行政院國家科學委員會專題研究計畫 期末報告

機電系統可靠度設計分析與預兆式健康管理作業

計 畫 類 別 ：個別型
計 畫 編 號 ：NSC 100-2221-E-237-001-
執 行 期 間 ：100 年 08 月 01 日至 101 年 08 月 31 日
執 行 單 位 ：德霖技術學院機械工程系(科)

計畫主持人：蔡有藤
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報 告 附 件 ：出席國際會議研究心得報告及發表論文

公 開 資 訊 ：本計畫可公開查詢

中 華 民 國 101 年 08 月 17 日

中 文 摘 要 ： 隨著消費型態改變，產品功能品質要求越來越高，可靠度也越來越被重視，為避免設計不足或設計過當現象，考慮可靠度、壽命需求，進行可靠度設計是必要手段，本計畫建立系統可靠度設計發展架構，分析系統功能特性、可靠度需求，對系統進行可靠度最佳化配置，對傳動機構定義設計輸入、輸出變數，分析設計變數不確定性，建構輸入變數分析模型，利用強度應力干涉理論，預測分析設計失效機率。透過機率設計與分析，結合最佳化方法，建構系統在不同負荷條件下的可靠度、失效機率、失效事件。相關技能有可靠度配置、隨機變數設定、機率分析、可靠度計算、設計變數敏感度、機率為基的風險和成本等。

中文關鍵詞： 可靠度設計、最佳化、維護性

英 文 摘 要 ： Following the change of consuming type, product quality is requested higher and higher as well as its reliability is to be emphasize. This project would construct the procedures of reliability optimization design for an automatic production device. The function characteristics and reliability needs of the system will be investigated to allocate reliability to the function modules. The CAD models of the conveyed device of parts would be constructed based on probabilistic theories. The uncertainty of the supply and the requirement as well as material strength and loading stress would be analyzed to develop the models of reliability design optimization. The failure probability under different load conditions, failed sensibility and operated risk, etc., would be also investigated for generating the optimal solutions of design.

英文關鍵詞： reliability design, optimization, maintainability

行政院國家科學委員會補助專題研究計畫成果報告

機電系統可靠度設計分析與預兆式健康管理作業

計畫類別：■個別型計畫　　□整合型計畫

計畫編號：NSC 100-2221-E-237 -001

執行期間：**100** 年 **8** 月 **1** 日 至 **101** 年 **8** 月 **31** 日

計畫主持人：蔡有藤

共同主持人：王國雄、諶其騮

計畫參與人員：陳紋鳫、何宗翰、陳嘉偉、孫愉翔

執行單位：德霖技術學院 機械工程系

中 華 民 國 **101** 年 **7** 月 **30** 日

行政院國家科學委員會專題研究計畫成果報告

機電系統可靠度設計分析與預兆式健康管理作業

A study of reliability design & analysis and prognostic health management for a mechatronic system

計畫編號：NSC 100-2221-E-237 -001

執行期限：100 年 8 月 1 日 至 101 年 8 月 31 日

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執行機構：德霖技術學院 機械工程系

一、中文摘要

隨著消費型態改變，產品功能品質要求越來越高，可靠度也越來越被重視，為避免設計不足或設計過當現象，考慮可靠度、壽命需求，進行可靠度設計是必要手段，本計畫建立系統可靠度設計發展架構，分析系統功能特性、可靠度需求，對系統進行可靠度最佳化配置，對傳動機構定義設計輸入、輸出變數，分析設計變數不確定性，建構輸入變數分析模型，利用強度應力干涉理論，預測分析設計失效機率。透過機率設計與分析，結合最佳化方法，建構系統在不同負荷條件下的可靠度、失效機率、失效事件。相關技能有可靠度配置、隨機變數設定、機率分析、可靠度計算、設計變數敏感度、機率為基的風險和成本等。

關鍵字：可靠度設計、最佳化、維護性

Abstract

Following the change of consuming type, product quality is requested higher and higher as well as its reliability is to be emphasize. This project would construct the procedures of reliability optimization design for an automatic production device. The function characteristics and reliability needs of the system will be investigated to allocate reliability to the function modules. The CAD models of the conveyed device of parts would be constructed based on probabilistic theories. The uncertainty of the supply and the requirement as well as material strength and loading stress would be analyzed to develop the models of reliability design optimization. The failure probability under different load conditions, failed sensibility and operated risk, etc., would be also investigated for generating the optimal solutions of design.

Key words: reliability design, optimization, maintainability.

二、計畫緣由與目的

研究顯示，產品之未來成本在設計時已被決定了 80 %， 亦即在設計階段已進行 80 %的產品成本決策，剩餘後續工程的影響僅 20 %，而一個產品設計成本據估計僅佔總成本之 5 %，因此，在設計階段發較多的時間進行設計分析與驗證，是值得研究的主題。透過產品功能分析，設計模擬，工程最佳化、品保方法的應用，可減少設計錯誤發生，降低產品開發成本。可靠度是代表產品功能的重要指標，它是一個系統或產品，在某一操作條件下、使用時間內正常工作的機率[1]，它透過設計手法建立在系統或產品內部，確保系統功能輸出正常。為避免設計浪費同時滿足產品功能需求，在設計階段考慮產品可靠度、壽命需求，結合 CAE 軟體進行模擬分析驗證是必要工作；而考慮後續使用維護作業，進行維護性設計(Design for Maintenance)，可降低產品壽期成本，發揮預期效能[2]。

