years, in parallel, the DVA users have developed numerous management procedures, application techniques and 'tricks of the trade' to model specific situations [10] [11] [12] [13] [14] [15] [16] [17]. The advances in software combined with the development of new procedures and techniques have substantially increased the capability of the DVA model and the complexity of the systems that can be modelled.

A core feature of a DVA model is the mathematical definition of the internal constraints that govern the size and shape of the individual component plus the external constraints that govern the location and motion of each component within the complete system. Drawing on previous work regarding assembly sequences, geometric tolerances and constraint theory, the aim was to develop a graphical method that could, in one document, hold all the following information:

- The dimension and datum schemes specified for each single component (and sub assembly if applicable)
- The connections between mating components and sub assemblies
- The known or intended sequence of assembly operations, including sub assemblies and the use of fixtures or jigs to locate the components.
- The known or intended operational sequence for systems that move between two or more fixed positions, or across a range or through a cycle of operational positions.
- The level of constraint, full, under or over for each component and sub assembly.

The assembly sequence for a product can be determined using Bourjault's method [18] or the "Cut-set" methods proposed by Baldwin [19]. The advantage of these methods is that the chosen assembly sequence is frequently in the form of a liaison, or assembly sequence diagram. These diagrams provide a simple unambiguous method of communicating the assembly sequence to the analyst.

The variation in size and shape of the component parts can be defined by means of geometric dimensions and tolerances (GD&T [20]. The same standard also defines a simple method of communicating this information to the analyst by means of the feature control frame. Krulikowski [21] notes that one of the benefits of GD&T is that it provides uniformity in drawing specification and interpretation, thereby reducing controversy, guesswork and assumptions.

The manner in which the component parts of an assembly are constrained relative to each other can be defined by means of screw theory [21] [23] [24]. Screw theory uses twist and wrench matrices to express the level of constraint between adjacent component parts. By subjecting these matrices to the appropriate mathematical manipulation [25], it is possible to determine if a series of assembly features that comprise an assembly joint or operation, are under or over constrained.

2. Related graphical methods

Graphical methods are frequently used to describe the relationships between the component parts or the component part features of an assembly. Several of these methods are based on the liaison diagram and include; assembly networks and datum flow chains [26] [27] which denote the presence of a relationship between component parts of an assembly, assembly orientated graphs (Mathieu & Marguet, 2001) and annotated liaison diagrams (Ballu & Mathieu, 1999, Whitney 2004b). The latter two techniques denote a relationship between specific pairs of mating features on adjacent component parts. Assembly orientated graphs [28] combined datum flow chains with liaison diagrams and introduced the concept of the propagation chain which described how variation propagates through the assembly. Three important rules concerning the construction of assembly orientated graphs were also defined by Mathieu & Marguet [28] namely;

- The graph has one and only one root node (base component) which is not located from another component.
- There is always one chain of relationships on the graph going from the root node to another node.
- The graph cannot contain a chain of relationships that loops upon its self. This would imply that the parts locate themselves.

The methods described above are all capable of denoting a relationship between the component parts of an assembly. Some of the methods are able to refine the nature of the relationship and denote the existence of a relationship between pairs of assembly features located on adjacent component parts. This level of detail may be sufficient when constructing simulation models for analysis using Monte Carlo based DVA software. However, when constructing a simulation model for analysis using vector loop based DVA software it is necessary to communicate the exact nature of the relationship between the component parts. This is beyond the present capabilities of the methods described above. A kinematic constraint map capable of conveying the exact constraint scheme to be used in the construction of a simulation model for analysis by vector loop based software would be of considerable benefit.

3. DVA software requirements

The major DVA platforms commercially available make use of two different analysis methods, Monte Carlo simulation and vector loop analysis [29] [30]. The two methods have differing requirements with regard to the amount of information required to build the simulation model. Those platforms based on Monte Carlo simulation are often capable of accommodating a certain degree of under and/or over constraint in the simulation model. Over constraint is often accommodated by employing the principle of constraint redundancy, while under constrained components may be held in their nominal positions. In such platforms, the assembly of two adjacent component parts is achieved by means of a single assembly operation. To complete the assembly operation it is only necessary to define the primary, secondary, etc locations in terms of mating pairs of assembly features one of which is designated as the target feature, the other as the object feature of the mating pair.

The use of a vector loop based analysis method requires additional information to be available when constructing the simulation model. Kinematic constraint of the simulation model is a necessary condition for analysis in vector loop based systems, the exact constraint scheme must therefore be available. The other reason for the increased data requirement is that these systems often treat the assembly of each pair of mating features (assembly location) as an entirely separate assembly operation. One advantage of this approach is that it is unnecessary to specify which mating pair is the primary, secondary, etc location. Several of the software platforms are capable of automatically applying a constraint scheme to each individual pair of mating features based on the type of joint involved (e.g. plane to plane or pin in hole etc).

However, manual intervention is frequently required to attain kinematic constraint of the assembly, as vector loop based platforms rarely utilise the constraint redundancy and nominal position holding techniques common in Monte Carlo simulation based platforms. The manual intervention required when constraining a vector loop based simulation model has the advantage of permitting a specific constraint scheme to be applied to the simulation model as opposed to the automatic, but less than transparent, constraint scheme applied to many Monte Carlo simulation based models.

One area of simulation model construction that is very software specific is the order in which assembly features appear in the simulation model tree. Certain vector loop based platforms use the relative position of an assembly feature in the model tree to define the assembly feature precedence in the simulation model. The assembly features of whichever component part is closest to the base component in the model tree act as the target features in the assembly operation. Both of these requirements can be conveyed by means of an appropriately formatted assembly sequence diagram.

4. Assembly sequence diagram

The main function of the assembly sequence diagram is to convey the assembly sequence. However, by appropriate formatting of the assembly sequence diagram the appropriate assembly feature precedence can be communicated simply and effectively. The preferred format is shown in Figure 1, where the assembly operations run horizontally across the top of the diagram and the component parts run vertically down the left side. Each new component part is added below the preceding part, when read from top to bottom, this will place the component parts in a suitable order to maintain the correct assembly feature precedence between component parts in the simulation model when using vector loop based software.
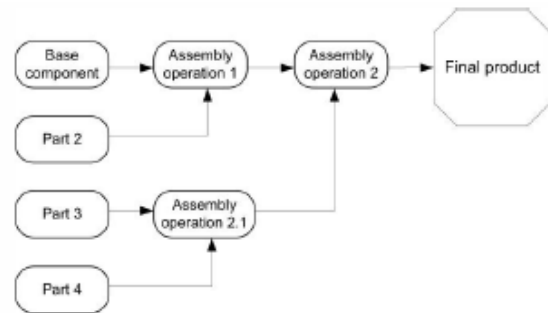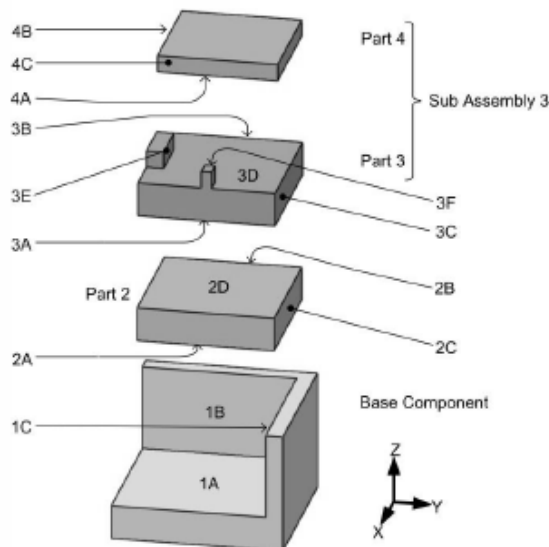
Fig 1 Exploded view of the assembly, showing the assembly features, and the assembly sequence diagram

5. Basic features of the kinematic constraint map

The kinematic constraint map is based on the assembly sequence diagram shown in Figure 1. When constructing the kinematic constraint map the first stage, in a bottom up construction, is to add assembly features to each of the component parts of the assembly sequence diagram. The part icon used in the assembly sequence diagram is replaced by a bounding box for the assembly features of each part. Similarly, the icon for assembly operation 1 is replaced by an assembly operation bounding box that includes the Base Component and Part 2 complete with their assembly features (Figure 2). To differentiate between the two and to signify the change from icon to bounding box the line formats used for the part box and assembly operation box are changed to broken lines.
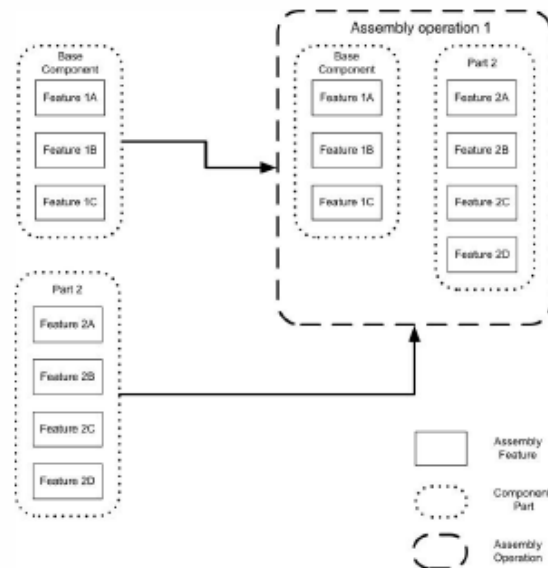
Fig 2 Assembly features for assembly operation 1

The next stage in constructing the kinematic constraint map is to apply the constraint scheme to the assembly operation. In a black and white environment the constraint scheme, which has been pre defined by the design engineer, is applied by means of specially formatted connecting lines (Figure 3). There are six

different connectors, one for each degree of freedom constrained. Each degree of freedom is aligned to, and associated with, one of the principal axes of the assembly global co-ordinate system. The format of the connecting line (Table 1) indicates which of the translation (Tx, Ty, Tz) or rotation (Rx, Ry, Rz) degrees of freedom are constrained by each pair of mating assembly features. When colour is available, the connecting lines may also
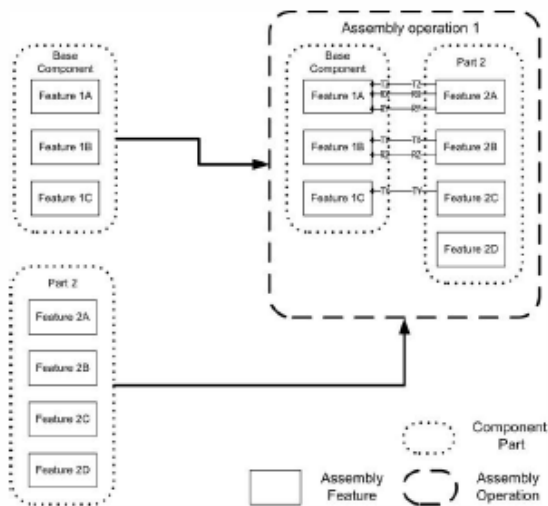
Fig 3 Kinematic constraint map for assembly operation 1

Table 1 Connecting Line Format and Degree of Freedom Constrained

Line Format	Degree of freedom constrained
—TX——TX——TX—→	Translation along X axis
—TY——TY——TY—→	Translation along Y axis
—TZ——TZ——TZ—→	Translation along Z axis
—RX——RX——RX—→	Rotation about X axis
—RY——RY——RY—→	Rotation about Y axis
—RZ——RZ——RZ—→	Rotation about Z axis

be colour coded to assist identification. The arrowhead on each connecting line, points towards the target feature in each pair of mating features. This serves two important purposes; firstly, it identifies the target feature in each mating pair of features and secondly it allows the constraint propagation chain to be examined for inconsistencies in the flow of constraint such as loops or retrograde links.

The inclusion of a second assembly operation in the assembly process (Figure 4) brings into play the concept of the intermediate product, bounding box. This is a temporary feature used during the construction of the kinematic constraint map. The component parts and their associated assembly features contained within the intermediate product, bounding box have been assembled, either directly or indirectly to the base component of the assembly in the preceding assembly operations. Thus, any assembly feature contained within the intermediate

Fig 4 Kinematic constraint map for assembly operations 1 & 2

product bounding box will be a target feature for any subsequent assembly operations.. The size and content of the intermediate product bounding box will change with each successive assembly operation until the last component is assembled when it becomes redundant and is removed from the kinematic constraint map. Figure 4 also illustrates how the presence of a sub assembly in the assembly process is represented in the kinematic constraint map. Part 4 is assembled to part 3 in assembly operation 2.1. The assembly operation is designated 2.1 to indicate that it is a subsidiary assembly operation of assembly operation 2. Sub assembly 1 which results from assembly operation 2.1 is then assembled to the intermediate product in assembly operation 2 to form the final product. The subsidiary assembly operation 2.1 is represented in the kinematic constraint map in exactly the same manner as a normal assembly operation. However the product of the assembly operation 2.1 is contained within a sub assembly bounding box as can be seen in assembly operation 2 (Figure 4). The sub assembly bounding box has two functions; firstly it denotes that parts 3 and 4 were added simultaneously to the assembly as a sub assembly rather than sequentially as individual component parts. Secondly as the sub assembly is assembled to and located from the intermediate product, any assembly feature within the sub assembly bounding box is a potential object assembly feature in assembly operation 2.

Fig 5 Turnbuckle

Consideration must be given to the choice of base component for a sub assembly as the constraint scheme

applied to the "sub" assembly when it is the final product of its own assembly sequence may be different to that applied when it is only a component part of a larger product. Consider a turnbuckle consisting of a central barrel into either end of which is screwed an eye bolt (Figure 5). When assembling the turnbuckles one obvious method would be to clamp the barrel and screw in the eye bolts, one at each end. This would make the barrel the base component of the assembly (Figure 6). However, when the turnbuckle is part of a larger assembly it is

Fig 6 Constraint scheme of the turnbuckle as a final product

almost certain that one of the two eye bolts would be assembled to the intermediate product. In which case, the eye bolt that is assembled to the intermediate product, in this instance eye bolt 2, would locate the barrel and not the other way round see (Figure 7). If the barrel remained as
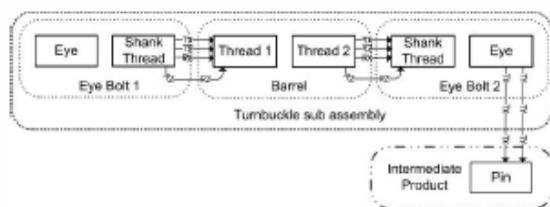
Fig 7 Constraint scheme of the turnbuckle as a component sub assembly in a larger assembly

the base component of the turnbuckle sub assembly, the overall assembly would contain two components that were not located from another component, the base component of the parent assembly and the barrel of the turnbuckle sub assembly. This is not permitted under the rules formulated by Mathieu and Marguet [28]. The situation can be avoided by ensuring that the base component of any sub assembly is one that is involved in an assembly operation to the overall assembly.

1.1.1 Abridged form of the KCM

The kinematic constraint map in Figure 4 shows the assembly of four component parts into a product. A complex assembly such as an internal combustion engine may contain hundreds if not thousands of component parts and would produce a highly complex and expansive KCM. Inspection of Figure 4 shows that all the component parts are repeated three times. Inspection of assembly operation 2 will show that it actually contains all the information from the preceding assembly operations, in this instance assembly operation 1 and assembly operation 2.1. The only information that this final panel of the KCM (assembly operation 2) does not convey in an unambiguous manner is the assembly sequence. Thus by using the assembly sequence diagram in conjunction with the final panel of the unabridged KCM (Figure 8) the same information can be communicated but in a more compact form. It will be noted that the intermediate product bounding box shown in Figure 4 is absent. This is because Figure 8 represents the finished product after the assembly operation has taken place while Figure 4

represents the assembly before assembly operation 2 takes place.

6. Secondary KCM functions

The primary function of a KCM is to communicate the constraint scheme of an assembly. A KCM also has several useful secondary functions. One of these is the ability to check a constraint scheme for under and/or over constraint in a simple but effective manner and identify which degree of freedom is improperly constrained. Consider Part 2 in Figure 8, the part bounding box is intersected by nine connectors representing constrained degrees of freedom. At first sight, this would suggest that Part 2 is over constrained. However, of the nine connectors six are outgoing to the Base Component and three are incoming from Part 3. An outgoing connector is one where the feature to which it is attached is acting as an object feature in the assembly operation. Similarly, an incoming connector is one that is attached to a target feature. Each connector will have an out going and an incoming end. The six outgoing connectors from Part 2 to the base component indicate that Part 2 is the object part in the assembly operation with the base component.
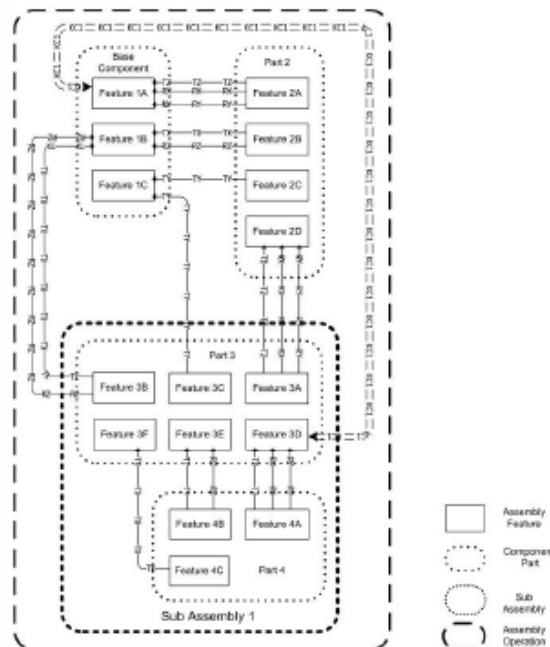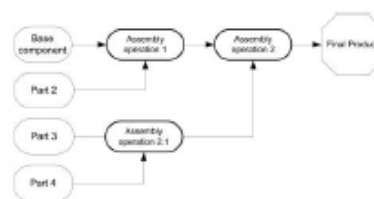
Fig 8 Assembly sequence diagram and abbreviated kinematic constraint map

The assembly sequence diagram identifies the operation as assembly operation 1. Each of the six connectors represents a different degree of freedom indicating that assembly operation 1 constrains all six degrees of freedom and thus Part 2 is kinematically constrained to the base

component. The three incoming connectors to Part 2 are linked to Part 3. The assembly sequence diagram indicates that Part 3 and Part 4 are added to the assembly during assembly operation 2 and that Part 2 has previously been assembled to the base component. Let us therefore consider the part bounding box enclosing Part 3. Six different outgoing connectors intersect the bounding box indicating that all six degrees of freedom have been constrained and Part 3 is kinematically constrained to the base component and part 2 of the assembly. The same information can be derived from the full KCM (Figure 4) without the need to refer to the assembly sequence diagram. The presence of the intermediate product bounding box indicates that all the enclosed components have already been assembled. As above, six different connectors intersect the Part 3 bounding box and in this instance the same six connectors intersect the intermediate product bounding box indicating that Part 3 is kinematically constrained to the intermediate product. The same process can be used to check the degree of constraint between sub assembly 1 and the intermediate product. In Figure 4 six different connector lines intersect both the sub assembly and intermediate product bounding boxes. This indicates that all six degrees of freedom are constrained and sub assembly 1 is kinematically constrained to the intermediate product.

Another useful secondary function of a KCM is the ability to generate constraint propagation chains for any given degree of freedom. Consider the constraint of Feature 3D (Figure 8) along the Z axis (Tz). The constraint propagation chain flows from Feature 3D through the body of Part 3 to Feature 3A and thence by contact to Feature 2D of Part 2 Where it flows through the body of Part 2 to Feature 2A and is transferred by contact to Feature 1A on the Base Component. The constraint of Feature 3D in Tx, however, follows a completely different path via features 3B and 1B. The ability to trace constraint propagation chains can be useful when a component part of the assembly is subject to a late design change. Consider the key characteristic KC1 represented by the double broken connecting line between features 1A and 3D in Figure 6. If the key characteristic consists of a measurement along the Z axis, from the description of the constraint propagation chains given above, if part 2 of the assembly is modified it is likely that the key characteristic will be affected. Whereas if the key characteristic is a measurement along either the X or Y axes it is unlikely to be affected by any modification of Part 2.

Practical experience has shown that the ability to generate constraint propagation chains for individual degrees of freedom is advantageous when examining the constraint schema of complex kinematic assembly systems. Consider the manual drive train of a rear wheel drive motor car that is in gear and with the clutch engaged. All the component parts are either bolted or splined together or meshing gears. The only exception is the clutch where the friction plate is rigidly clamped between the pressure plate and flywheel. The one common factor is that all of these joints between the drive train components constrain rotation. Yet when the crankshaft turns the rear road wheels turn indicating that rotation of the drive train has not been constrained. One method of constraining the rotation of the drive train is to apply the

hand brake. The rotational constraint propagation chain that began at the crankshaft and propagated via the clutch, gearbox, transmission shafts, rear axle and road wheels is extended via the brake disc and calliper through the suspension system to the body in white and on to the base component of the vehicle. When and only when, in this particular instance, the unbroken rotational constraint propagation chain reaches the base component is the rotation of the crankshaft constrained. The ease with which such propagation chains can be detected and traced is one of the advantages of KCM's compared with other liaison diagram based methods.

7. Application

Kinematic constraint mapping has been applied to both simple and complex assemblies and was found capable of accurately recording and communicating the constraint schema and assembly features of the parent assembly. This includes assembly systems that are both complex and conceptually challenging due to the presence of nested simulation models [31] with near identical constraint schema. The differences between the constraint schema while small were significant. The use of a kinematic constraint map allowed these differences to be readily discerned and communicated in an exact and unambiguous manner.

8. Conclusions

The unabridged version of the KCM is capable of unambiguously communicating the assembly sequence, assembly features (both target and object features) and the exact constraint scheme necessary to construct a DVA simulation model and requires only limited technical knowledge to extract this information from the KCM. The abridged version conveys the same information in a much more compact manner but requires greater technical knowledge to extract the information. It is, however, still suitable for deployment in a normal engineering environment. Visual examination of either version of the KCM is capable of detecting over or under constraint conditions whether occurring singularly or simultaneously. It is also possible to determine which degrees of freedom are under or over constrained even in complex kinematic assemblies where the means of constraint is distant from the component being constrained. The main benefit of a kinematic constraint map is that it preserves the original design intent to a greater extent and reduces the number of errors due to misinterpretation, assumption and guesswork regarding the assembly constraint schema. This allows the construction of a more realistic simulation model.

The examples given in this paper clearly show the value of the KCM method as a preparation aid when building a DVA model. In one document a KCM provides a concise and comprehensive graphical method to identify and record the physical constraints that govern the location and (where applicable) the motion of each component within a mechanical system. Once complete, the KCM contains all of the information listed below and this is sufficient to fully define the structure of the subsequent DVA model.

- The bounding boxes show what are internal constraints within a component (or sub assembly) or external constraints between components or sub assemblies.
- Constraint lines connecting features within the same bounding box (single component or sub assembly when sub assembly level dimension and datum schemes are specified) identify the internal constraints that govern the size and shape of that single component (or sub assembly)
- Constraint lines connecting features within different bounding box (between components) identify the connections and the external constraints that govern the location or motion of the components.
- The direction of the constraint lines and the order of the KCM layers show the assembly and the operation sequences for the components.
- The number and type of constraint lines crossing any bounding box show whether the component or sub assembly contained with the bounding box is fully, under or over constrained. One constraint of each type (Tx, Ty, Tz, Rx, Ry & Rz) indicates full constraint. Less than one of each type shows under constraint, more than one of each type shows over constraint.
- The consistency and integrity of the proposed constraints is shown by achieving full constraint and the absence of unexpected constraint loops or constraint dead ends.

9. Areas for further work

The method described in this paper is applied to an assembly system that is aligned to a single global co-ordinate system. In the real world, even comparatively simple assembly systems may be aligned to a global co-ordinate system and one or more local co-ordinate systems. The method of plotting kinematic constraint maps therefore requires further development to allow it to record assembly systems with multiple co-ordinate systems.

The present method of plotting kinematic constraint maps utilises a bottom up approach to their construction. The popularity of the top down approach when assigning key characteristics and constraints, suggests that a top down method of constructing kinematic constraint maps may be beneficial.

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Appendix B **The use of virtual fixtures, jigs and gauges in dimensional variation analysis, simulation models.**

The Use of Virtual Fixtures, Jigs and Gauges in Dimensional Variation Analysis, Simulation Models

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Abstract

This paper describes the use and deployment of, virtual fixtures, jigs and gauges to locate, align and measure features in simulation models used in the dimensional variation analysis (DVA) of assembly systems. The particular example chosen in this paper is a McPherson strut suspension. The correct use of virtual fixtures, jigs and gauges can significantly improve the accuracy and realism of the simulation model and thus the DVA output. In kinematic assembly systems, the use of virtual fixture, jigs and gauges is often essential to produce a working simulation model.

Keywords; Dimensional variation analysis, Dimensional variation behaviour, Virtual fixtures, Virtual jigs, Virtual gauges.

Introduction

At the heart of the dimensional variation analysis, (DVA) process is the simulation model **[1]**. The simulation model is a 3D computer model that, as the name suggests, is intended to simulate the real world behaviour of the assembly system. The simulation model consists of three main layers; the first of these is the CAD geometry. The CAD geometry supplies the nominal size and shape of the component parts in the assembly. Those features of the nominal parts involved in the assembly process are then overlaid with assembly features. The assembly features introduce the variation in size and shape of the component parts due to the manufacturing process and any associated tolerances applied to the component parts. The final layer in the simulation model is the assembly operations. The assembly operations mimic the assembly of the component parts into the final product. This is achieved by mating an assembly feature on one component part with its corresponding assembly feature on an adjacent component part. The mating of the two assembly features constrains one or more degrees of freedom to mimic the assembly process. The analysis process consists of applying assembly level measurements to the simulation model that characterise the assembly characteristics or attributes of interest and analysing the simulation model using appropriate DVA software.

In a simple static assembly, the default orientation of the component parts supplied by the

CAD geometry may be sufficient to analyse the assembly. However, in a large and complex kinematic assembly system with one or more movement ranges the default orientation of the component parts provided by the CAD geometry is unlikely to be sufficient to allow analysis. One method of dealing with movement ranges in kinematic systems is to divide the movement range into a series of incremental steps, each of which is analysed separately. In the case of vector loop based DVA software each incremental step in the movement range becomes a separate configuration of the simulation model and it is only necessary to realign the simulation model to each configuration to permit analysis. This raises the question as to what method will be used to align the component parts of the simulation model into the various assembly configurations required without the need for a complete rebuild of the CAD and simulation models.

Where the assembly is particularly large it may not be viable to construct a simulation model of the entire assembly system. To conserve resources and reduce analysis times the simulation model may consist of only part of the overall assembly. In such circumstances, a method must be devised of locating the simulation model relative to the parent assembly and of mimicking any relevant dimensional variation behaviour of those component parts excluded from the simulation model. To be considered effective a simulation model must be capable of measuring and evaluating the effects of variation on the chosen system attributes or key characteristics. If no appropriate CAD geometry exists to support the necessary measurement features a means of incorporating such geometry into the simulation model must be devised. The method chosen must be capable of accommodating any changes in size or orientation of the required measurement feature in kinematic systems as they progress through their movement ranges.

This paper proposes methods whereby virtual constructs are incorporated into the simulation model of an assembly or partial assembly to enable the location in space of the assembly, the alignment of the component parts into specific configurations and the measurement of the desired system attributes or characteristics where no suitable geometry is present.

Related work

Rosenberg [2] first proposed the concept of virtual fixtures in connection with the telemanipulation of robotic devices. Rosenberg's virtual fixture was a metaphor used to explain the use of abstract sensory information overlaid on top of reflected sensory feedback from a remote environment. In Rosenberg's work, much of the abstract sensory information was haptic feedback but this work established the concept that virtual fixtures could be used to guide and align objects in a virtual workspace. He also noted that while virtual fixtures could be functionally equivalent to real world fixtures, due to their virtual nature, they could occupy the same physical space as other objects in the workspace. Thus, the location and configuration of a virtual fixture is not compromised by the presence of other objects in the workspace. This early work has been developed to produce the modern surgical virtual fixture used in robotically assisted cardiac surgery [3]. These are extremely complex 3D, real time constructs with multi modal feedback systems. While the virtual fixtures used in DVA simulation models are generally less complex, the surgical virtual fixture amply demonstrates the potential of the concept.