可靠度設計常用於精密且複雜之工程系統設計中，可靠度設計也可說是工程機率設計，它利用強度-應力干涉理論(Strength-Stress Interference theory， SSI)進行設計，在強度與應力隨機變量為常態分佈且均互相統計獨立的條件下，對承受單負載或複合負載元件進行可靠度設計與分析，可靠度隨需求條件而變，在相同供給（強度）下，若需求（應力）越低可靠度將越高[3]。一個好的設計是要避免設計的可靠度過當或不足，透過 CAE 軟體的分析可避免這問題發生，過去，主持人在可靠度設計領域已做過研究，如考慮可靠度、維護度，發展妥善率為基的設計[4]，利用幾何規劃進行可靠度最佳化設計[5]，利用基因演算法進行系統可靠度最佳化設計[6]等。而欲對產品

進行壽命設計，考慮累積損傷問題如：疲勞、磨耗、腐蝕、老化等是常被研究主題[7]。

隨著裝備的性能不斷提高，構造也漸趨複雜，可靠度預估(reliability prediction)是一種在系統與裝備研製過程中，針對正在研發或已經存在的產品設計，提供設計固有可靠度期望值的指引，定量評估與驗證其可靠度現況是否符合規定需求的技術。預估結果可作為評估及比較各種不同設計方案間的差異、提供管理階層作為各種方案之間擇優決策、以及規劃、分配預算與時程的參考。其目的在確認當代設計與製造技藝水準的適用性與可行性，評估零件選用情況，掌握零件及應力效應，了解設計弱點，確定設計之關鍵零組件，從可靠度觀點提供設計保證作業的改進建議，評估設計品在各階段的可靠度現況，適時反應現階段研製產品的可靠度水準是否滿足可靠度需求，作為規劃執行可靠度成長試驗與可靠度鑑定試驗之指標，提供產品未來維修保養與成本分析之參考。

本計畫對一機電系統之機構進行可靠度設計，分析系統失效問題，透過故障樹分析方法(FTA)及失效模式影響分析(FMEA)方法，找出機械系統失效關鍵點，依據功能需求建構系統模型，將系統的可靠度需求配置到系統以下的各個組合模組可靠度需求，建立可靠度失效模式；利用 CAE 及統計分析軟體，分析系統在隨機負荷下之可靠度及預期壽命，分析機台內組件負荷情況、失效模式，收集關鍵組件失效數據，利用最大相似度(Maximum likelihood)，估算可靠度退化參數[8]，預估系統使用壽命，結合工程最佳化方法發展系統可靠度設計法，建立系統可靠度退化模型，可靠度參數估計法，同時分析系統維護性，平均失效時間、停機維護時間，建立以妥善率為中心設計架構。

三、研究方法

本計畫對一自動化機台進行可靠度設計，該機台由控制、動力、機構、感測等模組組成，依據系統可靠度需求，進行可靠度配置、可靠度設計。本計畫分析設計變數不確定性（負載、材料特性、幾何形狀等），輸入變數數學模型、機率分佈，建構機率設計發展架構（如圖 1所示）。

相關技能有：隨機變數定義、可靠度計算、可靠度配置、最佳化模型等。發展程序歸納如下：(1) 定義製程與收集數據，(2) 建立設計變數模式 X1, X2, etc.，(3) 建立製程數學模式 Y = f (X1,X2,.)，(4) 建立預測數學模式，(5) 整合預測數學模式，(6) 機率分析技術應用、運用結果&開發計劃。

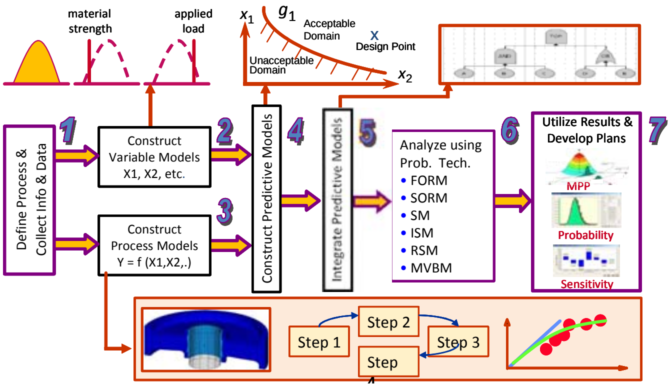


圖 1. 機率設計架構

本計畫對一傳動機構進行設計，分析系統功能特性、設計需求，設計輸入、輸出變數，組件不同負荷大小及分佈情況，考慮機件設計變異，結合機率理論、最佳化方法與田口法進行可靠度最佳化設計，分析預測組件可靠度、失效模式、機率敏感度，估算系統在不同負荷條件下的可靠度、失效機率等。

系統可靠度

決定設計方案後，依據系統可靠度目標，配置規劃各子系統及零組件可靠度，進一步進行可靠度設計，建立可靠設計模式、可靠度設計準則，定性、定量評量標準，有關系統可靠度設計程序如圖 2所示。本計畫訂定各功能模組性能需求，利用強度應力干涉理論，建立各模組可靠度估算模式，依據系統組合關係建立系統可靠度方塊圖，結合可靠度理論與最佳化方法，對系統進行可靠度最佳化設計，進一步規劃使用時維修策略，評估系統妥善率及總成本，以為設計修改及決策依據。

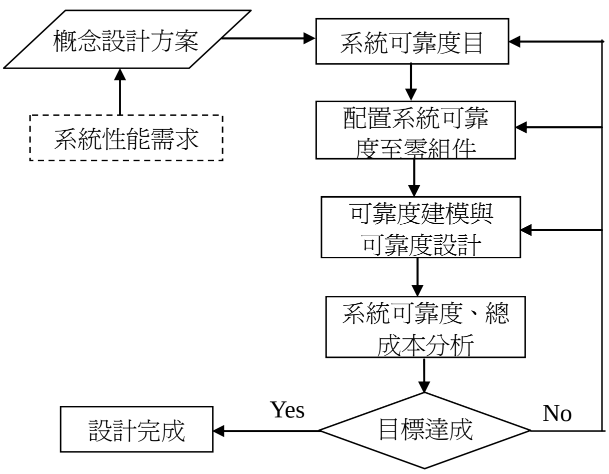


圖 2. 系統可靠度設計程序

可靠度配置

可靠度配置是將系統可靠度目標分配至每一分系統、單機、零組件等層次之可靠度目標值，建立各個組合層次可靠度目標需求，本計畫利用ARINC 配當法對各功能模組配置適當可靠度，以獲得系統最大可靠度，配置時考慮系統及零組件可靠度等因素加以權衡擇適(trade-off)，取得更符合經濟效益的設計產品，配置流程如圖 3。

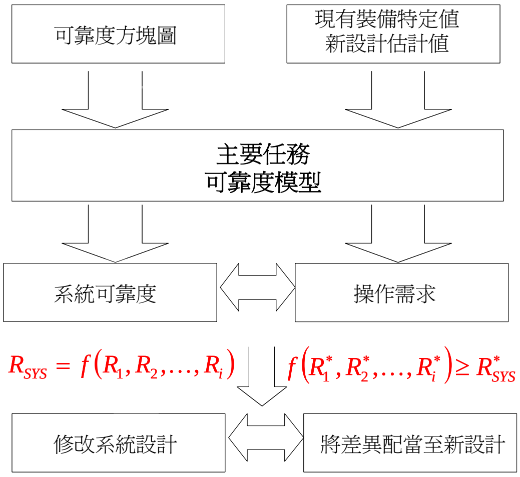


圖 3. 可靠度配當邏輯

可靠度設計

可靠度設計主要在避免設計不足或設計過度現象發生，使設計發生最大效能；本計畫分析傳動裝置負荷情況、機件公差對定位精度影響、機件疲勞問題，建立可靠度分析模型，結合強度、應力干涉理論與最佳化技巧，對該裝置進行可靠度分析與設計。可靠度設計是從功能供給(Supply)與需求(Requirement)的觀點進行設計，考慮設計的不確定性，如使用時的正常狀況、最惡狀況等，從機率觀點對產品進行安全性設計，避免設計不足或設計過當現象，如圖 4所示。

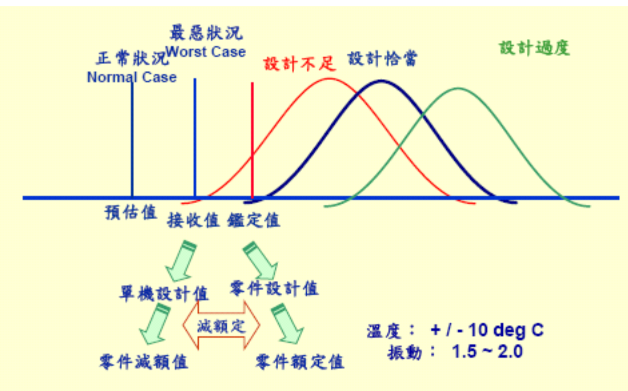


圖 4.可靠度設計

可靠度指標

進行可靠度設計時，可靠度設計變數可利用一次矩與二次矩來量測，當無明確的機率分佈資料時，可分析變數分佈型式，根據等效的常態分佈計算相對應的機率，可得到可靠度指標(reliability index) Z，如圖 5所示。

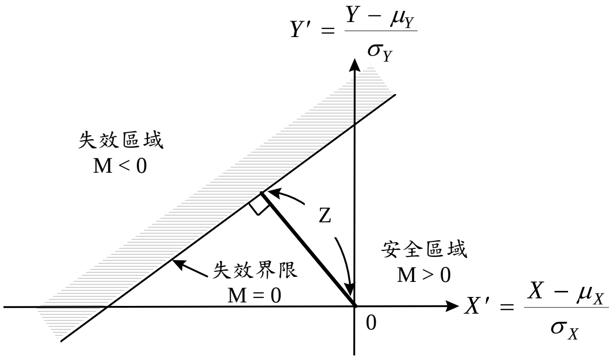


圖 5. 二次矩可靠度指標示意圖

依定義可靠度為 X>Y 時之機率，所以M=X-Y 即為安全邊際(safety margin)。系統的"安全狀態"可定義為 M>0，而"失效狀態"則為 M<0，當 M=0 時所代表的便是區分安全與失敗的邊界，即"臨界狀態"(limit state)，現在將變數轉換成標準化之變數，定義其為簡化變數(reduced variables)(X', Y')，可得簡化後變數的失效狀態及安全狀態空間。限制狀態 M=0，用簡化變數表示可寫成