Ikonomov [4] described the use of virtual gauges to extend the range of measurements that could be made using co-ordinate measuring machines (CMM's). Ikonomov et al. concluded that the use of virtual gauges closely resembles real gauge measurements and was in accordance with the requirements of the ISO tolerance system. Indeed one of the most

significant current uses of virtual gauges is in the definition of geometric dimensions and tolerances (GD&T) **[5]** in standards such as ASME Y14.5 **[6]**.

Definition and description of virtual fixtures, jigs and gauges

Virtual fixtures, jigs and gauges are virtual constructs that are added to the simulation model of an assembly under investigation. The basic construction of the virtual constructs is similar and it is only the use to which they are put that distinguishes between virtual fixtures, jigs and gauges. A virtual fixture is a construct that is used to locate an assembly in space. A virtual jig is a construct used to align the component parts of an assembly into a specific configuration and a virtual gauge is a construct that enables any form of measurement. These constructs consist of features of size, mathematical features or more commonly a combination of the two.

Virtual fixtures

A virtual fixture operates in a similar manner to a real world fixture in that it locates one or more component parts in space relative to a predefined co-ordinate system. Since the component parts are located from the virtual fixture the virtual fixture usually forms the base component of the simulation model. In such cases the coordinate system of the virtual fixture will form the global co-ordinate system of the simulation model. As virtual fixtures are most commonly used where only part of the parent assembly is being simulated the global origin of the virtual fixture may be at some considerable distance from the CAD geometry of the virtual fixture. It is important to maintain the same relationship between the global origin and any assembly features of the virtual fixture as exists between the global origin and the same assembly features in the parent assembly. Another important function of the virtual fixture is to mimic the dimensional variation behaviour of component parts that are not present in the simulation model of a partial system. Consider the virtual fixture shown in Figure 1. In this instance, the virtual fixture mimics the dimensional variation behaviour of the body in white, which is not present in the model.

The virtual fixture locates the upper end of the shock absorber, the steering rack and the inboard end of the suspension arm in space. This is achieved by means of features of size (holes, surfaces etc) contained within the virtual fixture. As the locations are features of size, geometric dimensions and tolerances can be applied to them to mimic the dimensional variation behaviour of the body in white. The appearance of the solid geometry that makes up the remainder of the virtual fixture is largely unimportant provided it does not interfere with, in this instance, the suspension system. However, the geometry of the virtual fixture does perform an important function, particularly so when vector loop based analysis software is employed in that it allows closure of the vector loops without which analysis cannot take place. In this instance the virtual fixture locates the components of the suspension system and mimics both the geometric variation and constraint behaviour of the absent body in white.

Virtual jigs

Virtual jigs are frequently used in conjunction with kinematic assembly systems. A kinematic system has, by definition, one or more continuous movement ranges. One method of simulating the behaviour of such systems is to divide each movement range into a series of incremental steps each of which is represented by a different configuration of the simulation model. Virtual jigs are then used to align the simulation model into each of the configurations.

Consider the suspension system shown in Figure 1. This has two movement ranges, suspension travel and the steering lock. The ride height of the system is set by means of the suspension virtual jig. This consists of a series of mathematical planes set parallel to the ground surface. The planes are, in this instance, attached to the virtual fixture, which acts as a carrier for the mathematical features. This is permissible as the virtual fixture is the base component of the assembly and thus the only component that does not move relative to the global origin of the assembly. A mathematical point attached to the outboard end of the stub axle is then assembled in turn to each of the planar features to set the ride height. Kinematic systems are by definition under constrained, a secondary function of the virtual jig is to supply some of the additional constraint required to achieve kinematic constraint of the system. This is particularly important where a vector loop based simulation is employed as kinematic constraint is a necessary condition for simulation. The second virtual jig shown in Figure 1 is the steering jig this is similar in nature to the first jig in that it consists of a series of mathematical planes attached in this instance to the steering rack. In this instance, the mathematical planes are assembled to a feature of size, namely the end face of the boss on the virtual fixture through which the steering rack passes. If any tolerances are applied to the face of the boss they will prevent the virtual jig from exactly aligning the system to the required configuration. Virtual jigs can also be used to align sub assemblies where one of the parameters is implicitly defined **[7]**. In such circumstances, the resultant virtual jig tends to be both large and highly complex.

Virtual gauges

A virtual gauge is any construct that enables measurement within the simulation model. A virtual gauge is usually employed to provide a measurement feature when no suitable CAD geometry exists. Consider the suspension system shown in Figure 1. To determine the effect of variation on the caster angle of this system it necessary to measure the angle between the vertical plane (ZY) and the steering axis when viewed from the side of the vehicle. The first problem in achieving this is that there is no suitable geometry linking the upper and lower pivot points of the McPherson strut and thus no suitable measurement feature. In an attempt to resolve this issue a mathematical line feature was added to the CAD geometry as a measurement feature. For convenience, the line feature was attached to the suspension knuckle, which acted as a carrier. Construction of the simulation model entailed constraining the measurement feature to the upper and lower pivot points of the steering axis to ensure that the measurement feature accurately followed any movement of these points. However, this resulted is a significant over constraint of the suspension knuckle in the simulation model precluding analysis by vector loop based DVA software. To allow analysis the constraints were removed. However, visual inspection of the simulation model in various configurations raised doubts as to the validity and accuracy of this approach. To resolve the issue a long needle like component part was constructed. This acted as a carrier for a mathematical line feature and a mathematical point feature at one end. The line feature was co-incident with the centre line of the needle geometry. The component was added to the simulation model, the point feature was assembled to the upper pivot point of the McPherson strut in the simulation model and the three translational degrees of freedom constrained. The line feature was assembled to the lower pivot point of the strut and rotation about , and translation perpendicular to, the line feature were constrained. The needle like component and the mathematical features it carried formed the virtual gauge, which was kinematically constrained in the simulation model without over constraining the simulation model (Figure 1). This approach was able to accommodate the changes in size and orientation of the

steering axis in the various simulation model configurations.

When constructing the solid geometry of a virtual gauge, simplicity is the key as it rarely plays a role in the measurement process. Consider the needle shaped virtual gauge described previously. Despite the solid geometry being a simple cylinder that tapers to a point at one end the solid geometry is unnecessarily complex. A superior version of the gauge would consist of a short cylinder with a mathematical line coincident with the centre line and extending a considerable distance beyond the ends of the cylindrical solid geometry. The needle shaped version has the disadvantage that when attempting to select the mathematical point at the pointed end, the point feature is co-incident with both the end of the mathematical line and the vertex of the tapered solid geometry, which may cause considerable uncertainty as to which of the three features, has been selected. The vertex of the tapered solid geometry is a feature of size and as such may be subject to geometric dimensions and tolerances. If selected inadvertently it may cause a significant error in the measurement. For this reason, all tolerances applied to features of size in a virtual gauge should be set to zero unless explicitly specified to the contrary.

Where suitable CAD geometry does exist, the measurement features may be added directly to the extant geometry provided it does not cause over constraint of the simulation model. Consider measurement of the toe-in of the suspension system shown in Figure 2 by using the difference in distance from the longitudinal centre plane of the vehicle to a point on the leading and trailing flanges of the wheel. In this instance, the measurement feature consists of a mathematical line attached to the wheel rim, which commences at the leading flange, passes through the wheel centre and terminates at the trailing flange. Due to the effect of the camber angle on the system the measurement feature must be horizontal if it is to give an accurate measurement. When the measurement feature was added to the wheel CAD geometry it was set horizontal but subsequent assembly operations have influenced this alignment and it is not reliable (Figure 2). Fortunately, the wheel rim is inherently under constrained and free to rotate about the stub axle. Since kinematic constraint is a necessary condition for analysis in a vector loop based DVA simulation, constraint of the rotational degree of freedom can be used to align the toe-in measurement feature without over constraining the simulation model. This can be achieved by assembling the linear measurement feature to the appropriate planar feature of the virtual suspension jig. This ensures that the measurement feature is both horizontal and passes through the spin axis of the wheel

In this particular instance the measurement feature functions as both a virtual jig; in that it is used to align the wheel rim to the horizontal, and as a virtual gauge in that, it enables measurement of the toe-in. If, however, the measurement feature, or the virtual suspension jig, were excluded from the simulation model for any reason it would be necessary to constrain the rotation of the wheel by some other means. For this reason it is considered preferable to use a virtual gauge that has its own dedicated solid geometry (Figure 3) rather than rely on the solid geometry of an extant component part of the assembly. The advantages are that no modification to the constraint scheme of the parent assembly is required regardless of whether the dedicated solid geometry of the gauge is present or not. Secondly the virtual gauge and dedicated solid geometry is entirely self contained and could be reused as part of a toolbox of ready made virtual gauges. The virtual gauge shown in Figure 3 consists of a sub assembly of four component parts. The four component parts are the base, which sits on the ground plane, The upper section which incorporates the horizontal blade, the vertical blade and the central pin. The central pin is assembled to a point on the outboard end of the stub axle, which sets the height of the virtual gauge and ensures that both the horizontal and vertical blades are co-incident with the wheel centre. The vertical and horizontal blades are each assembled to the

wheel rim by means of two point contacts. In the case of the horizontal blade this causes the entire virtual gauge to rotate about the vertical axis until the horizontal blade makes contact with the wheel rim. This has the advantage of ensuring that the vertical blade remains perpendicular to the face of the wheel rim. The vertical blade pivots about its centre to align to the wheel rim without affecting the alignment of the rest of the virtual gauge. In this particular instance the vertical blade is used to provide a measurement feature to enable measurement of the camber angle while the horizontal blade is used in the measurement of the toe in or steering angle. One advantage of using the more complex sub assembly for a virtual gauge is that once it is assembled to the suspension system it is self-adjusting regardless of how the simulation model is subsequently configured.

The design of the virtual gauge while more complex than strictly necessary does allow it to be reused on any suspension system of approximately the same physical size. The additional complexity of the design also means that the virtual gauge more closely resembles a real world artifact and thus its use is more intuitive than a minimalist design.

Validation

The validation of the use of virtual fixtures, jigs and gauges is largely empirical in nature. If the simulation model in which the virtual constructs are deployed exhibits the same dimensional variation behaviour as the full assembly and can be aligned into the desired configuration(s) without compromising the assembly constraint scheme while permitting accurate measurement of the desired system attributes then the virtual fixtures, jigs and gauges deployed are considered to be valid.

Consider the McPherson strut suspension system shown in Figure 1. If the complete vehicle were modelled then the suspension system would, in this instance, make contact with the body in white at the upper and lower mounting points and the steering rack. It is only at these points that variation can propagate between the suspension system and the body in white. Thus provided that the virtual fixture contains the same assembly features, tolerances and is aligned to the same global coordinate system the dimensional variation behaviour of the virtual fixture will be indistinguishable from that of the body in white regardless of the overall size and shape of the virtual fixture.

In the real world when the driver turns the steering wheel to go round a corner, the motion is transmitted via various components to the steering rack. Depending on the input from the steering wheel the steering rack will either move to the left or right. The virtual steering jig shown in Figure 1 produces exactly the same result in that the position of the steering rack can be set to any desired position to the left or right of its default position.

Virtual gauges are simpler to validate as they contain a measurement feature. Consider the virtual gauge shown in Figure 1. The virtual gauge was added to the simulation model to enable measurement of the castor angle of the suspension system. The virtual gauge performed two functions; it introduced a measurement feature that linked the upper and lower pivot points of the McPherson strut, which defined the steering axis of the suspension system. Once defined the inclination of the steering axis to the vertical when viewed from the side of the vehicle could be measured to give the castor angle [8]. The positioning of the virtual gauge and its ability to follow changes in the orientation of the suspension system can be validated by the simple expedient of including two check measurements in each configuration of the simulation model. These measurements simply measure the distance between the measurement feature of the virtual gauge and the upper and lower pivot points of the McPherson strut which should be zero.
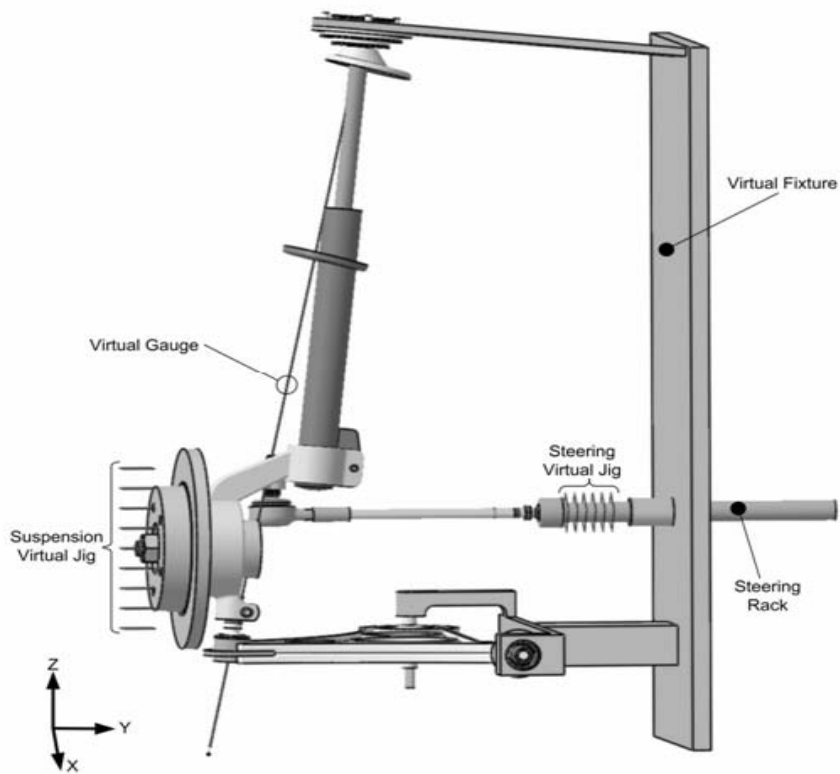
Figure 1 Examples of virtual fixtures, jigs and gauges as applied to a McPherson strut suspension system.
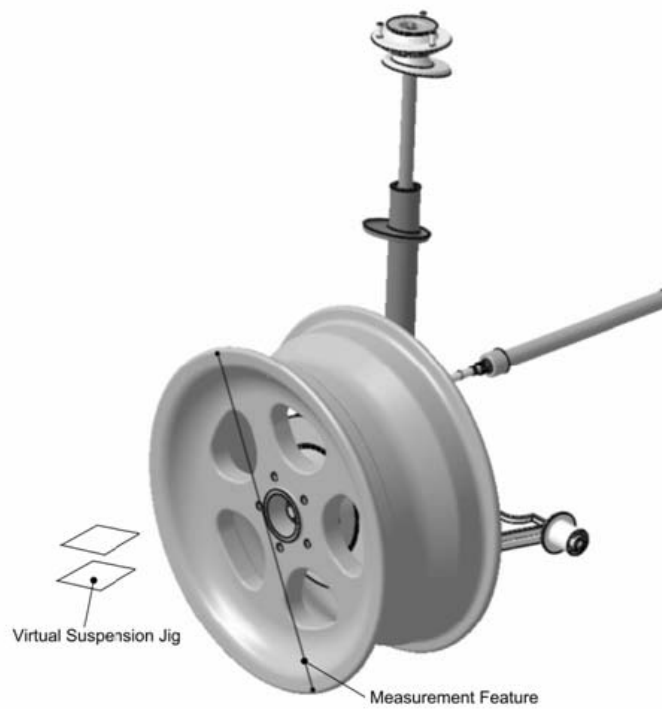
Figure 2 Measurement feature attached to the wheel rim
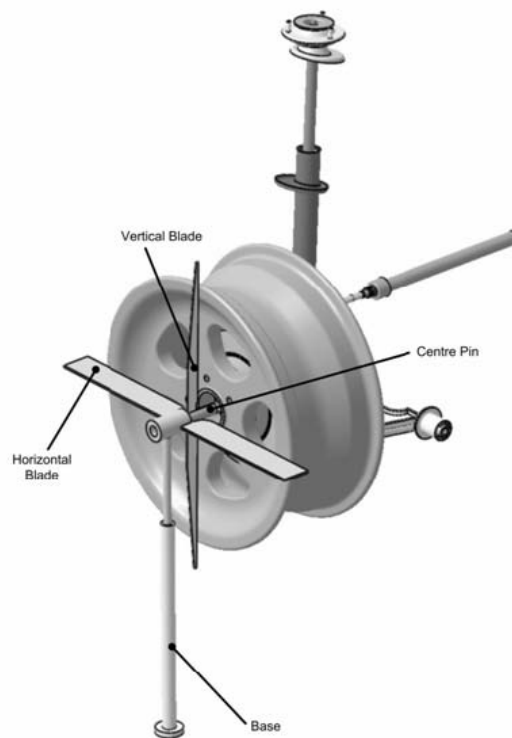
Figure 3 Virtual gauge with dedicated solid geometry