$$\sigma_X X' - \sigma_Y Y' + \mu_X - \mu_Y = 0 \qquad (1)$$

其中 μ 及 σ 分別表示變數之標準差與平均值。由原點到此線的距離即可量測可靠度，此距離 Z 可由幾何關係求得：

$$Z = \frac{\mu_X - \mu_Y}{\sqrt{\sigma_X^2 + \sigma_Y^2}} \qquad (2)$$

當 X 及 Y 為常態分佈(normal distribution)時，Z 即為可靠度指標。可靠度指標 Z 得到後，便可計算系統的可靠度

$$R = \frac{1}{\sqrt{2\pi}} \int_{-z}^{\infty} \exp\left(-\frac{z_1^{\,2}}{2} \right) dz_1 \qquad (3)$$

或可簡化成

$$R = 1 - \Phi(-Z) = \Phi(Z) \qquad (4)$$

其中(Z)表示標準常態分佈函數。

可靠度最佳化設計

本計畫建立可靠度目標函數，將單變量設計問題轉成多維隨機變量設計問題，以進行可靠度最佳化設計。對 n 維隨機變量函數 $Y=g(y_1,$

$y_2,\ldots y_n$)，用泰勒級數對變數之平均值展開，忽略高階項，並假設各隨機變數彼此獨立，則 Y 的平均值(μ_Y)及變異數(σ_Y)可分別表示如下[5][6]：

$$\mu_Y = \overline{Y} = g(\overline{y}_1, \overline{y}_2, \ldots, \overline{y}_n) \tag{5.1}$$

$$\sigma_Y^2 = \sum_{i=n}^{n} \left(\frac{\partial Y}{\partial y_i}\right)_{\overline{y}_i}^2 \sigma_{i_j}^2$$
$$= \left(\frac{\partial Y}{\partial y_1}\right)_{\overline{y}_i}^2 \sigma_1^2 + \left(\frac{\partial Y}{\partial y_2}\right)_{\overline{y}_i}^2 \sigma_2^2 + \cdots\cdots + \left(\frac{\partial Y}{\partial y_n}\right)_{\overline{y}_i}^2 \sigma_n^2 \tag{5.2}$$

考慮強度(μ_X, σ_X)、應力分佈(μ_Y, σ_Y)，若可靠度目標是 R>=Φ(z)，定義變異係數 $\gamma = \sigma/\mu$，則由可靠度指標可導出下式

$$\left[1-(\gamma_Y z)^2\right]\mu_Y^2 - 2\mu_X\mu_Y + \mu_X^2 - (\sigma_X z)^2 \ge 0 \tag{6}$$

解上式，可得應力如下

$$\mu_Y \le \frac{\mu_X - \sqrt{\mu_X^2 - (1-\gamma_Y^2 z^2)(\mu_X^2 - \sigma_X^2 z^2)}}{1-\gamma_Y^2 z^2} \tag{7}$$

$$\mu_X \ge \frac{\mu_Y + \sqrt{\mu_Y^2 - (1-\gamma_X^2 z^2)(\mu_Y^2 - \sigma_Y^2 z^2)}}{1-\gamma_X^2 z^2} \tag{8}$$

針對不同設計組合方式，最佳化求解建模方式亦不同，主要形式如下：

- **串聯系統問題**

任一單元組件失效將導致整個系統失效，所有系統的狀態函數均需被滿足，系統失效率將是所有單元失效區域的聯集，其幾何關係表示如圖 6

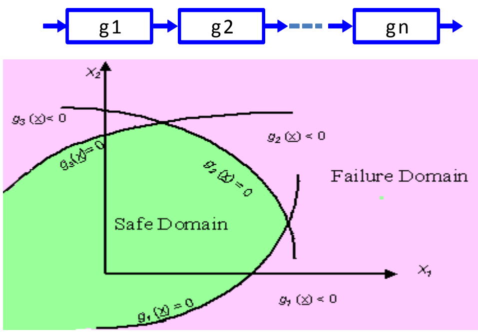


圖 6.串聯系統最佳化

- **並聯系統問題**

並聯系統中只有在所有元件失效時才會造成系統運轉停擺，只要有一系統的狀態函數被滿足即可，系統失效率將是所有單元失效區域的交集，其幾何關係表示如圖 8

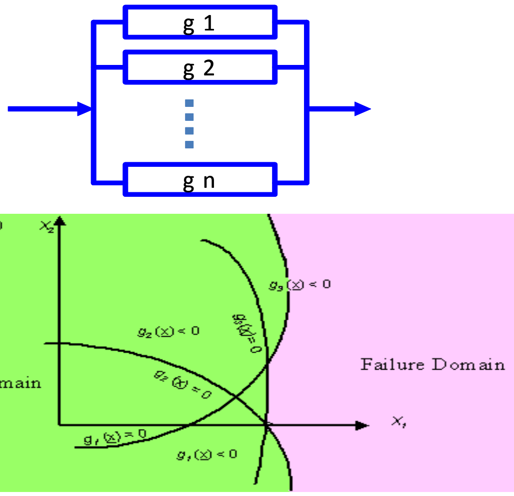
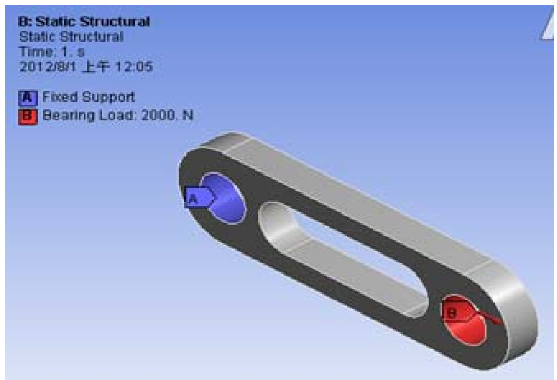


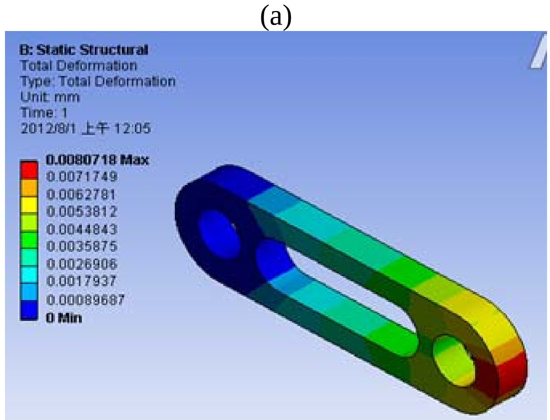
圖 8. 並聯系統最佳化

四、結果與討論

以一傳動裝置為例，包含驅動裝置、傳動機構，將其組合視為串聯關係，透過可靠度配置決定各機件可靠度需求，再對機件進行最佳化設計，針對此裝置之連桿，一端固定、一端受 2000N 之力，其負荷情況及總變形如圖 9所示，



(a)



(b)

圖 9. (a)負荷, (b)變形

設計變數(內槽圓弧半徑 7-9 mm) 對機件重量、應力、變形影響分析如下：

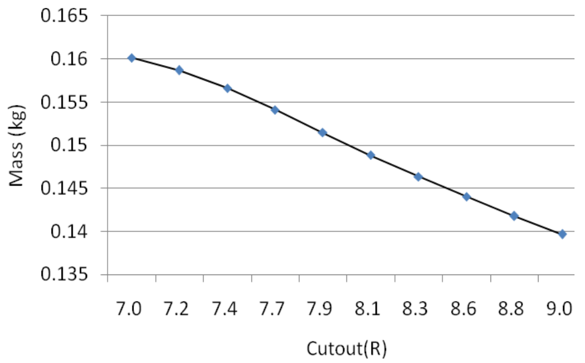


圖 10.不同弧徑之機件重量

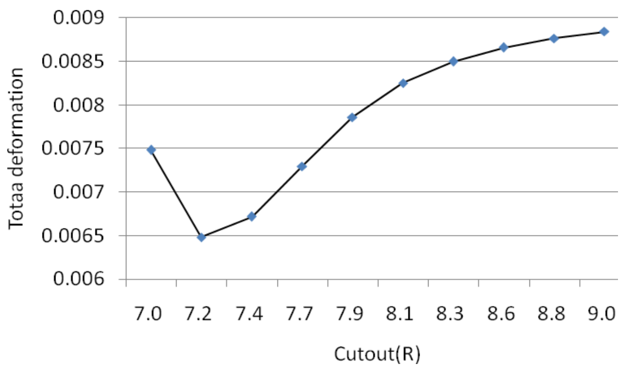


圖 11.不同弧徑之變形量

由圖 10分析結果可知，若以最小重量為目標，槽寬應最大，設計半徑應為 9 mm；若以總變形最小化為目標，則最佳設計點應為 7.2 mm(圖 11)。針對可靠度需求，具最小重量及變形量的最佳設解如下：

Cutout(R)	Mass(kg)	Stress(Mpa)	Deformation (mm)
9.00	0.1397	42.56	0.0087

五、計畫成果自評

本計畫藉由定性、定量分析程序來輔助系統設計的不確定性，研究機電系統可靠度設計相關技術，主要有可靠度設計與可靠度配置模型建立，機件可靠度設計與 CAD 分析模型的建立，二次矩可靠度估算法及在 CAD 分析的應用，可靠度最佳化數學模型與敏感度分析法的建立，多目標系統可靠度函數建立與最佳化求解。利用 ANSYS 對傳動裝置關鍵組件進行應力分析，分析設計變數對功能輸出影響，結合可靠度、最佳化等相關技術，建構系統可靠度最佳化設計發展架構。

本計畫研究內容與原計畫目標大致相符，其中系統設計的成本預估模式、可靠度最佳化、設計為維護等已初具規模，相關成果已投稿至 2012 全國自動化科技研討會，同時著手整理擬投稿至國內外相關期刊。此外，疲勞失效常是設計考慮因素，為預估疲勞壽命，本計畫亦對機件建立疲勞壽命預估方法，結合試驗與模擬數據，建立疲勞壽命可靠度預估法，相關研究成果已發表於國外研討會[9]。進一步擬結合信號量測、性能衰退預測模式，發展預兆式健康管理作業。

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出席國際會議心得報告

會議名稱：美國機械工程協會 2012 設計工程與電腦資訊工程研討會，
Proceedings of the ASME 2012 International Design Engineering Technical Conferences & Computers and Information in Engineering Conference IDETC/CIE 2012