Conclusions

The maximum benefit from the deployment of virtual fixture jigs and gauges is obtained when simulating complex kinematic systems with multiple movement ranges. Little or no benefit may result from utilising virtual fixtures and jigs in the simulation of simple static assemblies. Virtual gauges may be used to advantage in any type of assembly where there is no suitable solid geometry to provide measurement features in the simulation model. This is particularly so if the measurement feature is subject to changes in size and orientation.

Virtual fixtures often constitute the base component of a simulation model as their function is to locate assembly components in space. The location and orientation of the virtual fixture co-ordinate system is thus of importance as it may constitute the global co-ordinate system for the entire simulation model. Virtual fixtures frequently contain features of size as these can be used to mimic the dimensional variation behaviour of components that have been excluded from the simulation model on grounds of size or complexity.

The mathematical features of a virtual jig may be attached directly to an existing component part as they are used to align inherently under constrained parts to a fixed configuration. The unconstrained degrees of freedom are available to effect alignment of the simulation model. The preferred format for a virtual gauge consists of limited but dedicated solid geometry that acts as a carrier for one or more mathematical features. The mathematical feature(s) acting as the measurement feature(s) in the simulation model. Where virtual gauges contain features of size care must be taken to ensure that these do not act as unintentional sources of variation in the simulation model. This format of virtual gauge prevents the

simulation model becoming over constrained and the self contained nature of the virtual gauges also has the potential for reuse in other simulations

Further Work

Many of the designs used for the virtual constructs have been unnecessarily complex and reflected real world practices. This is especially so with regard to virtual gauges. The possibilities of reducing design complexity while retaining functionality needs to be explored further with a view to producing minimalistic but fully functional virtual gauges. The impact such minimalist designs would have on the re-use of virtual gauges also needs to be investigated.

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Appendix C   **A method to visualise the 3D dimensional variation behaviour of kinematic systems**

A Method of Visualising the 3D Dimensional Variation Behaviour of Kinematic Systems

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Abstract

At present, the output from the dimensional variation of an assembly is often in the form of a statistical process control chart or histogram. Such output, while informative, is not particularly suited to the evaluation of dimensional variation behaviour in complex kinematic assembly systems. This paper presents a graphical method of visualising the 3D dimensional variation behaviour of any chosen point feature in a kinematic assembly system by the use of 3D bounding surfaces. The bounding surfaces describe a volume such that there is a known probability that the chosen point is contained within the volume. The bounding surfaces and the volume they describe visualise the dimensional variation behaviour of the chosen point feature as it traverses one or more of the kinematic assembly system's movement ranges. This paper demonstrates that the visualisation allows the dimensional variation behaviour of the chosen point to be presented in a compact and readily comprehensible manner that enables evaluation of the analysis output within the context of the original simulation model geometry. An example of the method applied to a simple kinematic system is also shown.

Keywords; Dimensional variation analysis, Dimensional variation behaviour, Information visualisation.

Introduction

A kinematic assembly is one that has one or more continuous ranges of movement. The assembly can occupy any position between the limits of the movement range(s). A frequently used method of simulating the dimensional variation behaviour of kinematic systems is to divide the movement range(s) into a series of increments.

Where the assembly system only has a single movement range then each increment will form a separate configuration of the simulation model. Where there are two or more movement ranges each unique combination of increments will form a separate simulation model configuration. The number of simulation model configurations will rise exponentially with the number of movement ranges in the assembly system. Each of the simulation model configurations is analysed separately. The method is analogous to the individual frames in a cinema film which when recombined give the impression of movement on the screen. One disadvantage of the method is that it produces large volumes of numeric data.

Juster et.al. [1] make the point that tolerance analysis systems often produce data in the form of histograms that require interpretation by an experienced analyst to establish the dimensional variation behaviour of the assembly. A basic tenet of information visualisation [2] is that information structures are more easily interpreted, if they can be visualised. Hansen and Johnson [3] make the point that data visualisation is an indispensible part of the scientific discovery process. The benefits of visualising medical data such as ultrasound, CT and MRI scans are well known. However, the data produced by the scan still requires significant computer manipulation using dedicated software to produce the visualisation [4]. The visualisation of the effects of variation on the perceived quality of products and in particular automobiles is well established [5] as are the financial benefits. Unlike the real time dynamic visualisations found in medicine the visualisation of perceived quality is applied to essentially static systems. Given the current economic climate manufacturing companies are reluctant to purchase dedicated software unless the financial benefits have been demonstrated. Thus, a method of visualising the dimensional variation behaviour of a kinematic assembly that only utilises software one could reasonably expect to find in a company undertaking dimensional variation analysis would be beneficial.

Depending on the assembly under investigation, visualisation of the analysis output may not be necessary, it may be possible to interpret the information by the simple expedient of collating and tabulating the analysis results from each individual configuration. However, where the intrinsic information is represented by more than one parameter such as polar or Cartesian co-ordinates it may still be difficult to interpret the data. It would of course be possible to visualise such data as a 3D graph. Unfortunately, in doing so the data is completely divorced from the assembly system, making it difficult to relate the data to the kinematic assembly that produced it. What is required is a method of introducing the analysis data back into the simulation model of the assembly system that produced it. This would allow the visualisation and evaluation of the dimensional variation behaviour within the context of the original kinematic system. Such a method would also in part alleviate a limitation of most CAD systems [1] which normally only display nominal geometry.

This paper proposes just such a method whereby post processed dimensional variation analysis data is introduced into the simulation model of the assembly system. The data is used to create bounding prismatic surfaces that visualise the dimensional variation behaviour of a specific point in the assembly within the context of the simulation model.

Visualisation of the effect of variation on a point

The proposed method is intended to visualise the dimensional variation behaviour of a selected point in the assembly. The dimensional variation behaviour of an assembly is determined by means of dimensional variation analysis (DVA) software. This simulates the propagation of variation through an assembly system and determines the effect of this variation on selected assembly level measurements or system attributes within the assembly. The first stage of the process is to select the point of interest and locate it in 3D space.

Point location

A point feature P within the simulation model is selected, or if necessary added, which is appropriate for the dimensional variation behaviour to be visualised. The point feature is located in space by means of three directed assembly level measurements. As the feature is a mathematical point it does not require orientation. The three measurements commence at the global origin, and are aligned to the principal axes (X, Y, Z) of the global co-ordinate system (Figure 1).

    The values of the directed measurements give the Cartesian co-ordinates of the point P. The global origin is used as the reference feature for the measurement as it is one of the few features in the assembly system known to remain fixed. Potentially, most assembly features in the simulation model are subject to relative motion as the assembly traverses its movement range and/or due to the effects of component part variation.

    The dimensional variation behaviour of the point P, as the assembly traverses its movement range, is simulated by dividing the movement range into a series of incremental positions. Each of the positions represents a simulation model configuration, which is analysed separately. The incremental direction, in which the model configurations increment is aligned to the axis of the global co-ordinate system that contains the largest component of the movement range direction, in this instance the Z axis. The planes in Figure 1, which, represent the incremental configurations, are aligned normal to the incremental direction, and positioned by fixing, in this instance, the Z co-ordinate so as to pass through the point of interest, P. The position of the point P can move within the incremental plane but by definition, it cannot move out of the plane as, in this instance, the Z co-ordinate is fixed to align the simulation model into the correct configuration. The X and Y co-ordinate values are defined by the dimensional variation behaviour of the point P.
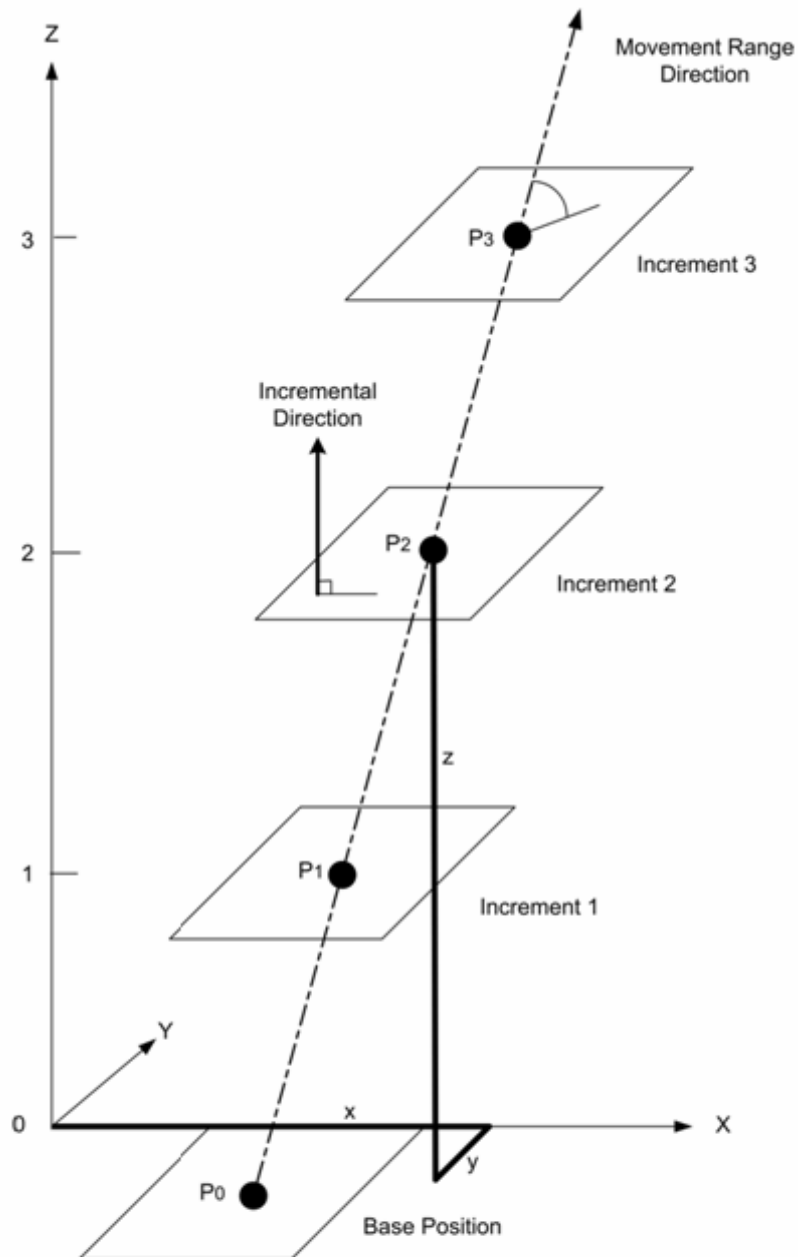
Figure 1 Relationship between the movement range direction, incremental direction and the point P

Generation of variation points

Since the simulation model takes into account, a significant number of input variables when simulating the system behaviour the output is usually in the form of a normal distribution curve. The X and Y co-ordinates are represented by a mean value (μ) and the standard deviation (σ) of the variation distribution curve. The probability that the X co-ordinate has a value of X less than or equal to x is given by the cumulative

distribution function $\Phi(x)$ (equation 1)

$$\Phi(x) = \frac{1}{\sqrt{2\pi\sigma_x^2}} \int_{-\infty}^{x} e^{-(x-\mu_x)^2/2\sigma_x^2}$$ (eq 1)

By the appropriate use of integration limits (lower limit x, upper limit ∞) equation 1 can give the probability that X is greater than or equal to x. This is known as the Q function where,

$$Q(x) = 1 - \Phi(x)$$ (eq 2)

Thus for any given value X or Y it is possible to calculate the probability that X>x or Y>y. However, the probability that the point (X, Y) lies outside the point (x, y) is given by equation 3.

$$P(X,Y) = Q(x)Q(y) \text{ where } X > \mu_x, Y > \mu_y$$ (eq 3)

To determine the probability that the point (X, Y) lies outside the locus of points on the incremental plane requires three additional equations one for each quadrant of the incremental plane.

$$P(X,Y) = Q(x)\Phi(y) \text{ where } X > \mu_x, Y < \mu_y$$ (eq 4)

$$P(X,Y) = \Phi(x)Q(y) \text{ where } X < \mu_x, Y > \mu_y$$ (eq 5)

$$P(X,Y) = \Phi(x)\Phi(y) \text{ where } X < \mu_x, Y < \mu_y$$ (eq 6)

Consider equation 3, if P(X,Y) is fixed at a value of, for example, 10^{-3} then there is a 1:1000 possibility that the point (X, Y) lies outside the point (x,y). The probability that each individual co-ordinate exceeds a given value is controlled by three variables (eq 1), μ the mean value of the variation distribution, σ the standard deviation of the distribution and the value of x or y. By substituting these parameters into equation 3 equation 7 is obtained.

$$Q(x, \mu_x, \sigma_x)Q(y, \mu_y, \sigma_y) = 10^{-3}$$ (eq 7)

The output from the dimensional variation analysis provides values for μ_x, μ_y, σ_x and σ_y. Thus, y can be determined for any given value of x contained by the variation distribution curve. However, this solution only applies where both x and y are greater than their mean values. Co-ordinate pairs for the remaining three quadrants of the incremental plane are generated by solutions based on equations 4, 5 and 6.