發表論文：人工牙根疲勞壽命估計與可靠度之研究，A study of fatigue-life estimation and reliability analysis for dental implants
單位名稱：德霖技術學院 機械系
報 告 人：蔡有藤
會議時間：2012 年 8 月 12-15 日
會議地點： 美國芝加哥 凱悅飯店，Hyatt Regency, Chicago, Illinois, USA

一、參加會議經過

這次研討會是由美國機械工程協會(ASME)舉辦，該協會每年均在不同地點舉辦會議論談，主要探討和機械設計及電腦資訊工程相關領域技術，本次會議共有 10 個和機械領域相關研討會同時舉行，會議日期在 2012, 8/12-8/14 四天，地點在美國芝加哥凱悅飯店舉行。年初收到邀稿通知，覺得此研討會討論主題和我研究領域很接近，對我教學研究工作很有幫助，因此決定投稿參加。

投稿後於五月底收到稿件接受通知，就開始申請簽證、規劃參加行程，第一次去美國，對當地並不熟，申請過程花了一些時間查詢才完成。會議日期因是暑假(旅遊旺季)，訂購機票並不是那麼順利，較便宜機票均訂不到，花了很多天查詢與旅行社聯繫，最後才訂到機票(台北飛舊金山，再轉機飛芝加哥)，但機票費用超出預算很多(65,000)，出乎我的預料之外。

我搭的飛機是 8/10 日 PM11：30 班次，出發時我提前二個小時到機場辦理行李托運、登機，一切還算順利，飛行 12hr 到舊金山，進行轉機飛芝加哥，經過 4hr 飛行到芝加哥時已是清晨 6 點，下飛機後到服務台詢問到市區的路線，確定路線後就購票搭地鐵去預定的住宿旅館，一路上感覺一切還算 OK，沒發生什麼麻煩。放好行理後，就去研討會地點熟悉

環境，研討會地點離我們住宿地方步行大約 30 min，會場剛好有研討會在舉辦，詢問服務人員確認研討會地點無誤後，就去附近逛，瞭解這邊的地理環境。

芝加哥市區感覺不錯，高樓大廈很多，街道也很寬敞，靠近密西根湖處環境非常幽美，街上各式各樣的人多有，走在街上讓我覺得美國是一多元融合的的民主國家。

二、與會心得

這研討會目的是要帶給各國研究人員與實際應用者在機械領域經驗分享與互相學習機會，此次會議共有 10 個研討會同時舉行，約有 500 篇論文被評選在會議議程中發表，參加人員超過 30 個國家，參與的學者來自台灣、美加、歐洲、北歐、大陸、韓國、日本、東南亞、中東、印度、紐澳等國。相關會議主題有：車輛技術、電腦資訊工程、設計自動化、設計為製造，設計理論，機構和機器人、振動噪音等。第一天上午是國外學者主題發表(Keynote presentation)，下午參觀芝加哥復健學院，我均有參加。會議論文被安排在後三天 19 個 Sessions 舉行，我的文章被排在第三天 Session 1 上午第一時段發表，由於每個時段都有很多個 Sessions 同時在進行，所以我只能選則較有興趣的論文去聽。

這次研討會在芝加哥凱悅飯店的國際會議廳舉行，議場周邊及硬體設施都很棒，是一個適合舉辦國際會議的地方，主辦單位所提供的軟硬體設施、邀請的專題演講者、發表的論文，我覺得都不錯。令我感到不好的是會議手冊編排有點亂，查詢論文場次、地點、時間很麻煩，要花很多時間，餐點提供則是普通，並無五星級豪華飯店的等級。透過參加國際研討會，讓與會者有機會接觸到不同國家的研究人員、學者，經由會議討論交流，了解不同的思考方式，對於學術研究人員來說具有正面積極的幫助。

此次論文發表每一個人有 20 min 時間，很多論文都是教授本人發表，這可能是芝加哥環境很美，很多教授都想趁機到此走走吧，這無形增加會議品質，因教授講的通常比較清楚，聽者比較聽的懂演講者在講什們。我講的主題是人工牙根疲勞壽命估計與可靠度之研究，主要探討人工牙根在不同應力下之疲勞壽命，建立人工牙根退化模型及 S-N 曲線數學模式，因已參加過多次國際會議，加上事前適當的準備，發表論文已能從容應付，比起日本人、韓國人、新加坡人毫不遜色。參加國際研討會最大的障礙是語文，在會議休息期間與不同國家學者交談，我覺得台灣和亞洲國家學者的英文差不多，因為英文不是母語，溝通時有時會詞不達意，但結合肢體語言大多能瞭解對方意示，但和以英文為母語的歐美國

外學者交談，你會發現差很多，他講的英文較好也較能了解，你講的若表達不是很清楚，他亦能針對你的問題回答。

與會期間我偶爾會提出問題，大部分演講者都會耐心回答問題，會後中間休息及午餐時間，我也和很多學者交談、交換名片，這讓我瞭解對方的研究領域與其國家文化特色，增加不少生活知識與專業常識；由於此次是在英語系國家舉辦，很多溝通、問題解決都需靠英文，透過會議參加，讓我有機會練習使用英文，我覺得這是很難得的機會。

語文能力在國際場合是很重要的，若能用很流暢且字正腔圓的英文介紹研究內容，會讓人印象深刻，這方面需靠平常的努力與學習，唯有加強英文能力，才有辦法和國外的菁英學者相提並論。會後，與另一位台灣教授一起去參觀芝加哥一些景點，有博物館、紀念塔、港口等，這些景點都不遠，用走路就可到，讓我印象比較深刻的是博物館內原住民文化館，裏面有很多原住民手工藝製品，非常具有可看性，其中有一個從過去到現在變化劇場，解說員跟我講發展演變，讓我印象深刻。

三、建議

我有機會參與此次國際研討會，要感謝國科會對人才培育及計畫經費的支持，學校相關行政作業的協助，才讓我得以成行。在會議期間也讓其他人知道有德霖技術學院這所學校，這無疑是一項成功的學校外交。建議學校有機會能舉辦國際研討會，邀請國際知名學者來專題演講，讓師生有更多參與的機會，將可增廣見聞，提升學校國際知名度、能見度。另外學校若能簡化經費補助申請，提供較大的彈性及補助讓學校老師去參與，這對學校整體發展將有正面積極幫助的。

四、攜回資料名稱及內容

會議論文光碟一片。

附件：活動照片

註冊報到

論文發表會場前留影

會場一嶼

中場休息

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Proceedings of the ASME 2012 International Design Engineering Technical Conferences & Computers and Information in Engineering Conference
IDETC/CIE 2012
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A STUDY OF FATIGUE-LIFE ESTIMATION AND RELIABILITY ANALYSIS FOR DENTAL IMPLANTS

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ABSTRACT

Recently, dental implants (DIs) are extensively utilized on edentulous patients. The bio-compatibility & physical properties of DIs are severely specified since it belongs to the products of biomedicine. Generally, DIs must pass a series of tests before they are approved to use in human body. In this paper, a method of probabilistic fatigue-life estimation was proposed to fulfill reliability life prediction of DIs. The probabilistic form of fatigue-life evaluation is developed based on material constants namely fatigue strength coefficient and fatigue strength exponent. The procedure is developed based on the shift of the fatigue-life curve to the desired value of the probability of occurrence. This estimation model yields the life distribution in respect of the scatter of the cyclic properties of DIs.

The CAD models of DIs are first constructed to perform computer simulation analysis for establishing the fracture spots. The stress analysis and life estimation were carried out by ANSYS software. The simulation results are further compared with the experimental data obtained by fatigue testing to determine the estimated model of fatigue life. The parameters of the model were determined by linear regression method based on the combination of the simulated and experimental data. The reliabilities of DIs were further investigated to provide an index of life-safety of DI at different cyclic loads. The analyzed results may be useful while programming the fatigue testing of DIs.

Keywords: fatigue life, reliability, probabilistic form, dental implants.

1. INTRODUCTION

It is essential to ensure that the physical strength of implant design is endowed with sufficient resistance against fatigue failure. To assess whether the fatigue-life of implant design is sufficiently greater than the desired operational endurance, the relationship between the cyclic loads and the life must be determined [1]. Considering the actual operational conditions, a probabilistic approach of fatigue-life estimation is developed to meet the material properties and the loading frequently being random in nature, i.e. the life estimation should be in the form of a distribution function [2]. Based on the distribution function, the reliabilities of fatigue-life at different stress levels would be able to be calculated for which providing a risk assessment of DIs in use.

To satisfy the conditions the designed life > the required life, a design may need to be modified either from materials improved or from geometry structure redesigned. It means that the physical characteristics of a design have to meet the needs. It is not always possible to alter the operational loading to suit the design concept since the loading is governed by operational conditions. Among the strategies of designs enhanced, the scatter of material's mechanical properties plays a crucial role because the magnitude of the scatter could significantly affect the variety of fatigue-life and consequently the conditions of reliable operation for a design [3].

DI is an extensive alternative for the edentulous regions. In addition to the stability of the implants, esthetic is also another important issue. Most of the DIs was manufactured using pure Titanium by a series of manufacturing processes such as lathing, milling, punching, thread-cutting and surface treating, etc [4]. The DI must pass a dynamic fatigue test specified in ISO 14801 for ensuring the quality and the safety [5]. In this paper, the fatigue life of DIs was investigated by integrating the information of computer simulation analysis and fatigue test. Computer simulation analysis is to understand the life properties of DIs so that the fatigue test can be well programmed accordingly. The purpose is to construct an estimation model of fatigue life for DIs using less test cost, meanwhile keeping the accuracy in estimation.

The CAD models of the DIs are first constructed to perform stress analysis and fatigue life analysis. The geometry shapes and the dimensions of the DIs are offered by a dental clinic which is mastered in manufacturing of DIs [6]. The software ANSYS was used as a tool to analyze the stress distribution and the related life characteristics [7]. The critical components the abutment was determined according to the results in stress analysis. The fatigue life of the critical component was further studied as well as the safety factor and the sensitivity of loads. The estimated lives under different fracture models including Solderberg, Goodman and Gerber [8] were compared with the data obtained by fatigue tests to decide the optimal one used in life analysis.