Visualisation of generated point data

If a series of co-ordinate pairs generated by the above method is plotted on the incremental plane (Figure 2) a bounding line is produced. By generating series of co-

ordinate pairs for each increment in the movement range, linking them both within the incremental planes and between the incremental planes it is possible to generate a 3D bounding surface (Figure 2). In this instance, the probability that the point (X, Y) lies outside the bounding surface is 10^{-3} but bounding surfaces can be generated for any probability value or series of probability values. The points so generated could be used to plot a 3D graph as in Figure 2. However, by importing the points into the CAD platform used to create the original CAD geometry a 3D part can be created and then added to the original simulation model.
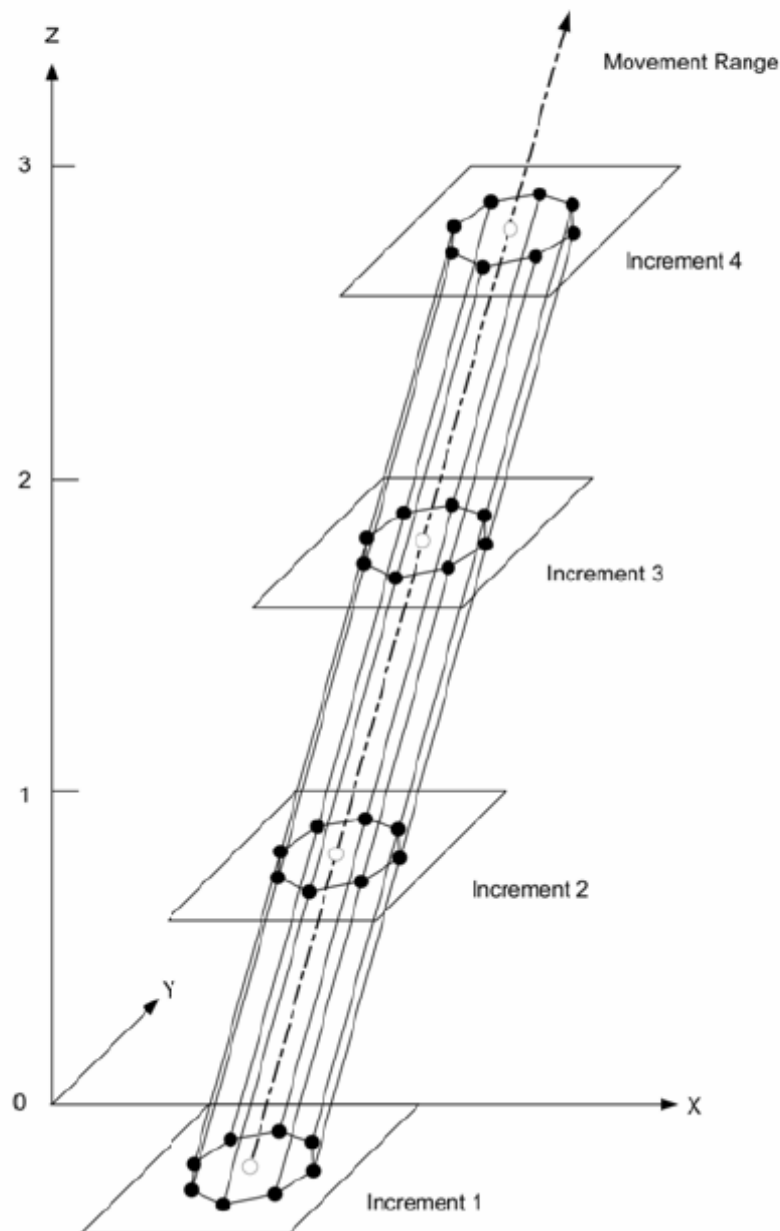
Figure 2 Creation of a prismatic bounding surface

Figure 3 shows a simple kinematic assembly with a single movement range. A point P is located at the right hand end of the round axle in Figure 3. The output data from the dimensional variation analysis of the three directed measurements along the global axes of the assembly to point P is given in Table 1.
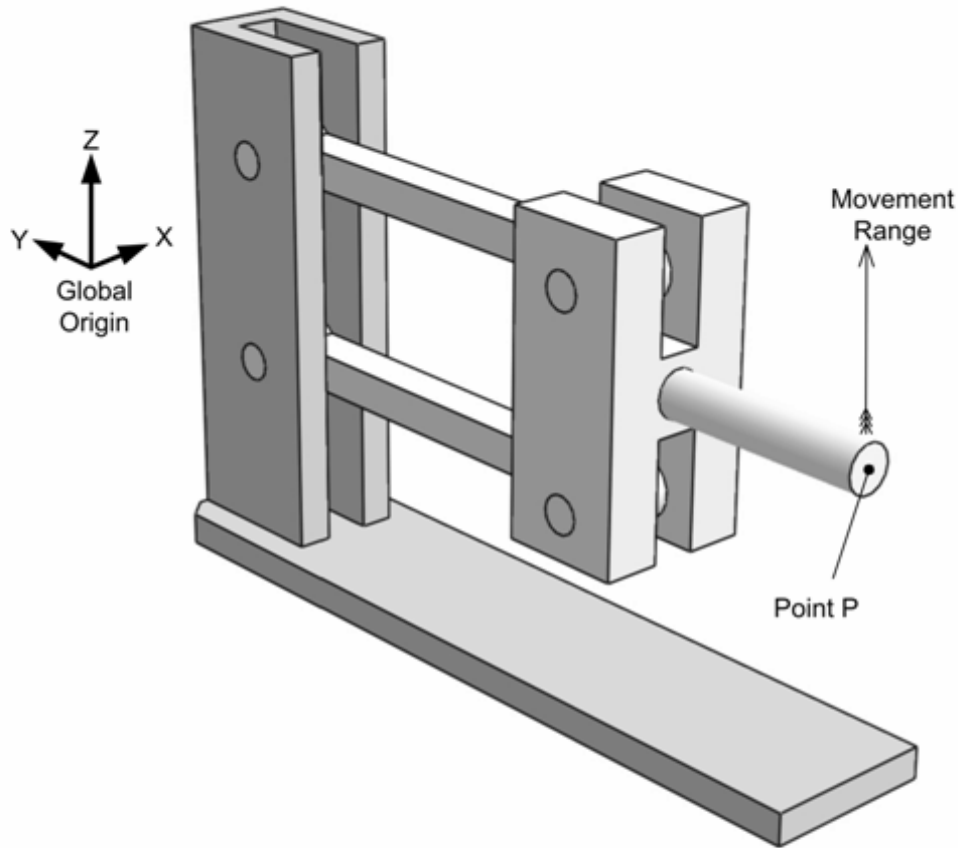
Figure 3 Simple kinematic assembly

Table 1 dimensional variation analysis output

μ_x	σ_x	μ_y	σ_y	μ_z	σ_z
35	1.84926	-281.417	0.156651	-30	0
35	1.72479	-283.970	0.139571	-20	0
35	1.61848	-285.807	0.126759	-10	0
35	1.52890	-286.954	0.118029	0	0
35	1.45542	-287.427	0.113403	10	0
35	1.39808	-287.232	0.112983	20	0
35	1.35752	-286.368	0.116858	30	0
35	1.33486	-284.821	0.125085	40	0
35	1.33167	-282.570	0.137768	50	0

Table 1 shows that the standard deviation of the X co-ordinate σ_x is approximately one order of magnitude larger than that of the Y co-ordinate σ_y but the dimensional variation behaviour of the point P is obscured by the need to combine the three co-ordinates. The Z co-ordinate has, in this instance, a standard deviation σ_z of zero because the Z co-ordinate of the point P is fixed to align the simulation model configurations. The mean values of Y and μ_y are all negative to ensure directional consistency between the global axes and the measurement direction. The apparently fixed mean value μ_x of the X co-ordinate is purely coincidental and due to the fact that the assembly is not designed to move in the X direction. Thus any movement in X is due to the dimensional variation behaviour of the assembly which is symmetrical about, but does not in this instance affect, μ_x.

The data in Table 1 when processed by the method previously described and reinserted into the simulation model permits the construction of a prismatic bounding surface. The volume defined by the bounding surface renders the dimensional variation behaviour of the point P readily discernable within the context of the simulation model (Figure 4). The probability that the point P lies outside the volume contained by the bounding surface is, in this instance, 10^{-3}.
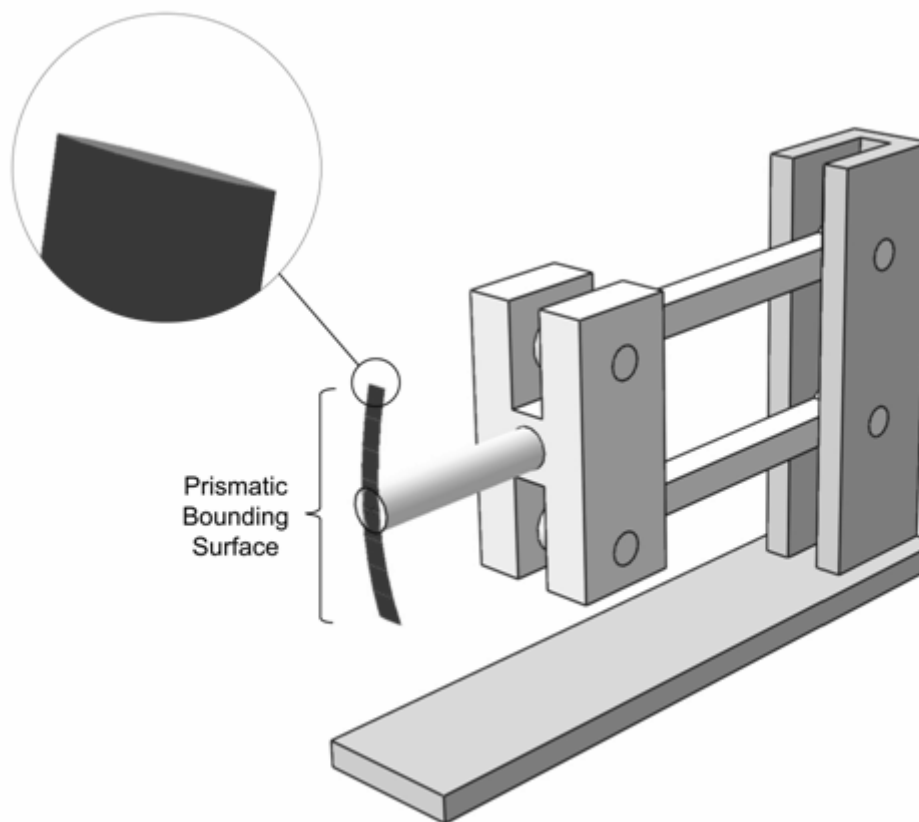
Figure 4 Dimensional variation behaviour of the point P at a probability of 1:1000

Visualisation of the effect of variation on an angle

It is a necessary condition of this method that the user be able to define, in the DVA software, the direction from which the angle to be measured is viewed (Figure 5).
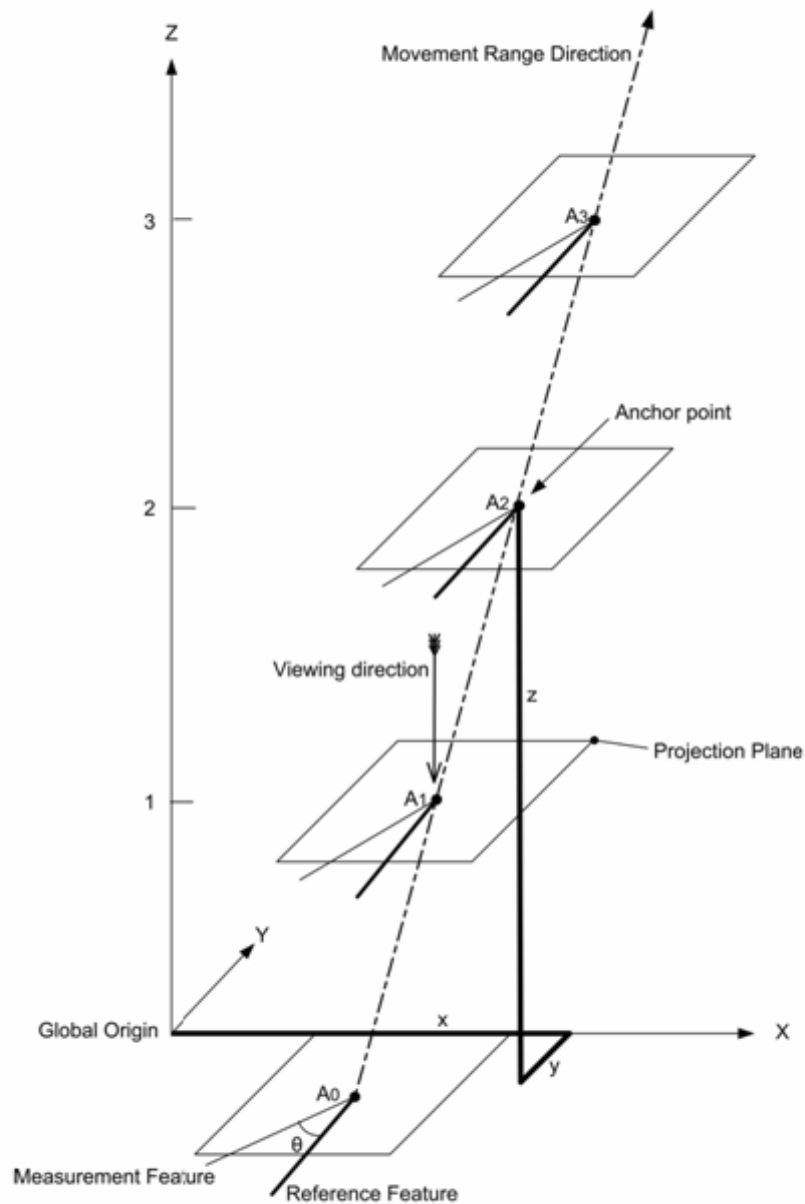
Figure 5 The relationship between the global co-ordinates, anchor point, reference and measurement features and projection plane

The visualisation of angular data is a two-stage process. The first stage is to choose a point in the simulation model to which the visualisation will be anchored. The chosen anchor point is the one most appropriate for, and that facilitates

interpretation of, the information being visualised. This is a largely subjective decision. Once chosen the anchor point is located in 3D space relative to the global co-ordinate system by exactly the same method used to locate the point P described above. The second stage of the process is to generate a reference frame. This describes how the chosen angle is to be measured and locates the visualisation in 3D space relative to the anchor point

Angular visualisation reference frame

The anchor point occupies the same relative position in each configuration of the simulation model and, although its absolute position may vary in each configuration of the simulation model, it acts as the origin for the orthogonal reference frame used to visualise angular data. In a 3D scenario, the angle of a feature (the measurement feature) is measured relative to a reference feature. If the two features are not co-planar, it is necessary to specify a viewing direction to complete the angle definition. The effect of defining a viewing angle is to project the reference and measurement features onto a plane normal to the viewing direction that passes through the anchor point. To reduce possible ambiguities when projecting planar features subject to compound angles, linear measurement and reference features are preferred. Where a suitable linear measurement feature does not exist, it will be necessary to add one to the simulation model. One axis of the reference frame is defined by the viewing direction while the projection of the reference feature onto the projection plane defines a second (Figure 5).

The third axis of the orthogonal reference frame will depend on whether a left or right handed axis system is used. One point that must be born in mind when defining the reference frame is that, as the angle between the movement range direction and the projection plane (Figure 5) decreases, then the quality of the visualisation will also decrease. When the movement range direction lies in the projection plane the visualisation will be reduced to a two dimensional entity and thereby rendered ineffective.

Generation of visualisation data

The output from a dimensional variation analysis (DVA) of an assembly system is usually in the form of a mean value and standard deviation for each system attribute analysed. The basic principle used to generate the visualisation data is the same as that for a point.

The mean value and standard deviation values produced by the dimensional variation analyses are manipulated mathematically to generate two points (B & C Figure 6) that bound the angular variation in the chosen system attribute for a given probability or number of standard deviations. The use of the reference frame reduces the manipulation of the DVA output to a 2D exercise in trigonometry. If the line 0A (Figure 6) which represents the mean value of the angle θ is of unit length then:

$$V\text{-} = \cos(\theta\text{-}\delta\theta) \qquad\qquad U\text{-} = \sin(\theta\text{-}\delta\theta)$$
$$V = \cos(\theta) \qquad\qquad\qquad U = \sin(\theta)$$
$$V\text{+} = \cos(\theta\text{+}\delta\theta) \qquad\qquad U\text{+} = \sin(\theta\text{+}\delta\theta)$$
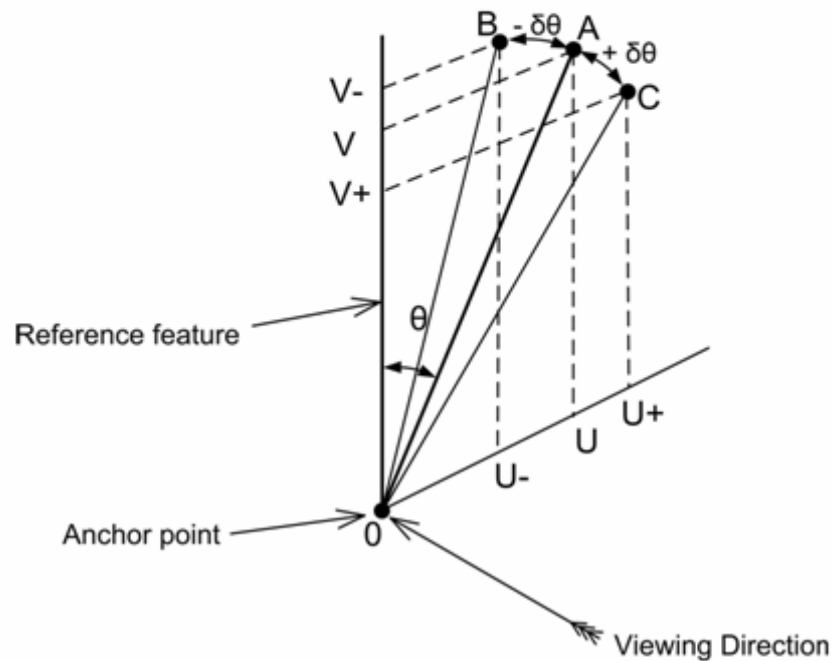
Figure 6 Arrangement for generating the limiting values for angular variation

If the reference frame is aligned to the global co-ordinate system it is only necessary to add the co-ordinates of the points A, B and C to those of the anchor point to arrive at the co-ordinate for the points A, B and C relative to the global co-ordinate system. If the viewing direction is not aligned to the global co-ordinate system, a further trigonometric manipulation will be required to convert the reference frame co-ordinates to those of the global co-ordinate system. The global co-ordinates of the points 0,A, B and C are then collated for all the configurations that comprise the movement range and imported into the simulation model where they are used to create three surface features (Figure 7). The surface comprised of points 0 and B (Figure 6) bounds the lower value of the angle θ while the surface created from the points 0 and C bounds the upper value of the angle. A third surface created from the points 0 and A represents the mean value of the angle. Unlike the closed prismatic surfaces formed when visualising a point, the surfaces of the angular visualisation remain open. The angular visualisation of variation has one advantage over the visualisation of point variation in that the bounding surfaces can be extended to any desired extent in the direction of the measurement reference feature, in this instance the centreline of the axle.
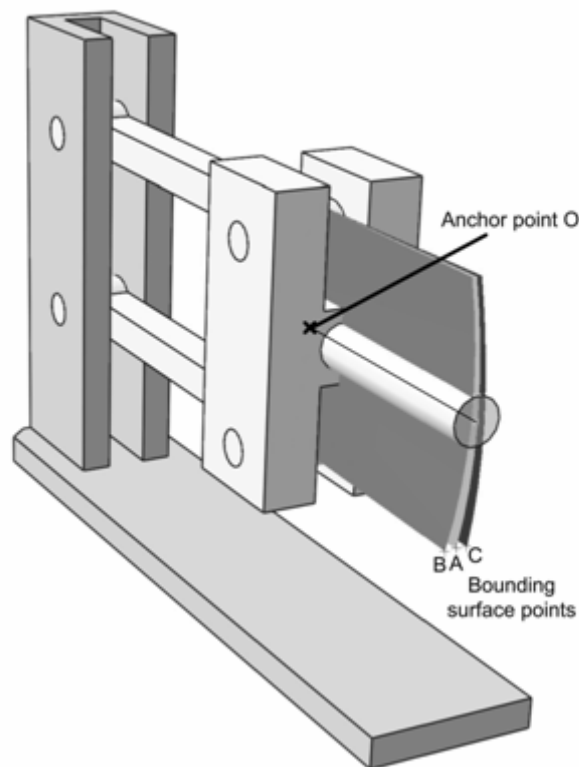
Figure 7 Visualisation of angular variation at a probability of 1:1000

The bounding planes shown in Figure 7 are extended from their nominal unit length to the same length as the axle in the simulation model. In this condition the bounding surfaces not only show the amount of angular variation in the axle centreline but also give an indication of the lateral movement of the outboard end of the axle due to angular variation.

Application

It is envisaged that the main application of the methods described above will be the presentation of complex DVA data to multi disciplinary audiences such as those found in product development or design review in a concurrent or simultaneous engineering environment. The ability to present complex analysis data in a simple manner will enable increased knowledge transfer and thus strengthen the concurrent or simultaneous engineering approach. To assist in the preparation of the visualisations, Two Excel based spreadsheets have been developed to automatically generate the data points used to construct the bounding surfaces. The spreadsheets are known to be compatible with Catia V5R16, Catia V5R19 and ProEngineer Wildfire 3 CAD platforms. A secondary application of the visualisation method is the detection of unexpected dimensional variation behaviour. Consider Figure 7, a visual

examination of the bounding surfaces shows that when the axle is in its lowest position the angular variation is at its maximum but decreases as the axle moves upwards. The same information is present in the tabulated data from which Figure 7 was derived, but the dimensional variation behaviour is not nearly so apparent and thus might not be recognised as such. While the cause of the unexpected behaviour is unknown, once detected it can be investigated further if necessary

The bounding surfaces are created within the simulation model and are thus CAD entities in their own right. It may therefore be possible, depending on the CAD platform used, to employ some of the native CAD interrogative functions, such as clash detection, to perform additional analyses on the bounding surfaces and their relationship with adjoining component parts of the assembly. Such functionality might be particularly useful where more than one set of bounding surfaces are present in the simulation model.

Conclusions

The two method described in this paper provide a means of presenting complex dimensional variation behaviour of kinematic assembly systems in a simple and easily comprehensible manner to audiences with a wide range of technical expertise. This ability enhances the concurrent or simultaneous engineering process by aiding knowledge transfer between the participants.

Further work

At present, the output from the Excel spreadsheet that generates the data points used to construct the bounding surfaces is converted manually from Excel format into a format readable by the CAD platform. Automation of this process would significantly enhance the functionality of the visualisation methods. A second area requiring further development is the means by which the generated data is imported into the CAD platform. At present the importation technique is very software specific. The development of a more generic importation technique applicable to a wider range of CAD platforms would be of considerable benefit. Additional work is required to develop a method capable of visualising the dimensional variation behaviour of linear features in addition to the method for point features described in the main body of this paper. This will require the consideration of the position of the two end points of the linear feature as well as the length of the feature. These parameters are not necessarily independent of each other and will thus complicate any visualisation technique.

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Appendix D   **The use of a two stage dimensional variation analysis model to simulate the action of a hydraulic tappet adjustor in a car engine valve train system**

The use of a two stage dimensional variation analysis model to simulate the action of a hydraulic tappet adjustor in a car engine valve train system.

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Abstract

Dimensional variation analysis (DVA) models have been used in the manufacturing industry for over 20 years to predict how minor variations in the size, shape and location of the components parts is likely to propagate throughout and affect the overall dimensions, operation and performance of a complete mechanical system. This paper is one of in series of four papers that describe how different techniques can be utilised to aid the creation and application of DVA models. This paper explains the development and use of a two stage DVA model to simulate the action of a hydraulic tappet adjuster and dimensional interdependence that exists between the adjustment of a hydraulic tappet and the actuation (opening & closing) of the cylinder valve. The three other papers cover the use of kinematic constraint maps to prepare the structure of a DVA model; the use of virtual fixtures, jigs and gauges to achieve the necessary component location and the required variation measurements, and the use of 3D plots to display large numbers of DVA results as a single 3D shape.

A hydraulic tappet adjustor performs two functions; it is part of the valve train system that actuates (opens & closes) the cylinder valve and it also self adjusts to take up any free play in the valve train system. These two functions, tappet adjustment and valve actuation, are separate operations that occur at different times during the valve train operating cycle and so need to be modelled as different configurations in a DVA model. In a conventional multiple configuration DVA model, each configuration has to be fully constrained and mathematically closed independently of any other model configuration. This requirement makes it difficult to include the interdependence between tappet adjustment and valve actuation. The two stage approach overcomes this limitation by allowing the output variation from the tappet setting configuration to be carried over into the valve actuation configuration and can thereby fully account for the interdependence between the two operations.

Keywords; *Dimensional variation analysis, Engine valve train, Dimensional variation behaviour.*

1. Introduction

DVA (dimensional variation analysis) models have been widely used by automotive companies [1] [2] [3] [4] and to a lesser extent by aerospace companies [5], [6] [7] [8] and other manufacturing companies [9] for over 20 years. A DVA model can simulate how minor variations in component size, shape and location are likely to propagate in all six degrees of freedom throughout the assembly and operation of a mechanical system. DVA models have proved very successful in predicting whether or not these minor component variations, when taken collectively, are likely compromise the overall operation, performance or quality of the complete system. The use of a DVA model provides the engineering team with the means to identify potential dimensional variation problems in advance, during the design phase, while there is still time to 'design out' the variation or to devise effective measures to control the variation once in production. As the software used to build DVA models has advanced over the years, in parallel, the DVA users have developed numerous management procedures, application techniques and 'tricks of the trade' to model specific situations [10] [11] [12] [13] [14] [15] [16] [17]. The advances in software combined with the development of new procedures and techniques have substantially increased the capability of the DVA model and the complexity of the systems that can be modelled.

Hydraulic tappet adjusters are part of the valve train system that actuates (opens & closes) the cylinder valve. Hydraulic tappet adjustors are fitted to automatically take up any free play in the valve train systems. The adjustment of the tappet length and the actuation of the valve occur at different times during the valve train operating cycle. The tappet length is adjusted on the cam heel when the valve is fixed in the closed position, whereas the valve opens and closes on the cam lobe when the tappet length is fixed. There are sufficient differences in terms of the timing and component locations to regard

the tappet adjustment and the valve actuation to be separate operations and this requires each operation to be modelled as a separate configuration in a DVA model. However, dimensional variation in the tappet adjustment could influence the opening and closing of the valve making the operation of the valve dependant on the tappet adjustment and creating an interdependence between the two operations. Variation in the valve train components can affect the operation of the valve train by causing variation in the valve timing and maximum lift. The inclusion of a hydraulic tappet adjustor takes up the clearance in the valve train and can compensate for some, but not necessarily all of the component variation. Depending on the exact system configuration, a hydraulic tappet adjustor can compensate for variation in the length of the valve stem, but not for variation in the length of the rocker arm or the height of the cam lobe.

In a conventional multiple configuration DVA model, each configuration has to be fully constrained and mathematically closed independently of any model other configuration. This requirement makes it difficult to include the interdependence between tappet adjustment and valve actuation. To deliver reliable results a DVA model should account for as many known sources of dimensional variation as possible. To not include a known variation source (the influence of tappet adjustment on valve actuation) in a DVA model of the valve system limits the capability of the DVA model and challenges the integrity if the DVA results. The aim was to overcome this limitation by developing a two stage approach that would allow the output variation from the tappet setting configuration to be carried over into the valve actuation configuration. The carry over of variation creates the required interdependence between the two configurations that simulate the tappet adjustment and valve actuation operations in the DVA model.

1.1. The advantages of hydraulic tappet adjustors over conventional screw type adjustors

Several large manufacturers such as Land Rover, Ford, Honda and Mitsubishi use hydraulic tappet adjusters in preference to the older screw type adjuster The hydraulic tappet adjuster has several advantages; the assembly of the valve train is simplified as no manual adjustment is required. Transient thermal effects during the engine warm up period can be accommodated thereby improving engine efficiency. The automatic adjustment of the hydraulic tappet adjusters compensates for certain types of long-term wear in the valve train, maintaining the engine in optimal condition for longer.

Considerable research has been undertaken on the effect of variations in the valve timing of engines [18] [19]. This work has shown that variations in the valve timing can have a significant effect on both engine performance and emissions. Indeed several variable valve timing systems [20] [21] [22] have been developed that exploit the fact. In such systems, the valve timing and lift are varied deliberately to enhance the engine performance. However, the valve timing and lift may also be affected by the inherent variation in the processes used to manufacture and assemble the components. This variation, unless

properly controlled, may accumulate to the point where it becomes detrimental to product performance. DVA is often used to resolve such issues when they arise [3] To determine the effects of such variation on the valve train requires the ability to simulate the dimensional variation behaviour of the entire valve train, including the hydraulic tappet adjusters, throughout the operational cycle.

In valve trains employing screw type tappet adjusters the size shape and location of the component parts are fixed and fully defined. Thus simulating the dimensional variation behaviour of the valve train is comparatively straightforward. In valve trains, containing hydraulic tappet adjusters the length of the hydraulic adjuster is variable, it is defined by the position of the adjacent components with which it is in contact. These adjacent components are subject to the effects of variation and they may also move in space as the valve train progresses through its operational cycle making it difficult to define the axial length of the hydraulic tappet adjuster The position of the hydraulic tappet adjusters in the valve train will depend on the specific engine design, but hydraulic tappet adjusters are commonly found in one of three positions, at the rocker arm pivot, between the cam and the rocker arm or between the rocker arm and the valve stem (Figure 1).
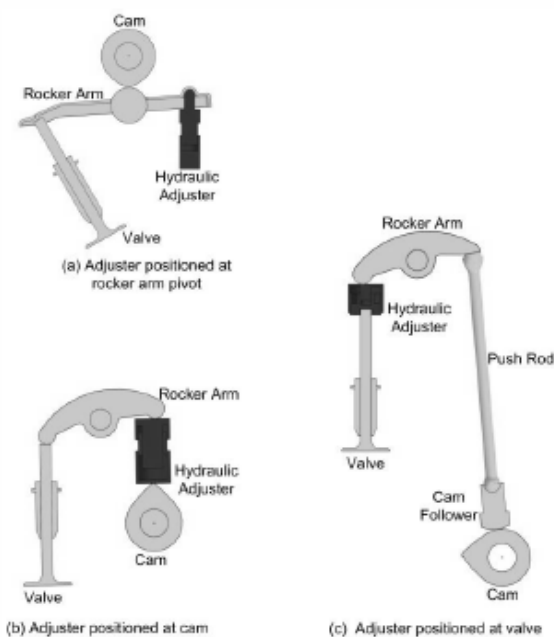
Fig 1 Common hydraulic tappet adjuster locations

This paper proposes a method for simulating the dimensional variation behaviour of valve train systems containing hydraulic tappet adjusters. The method is applicable to any of the three common locations for hydraulic tappet adjusters shown above and is capable of defining the axial length of the hydraulic tappet adjuster. The method has been developed for use with vector loop based DVA software and could be used in other DVA software.

2. Operation of hydraulic tappet adjusters.

In order to appreciate the significance of the assumptions made to simulate the behaviour of a hydraulic tappet adjuster it is necessary to consider the general principles behind how the adjuster operates. In its simplest form, the hydraulic tappet adjuster consists of a cylindrical barrel closed at one end with a spring situated between the closed end of the barrel and a very close fitting hollow plunger that slides in the barrel to form a telescopic strut (Figure 2). A series of oil galleries in the barrel and plunger allow pressurised oil, from the engine lubrication system, to enter the centre cavity of the hollow plunger and, by means of a spring loaded non-return valve in the plunger, into a compression chamber formed between the closed end of the barrel and the plunger (Figure 2). The cycle of operation for the hydraulic tappet adjuster commences when the engine valve closes and the load applied to the valve train by the compressed valve spring is removed. This allows the spring-loaded plunger of the adjuster to extend and take up any clearance in the valve train. As the plunger extends the volume of the compression chamber increases, reducing the pressure of the oil trapped within. This in turn opens the non-return valve of the plunger allowing oil to flow into the compression chamber until the pressures equalises at which point the non-return valve closes. When the cam follower makes contact with the cam flank, a load is applied to the hydraulic tappet adjuster by the valve train. This load compresses the oil trapped in the compression chamber and prevents the non-return valve from opening. The oil in the compression chamber is a high bulk modulus fluid that acts as if it were a rigid strut maintaining the relative positions of the hydraulic adjuster barrel and plunger and thus, the overall length of the hydraulic adjuster. The hydraulic adjuster remains in this state until the engine valve closes and the operational cycle begins anew.
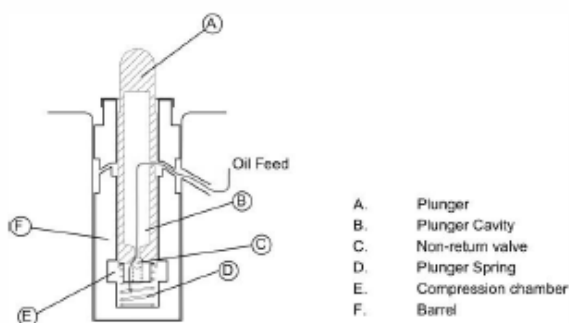

A. Plunger
B. Plunger Cavity
C. Non-return valve
D. Plunger Spring
E. Compression chamber
F. Barrel

Fig 2 Hydraulic tappet adjuster

The above can be considered the idealised operational cycle of the hydraulic tappet adjuster. In the real world, hydraulic tappet adjusters are manufactured in such a way that they exhibit tappet leak down. This is the controlled escape of oil from the compression chamber between the plunger and the adjuster barrel. This attribute is to ensure that the engine valve always returns fully to its seat.

3. Simulation assumptions

To simulate the dimensional variation behaviour of a hydraulic tappet adjuster certain assumptions are made concerning the operation of the hydraulic tappet adjuster these are

- When the cam follower makes contact with the cam flank the hydraulic adjuster becomes rigid, instantaneously.
- When in the rigid condition the length of the adjuster remains fixed

The first assumption is considered reasonable as the non-return valve built in to the plunger only remains open while there is a sufficient pressure differential across the valve to overcome the valve closing spring. This pressure differential will decline once the adjuster plunger reaches its maximum extension and the non-return valve may well close before the cam follower makes contact with the cam flank. The second assumption is necessary to determine the length of the hydraulic tappet adjuster in the simulation model. Once created the simulation model can then be modified to take into account tappet leak down. However, the inclusion of the tappet leak down is beyond the scope of the present paper. For the purposes of describing the method the system characteristics of interest are the valve lift for a given rotation of the camshaft and the cam angle when the valve first opens and first closes.

4. Modelling method

The basic method for analysing dimensional variation of an assembly to determine the dimensional variation behaviour is to construct a DVA model, which is then analysed. The DVA model consists of the CAD geometry that defines the nominal size, shape and location of the component parts. The CAD geometry is then overlaid with assembly features that define the extent of the component variation in the assembly. The degrees of freedom between mating assembly features on adjacent component parts are appropriately constrained to form the connections that join the component parts together in the desired arrangement or configuration.

A two part DVA model is created to simulate the behaviour of a system containing hydraulic tappet adjusters. This change in method is necessary as the axial length of the hydraulic tappet adjuster is defined by the relative location of the adjacent component parts to account for the interdependence between setting the adjuster length and the operation of the valve

4.1. DVA model part 1, setting the adjuster length

The first part of the DVA model has three objectives;
- To set the overall axial length of the hydraulic tappet adjuster
- To identify those variation source elements that contribute to the variation distribution of the hydraulic tappet adjuster overall length

- To identify those variation source elements that contribute to the hydraulic tappet adjuster variation distribution but also have secondary effects that do not affect the hydraulic tappet adjuster itself but do affect a key characteristic of the system and to quantify that effect

4.1.1. Defining the length of the hydraulic tappet adjuster

The method of defining the hydraulic tappet adjuster axial length is demonstrated using the component configuration shown in Figure 3; this simulates the behaviour of system in which the hydraulic tappet adjuster is located between the cam and the rocker arm. This system was chosen to illustrate the method, as it is the only arrangement in which it is necessary to both, determine the length of the hydraulic tappet adjuster and accommodate gross motion of the hydraulic tappet adjuster. In the other two arrangements (Figure 1), the hydraulic tappet adjuster either, acts as a pivot for the
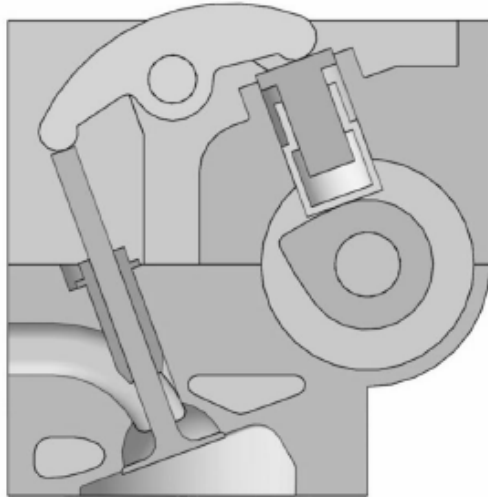

Fig 3 Component configuration used to define the length of the hydraulic tappet adjuster.

rocker arm and is thus static or the hydraulic tappet adjuster is in direct contact with the valve. In the latter case, a simpler solution is available. From the description of the operational cycle of the hydraulic tappet adjuster, when the valve is on its seat and the cam follower (hydraulic adjuster) is in contact with the base circle of the cam, the hydraulic adjuster will extend to fill the gap between the cam and rocker arm (Figure 3). The distance between the cam base circle and the contact point on the rocker arm is thus the length of the hydraulic adjuster. When the configuration changes and the adjuster is no longer in contact with the cam base circle it has been assumed that the hydraulic adjuster becomes rigid and of fixed length instantaneously. Thus, the length of the hydraulic adjuster defined in the arrangement shown in Figure 3 is applicable across the whole operational cycle of the valve train. The length of the hydraulic tappet adjuster is determined by the simple expedient of using an assembly level measurement to find the distance between the two ends of the adjuster when measured along the centre line of the adjuster. It should be noted at this point

that the two halves of the hydraulic tappet adjuster are modelled as individual component parts and not as a sub assembly. This is essential to ensure that the two halves of the hydraulic tappet adjuster are capable of independent movement in the simulation model. If they were added as a sub assembly the positions of the two halves relative to each other would be fixed. The complete sub assembly would be capable of independent movement but not the component parts.

The first part of the DVA model should be viewed as a virtual jig [23] used to align the two halves of the hydraulic tappet adjuster. The output from this part is the measured length and variation distribution of an aligned, hydraulic tappet adjuster. While the output defines the length of the hydraulic tappet adjuster it is not in a form that can be directly imported into the second part of the DVA model. The variation distribution represents the net effect of all the variation sources that influence the length of the hydraulic tappet adjuster. For example if variation increases the length of the valve then the length of the hydraulic tappet adjuster will reduce so that it still exactly fills the gap between the rocker arm and the cam. The variation distribution of the hydraulic tappet adjuster relies on a long valve being matched with a short adjuster and vice versa. When the data is exported, this link is broken and the possibility exists of a long valve being matched with a long adjuster in the measurement process. This would add an extra variation source to the DVA model rendering it inaccurate.

Previously it has been assumed that when the hydraulic tappet adjuster breaks contact with the cam base circle, it instantaneously becomes rigid and of fixed length, thus the standard deviation of the hydraulic tappet adjuster variation distribution is equal to zero. The mean length of the hydraulic tappet adjuster without its variation distribution can be imported into the second part of the DVA model. If the variation distribution of the hydraulic tappet adjuster is to be set to zero then the variation source elements that contribute to the variation distribution of the hydraulic tappet adjuster must be identified and also set to zero to avoid double counting any of the variation source elements.

4.1.2. Identifying which variation sources affect the hydraulic tappet adjuster axial length

The complete valve train contains numerous sources of variation. Some of these will affect the length of the hydraulic tappet adjuster others will not. Those variation sources or at least the elements of those variation sources that affect the length of the hydraulic tappet adjuster are negated by the action of the adjuster as it takes up any clearance in the system. By including a contributor analysis, a function common to most DVA software, in the analysis of the hydraulic tappet adjuster axial length those variation source elements that affect the length of the hydraulic tappet adjuster can be identified. This identification is aided by the introduction into the CAD model of a local co ordinate system. The local co ordinate system is aligned such that one axis is coaxial with the longitudinal axis of the hydraulic tappet adjuster (Figure 4)
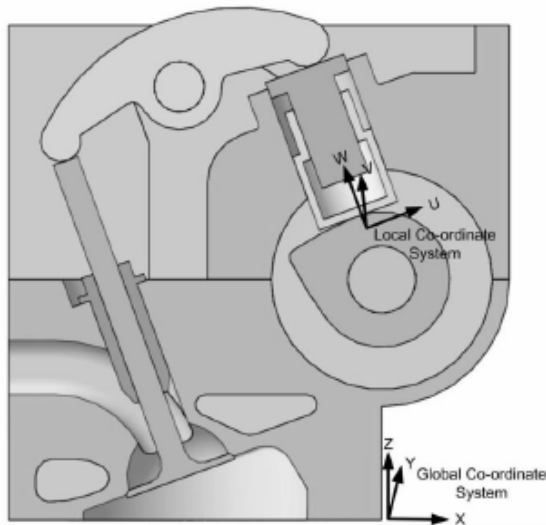
Fig 4 Local and global co ordinate systems

The assembly joints in the first part of the DVA model are aligned, where appropriate, to the local co ordinate system rather than the global co ordinate system. When using only the global co ordinate system any variation source that causes translation along either the X or Z axes (Tx, Tz) may affect the length of the hydraulic tappet adjuster to some extent. Variation in Tx and Tz may also influence parameters that, for example affect the valve lift or valve timing. However, using the local co ordinate system only those variation sources that have a Tw element are likely to affect the hydraulic tappet adjuster length thus reducing the number of potential contributors.

4.1.3. Identifying and quantifying variation source secondary effects

Some variation sources may produce both an obvious primary effect and a more subtle secondary effect on the system. Consider the length of the valve stem when setting the length of the hydraulic tappet adjuster (Figure 3), as the length of the valve stem increases or decreases it will cause the rocker arm to rotate slightly. This will in turn increase or decrease the distance between the other end of the rocker arm and the base circle of the cam effectively changing the length of the hydraulic tappet adjuster as it negates the variation. However, as the rocker arm rotates it will cause the contact point between the rocker arm and the valve stem to move across the face of the valve stem. This will change the effective length of the rocker lever arm (Figure 5). The lever arm is defined, in this instance, as the distance between the rotation axis of the rocker arm and the contact point between the rocker arm and the valve stem measured in the U direction (Figure 4). A similar affect will occur at the other end of the rocker arm at the contact point between the rocker arm and the hydraulic tappet adjuster. The extent of the variation can be determined by measuring the distances x and y (Figure 5) in the first part analysis. This will give a variation distribution for each measurement as well as a mean value. The two affects may well be dissimilar in extent. Thus, a unit movement of the hydraulic tappet adjuster may move the valve by a distance x/y or u/v depending on whether the valve stem is above or below

mean length. The question thus arises as to how the effects of variation are to be simulated especially so when a single variation source element gives rise to two different effects one of which influences the length of the hydraulic tappet adjuster and one which does not, but does influence a key characteristic of the system.
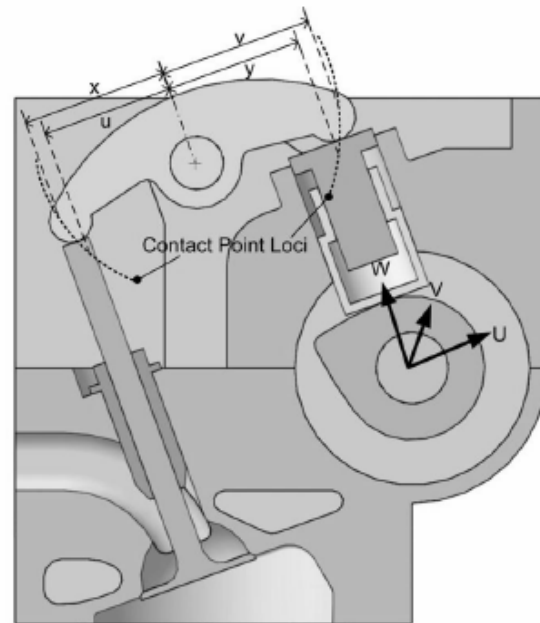

Fig 5 Effect of valve length on the rocker lever arm ratio

4.2. DVA model part 2, simulating the system behaviour

The first part of the DVA model identified and quantified the axial length of the hydraulic tappet adjuster, the variation sources that affect the axial length of hydraulic tappet adjuster and any secondary effects they may have on the other key characteristics of the system. The second part of the DVA model is dependent on this information to achieve a different set of objectives these are;

- To simulate the manner in which the hydraulic tappet adjuster negates the effect of certain variation source elements
- To ensure that any secondary effects of variation sources negated by the action of the hydraulic tappet adjuster are retained in the DVA model.
- To simulate and analyse the effects of variation on the valve lift and valve timing.

4.2.1. Simulating the behaviour of the hydraulic tappet adjuster

A significant feature of the second part of DVA model is that the valve is no longer in contact with its seat. In consequence, the length of the hydraulic tappet adjuster must now be defined externally. As noted earlier the variation distribution of the hydraulic tappet adjuster should not be imported into the second part of the simulation models as it creates an additional source of

variation. It should, however, be remembered that the hydraulic tappet adjuster is designed to negate the effects of variation that would otherwise create clearance or slack within the valve train system. The solution is to set the hydraulic tappet adjuster to the mean length as defined by the first part of the DVA model. Those variation source elements identified as contributors to the variation distribution of the hydraulic tappet adjuster are also set to zero. The variation sources contribute no variation to a length that does not vary, thus ensuring consistency between the two. A benefit of using a local co ordinate system and aligning the assembly joints to that system now becomes apparent in that most of the variation source elements that require modification will be Tw elements (Figure 4). The overall effect is consistent with the assumed real world behaviour of the system in that the hydraulic tappet adjuster is of fixed length and any variation sources that might cause clearance are negated by the action of hydraulic tappet adjuster.

4.2.2. Retention of secondary variation source effects

Despite the DVA model having two parts, the behaviour of a single valve train is being simulated. It is therefore important that the effect of each variation source element appears once and once only in the simulation model. Any source of variation that may affect the length of the hydraulic tappet adjuster must appear in the first part of the DVA model used to define the length of the hydraulic tappet adjuster. Equally, there may be sources of variation present that do not influence the length of the hydraulic tappet adjuster but do influence the valve lift or timing. These effects must be included in the second part of the DVA model. For example, it has been shown that a secondary effect of variation in the length of the valve can influence the valve lift (Figure 5). Yet in the second part of the DVA model the valve length of has been set to its mean value. The secondary effect therefore needs to be incorporated into the second part of the DVA model by some other means. To do so requires an assembly level measurement of x and y (see Figure 5) to be included, in part one of the DVA model. On analysis, it will give a variation distribution for x and y. This variation distribution can be incorporated into the simulation models by using a root sum of the squares (RSS) method to add it to the Tu element of the positional tolerance of the curved end of the rocker arm. Thus, the effect of a variable valve length on the valve lift and timing is present in the simulation model even when the valve length is fixed. This is possible because the centre of curvature of the end of the rocker arm drives the location of the contact point between the valve and rocker arm. The contact point is defined as the point of intersection between the curved end of the rocker arm and a line parallel to the valve axis that passes through the centre of curvature of the end of the rocker arm.

It has been stated that the effect of each variation source element must appear once and once only in the two parts of the DVA model. However, the sources of variation in the assembly may not be independent of each other. Consider the profile of the cam (Figure 3). In this particular instance, the cam profile consists of four facets, the base circle, the cam toe and the leading and trailing flanks. The four facets of the cam profile may be ground in a single operation to give a smooth profile with no

discontinuities. Thus if the base circle of the cam varies in size the adjacent cam flank must also vary to the same extent if profile discontinuities are to be avoided. If discontinuities do occur then the realism of the simulation model is called into question.

Part one of the DVA model must include any variation in the cam base circle as this directly affects the axial length of the hydraulic adjuster. Equally, any variation in the cam flanks or cam toe must be included in part two, as these will directly affect the valve lift. Variation of the cam flanks is influenced by variation of the cam base circle. Thus, any variation in the cam base circle must be included in the second part of the DVA model as it indirectly affects the valve lift. As a result, variation of the cam base circle is present in both parts of the DVA model. However, in part one, variation of the cam base circle directly affects the length of the hydraulic adjuster, which negates any effect on the valve lift. In the second part of the DVA model, variation of the cam base circle indirectly affects the valve lift through the cam flank. However, as the cam base circle does not make contact with any of the other component parts of the assembly once the valve is open only the cam flank is affected by variation in the cam base circle. Thus while the cam base circle acts as a source of variation in both parts of the DVA model the two different effects of this variation source appear once and once only.

4.2.3. Simulating the effects of variation on the valve lift and timing

To simulate the effects of variation on the valve lift and valve timing in the second part of the DVA model, two groups of simulation model configurations are required. The first group simulates the effects of variation on the valve lift, by fixing the angular position of the cam with, in this instance, a 5° increment in the cam angle between each configuration. The total rotational range covered being top dead centre (TDC) to 110° after TDC. The extent of the valve lift is then determined by an assembly level measurement incorporated in each model configuration. The second group, which has two configurations, simulates the effect of variation in the cam angle just as the valve is opening and closing. The position of the valve is fixed, in this instance, to 0.01mm off the valve seat with the hydraulic tappet adjuster in contact with the leading and trailing flanks of the cam respectively. This particular valve position was chosen as it describes two and only two positions in the operational cycle of the valve train whereas the valve is in contact with the valve seat for a significant portion of the operational cycle. The cam angle is then measured.

5. Analysis results

The modelling method described above has been applied to valve train systems containing hydraulic tappet adjusters in the three common locations and was found to produce a viable simulation model. Figure 6 shows the results obtained when simulating the behaviour of system in which the hydraulic tappet adjuster is located between the cam and the rocker arm (Figure 3). The analysis results for the valve lift and valve timing of the hydraulic tappet adjuster are compared against those from an

identical system but fitted with a screw type adjuster (Figure 6). The solid curves in Figure 6 represent the mean valve lift while the error bars represent the limit of variation in the valve lift at three standard deviations. Similarly, the columns represent the variation in the cam angle at the point where the valve is just opening or closing. The solid line represents the mean value while the shaded areas represent the variation at three standard deviations.
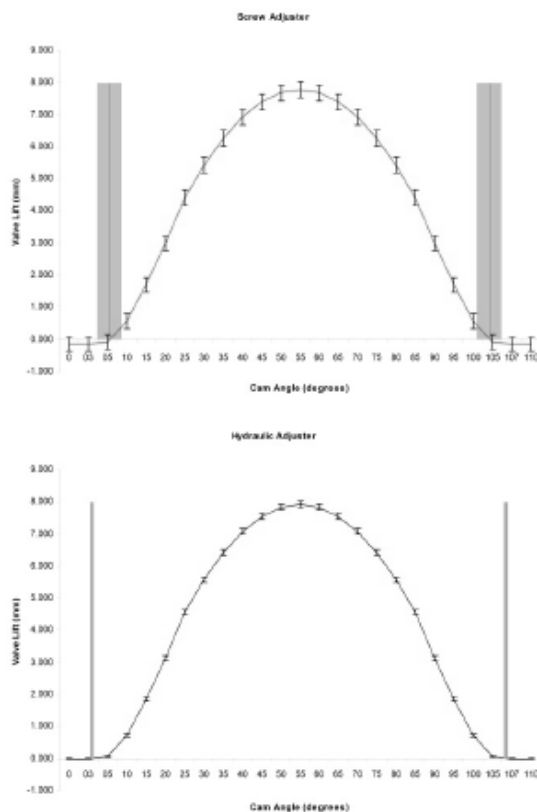

Fig 6 Valve train analysis output

The system fitted with the hydraulic tappet adjusters shows significantly less variation than that fitted with the screw adjusters. This is to be expected as the hydraulic tappet adjusters are specifically designed to negate some of the variation. A second feature to note is that the system fitted with the screw adjuster shows a slightly reduced valve lift compared to the hydraulic tappet adjuster system. This is a reflection of real world practice where, even when hot, a slight clearance gap is usually left in valve trains fitted with screw adjusters to ensure that the valve always closes on the compression stroke. This same clearance gap is responsible for the apparently negative valve lift as the valve closes in a system equipped with a screw adjuster. The analysis is performed using vector loop based software where vector loop and simulation model closure are a necessary condition for analysis. While it is possible to simulate a variable gap, it is difficult to model. In this instance it was considered simpler to maintain model closure by allowing all the component parts to remain in contact and to interpret the negative valve lift as the opening of a clearance gap.

A comparison of the contributors to variation in the valve lift shows some significant differences and similarities between the two systems (Figure 7). The three major contributors in the screw adjuster system are absent from the hydraulic system contributors as they are negated by the action of the hydraulic tappet adjuster. Perhaps more significantly the remaining contributors appear in both systems in the same sequence. This suggests that both systems behave in a similar manner and only the action of the hydraulic tappet adjuster in negating certain variation sources distinguishes the two systems. The overall analysis shows that the simulation of the valve train system containing hydraulic tappet adjusters is consistent with real world expectations.
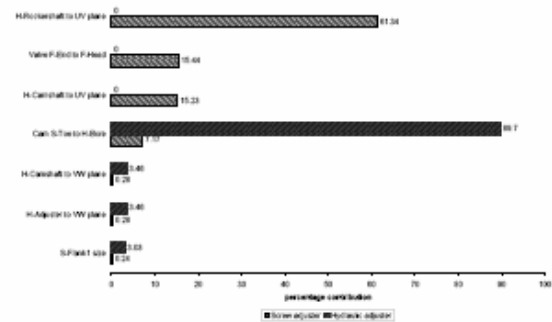

Fig 7 Comparison of contributors to valve lift variation

6. Conclusions

The method described in this paper provides the capability to model and simulate the dimensional variation behaviour of three common valve train configurations containing hydraulic tappet adjusters. This in turn enables the effects of variation on performance related characteristics such as valve lift and valve timing to be analysed. The described method while requiring a more complex two part simulation model can include and account for the all important interdependence between the adjuster setting and valve operation configurations and allows the analysis of assembly systems where one or more significant parameters are not defined from the outset.

Although the effect on the valve actuation from variation due to the tappet adjustment was small, never the less, this example of a hydraulic tappet adjuster still clearly shows that a two stage DVA model can be used to transfer the resultant variation from one model configuration into another model configuration. This increases the capability of the DVA model by allowing interdependence between one model configuration and another to be included and accounted for. The two stage approach relies upon:

- There being a suitable intermediate variable that can be used to carry over the variation. In this example of a hydraulic tappet adjustor, the 'carry over' variable was the effective length of the rocker arm. The output from the tappet adjustment stage was to calculate the resultant variation in the effective length of the rocker arm. This output

variation from the tappet adjustment stage was then carried over as an input to the valve actuation stage to create the required interdependence.

- The careful segregation of the component variations between each of the two stages to avoid any of the component variations from being double counted.

7. Further work

The method described in this paper has been applied to valve trains systems where the nominal valve lift and timing are fixed. However, the use of variable valve event (VVE) valve train systems is becoming more widespread. It will therefore be necessary to develop methods of simulating the dimensional variation behaviour of VVE valve trains and the hydraulic tappet adjusters they contain.

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