A model of fatigue-life estimation was developed based on the corresponded relationship between the loads related to the maximum load and the estimated lives. The model's parameters are decided by regression analysis method according to the combined data obtained by computer simulation analysis and fatigue test. The evaluated results are compared with the experimental data to identify the accuracy of estimation as well as to modify the boundary conditions of ANSYS analysis accordingly[9]. The estimated values of fatigue life and the standard errors were used to calculate the reliability of DIs depending on time. Validly, the proposed method can be used to evaluate the operational reliability of DIs from the material's fatigue perspective for which to provide a risk assessment of DIs in use.

2. PROBABILISTIC FORM OF FATIGUE LIFE

To calculate fatigue life of a design, the behavioral characteristics of material under cyclic loading need to be investigated. The fatigue life of a design is estimated generally based on the concepts of local stress and strain. The fatigue life curve representing material characteristics can be expressed with the equation proposed by Morrow[10] as

$$\varepsilon_a = \frac{\sigma_f^{'}}{E}(2N_f)^b + \varepsilon_f^{'}(2N_f)^c \qquad (1)$$

The equation is the summation of two separated curves for elastic strain-amplitude life ($\varepsilon_a^e - 2N_f$) and for plastic strain-amplitude life ($\varepsilon_a^p - 2N_f$). The parameter ε_a is the total strain amplitude, N_f is the number of cycles to fatigue failure, $\sigma_f^{'}$ is the fatigue strength coefficient, b is the fatigue strength exponent, $\varepsilon_f^{'}$ is the fatigue ductility coefficient and c is the fatigue ductility exponent.

The parameters $\sigma_f^{'}, b, \varepsilon_f^{'}, c$ represent the cyclic properties of material which usually are determined by linear regression method based on the experimental dependence $N_f = f(\sigma_a)$ and $N_f = f(\varepsilon_a)$. In reality, the material properties are random in nature. The degree of this randomness is given by the magnitude of the scatter of the experimental points along the regression line, representing the fatigue-life curve of material of a design. Generally, the values, $\sigma_f^{'}, b, \varepsilon_f^{'}, c$ obtained by regression analysis are corresponding to the 50 % probability of occurrence [11].

However, the computational estimations of fatigue damage, obtained by employing solely the parameters of the regression line, could be overestimated or underestimated depending upon mutual positions of the regression line and the experimental data. Therefore, fatigue damage calculations of material should be interpreted with probabilistic forms. By this approach of parameters determination, the fatigue life of a design can be expressed as a distribution incorporating the scatter of the material's cyclic properties.

The Basquin's equation often is used to express the low stress cycle fracture [12]. As a result, the elastic strain-amplitude life can be rewritten as the form

$$\sigma_a = \sigma_f^{'}(2N_f)^b \qquad (2)$$

where σ_a is the stress amplitude. Considering the experimental points $(\sigma_{a_i}, 2N_{f_i})$, the scheme for determination of the regression line of the P (%) probability of occurrence is shown in **Fig. 1**. Here, RL(50%) is the regression line; RL(1%) the regression line shifted to the value of P = 1 %; s_N the standard deviation of distribution; d(1%) multiple of standard deviation determining the value of P = 1 %; PDF probability density function (log-normal distribution of N_f)[13].

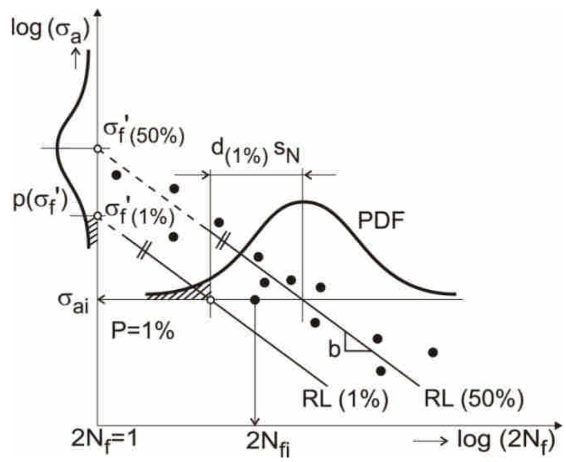


Figure 1. Scheme for determination of the regression line for P(%) probability of occurrence

While using this model, several assumptions are given as the following.

(1). the life (that is the number of cycles to fatigue failure N_f on a harmonic cycle load of amplitude σ_a) has log-normal distribution, and

(2). the values of the material constants $\left(\sigma_f^{'}, b\right)$ for the P % probability of occurrence are obtained based on the shift of the regression line to the desired value of P %.

This principle is schematically depicted for Basquin's relation in **Fig. 1**. The material constants $\left(\sigma_f^{'},\ b\right)$ the P% ranging from 1% to 99% are obtained from the set of the experimental points $\left(\sigma_{a_i}, 2N_{f_i}\right)_{i=1}^{n}$ for the fatigue-life curve in Eq.(2). Based on the methodology, only the value of the fatigue strength coefficient $\sigma_f^{'}$ is a function of P %, the value of b remains unchanged for all values of probability.

When utilizing the set of fatigue-life experimental points to determine the regression line in accordance with the scheme in **Fig. 1**, the stress amplitude σ_a must be considered as an independently variable and the number of cycles to failure as dependently variable. Therefore, it is necessary to perform linear regression in the coordinate system $(2N_f) = f(\sigma_a)$. The regression line equation can be expressed as

$$\log(2N_f) = -\frac{1}{b}\log\sigma_f^{'} + \frac{1}{b}\log\sigma_a \tag{3}$$

Statistical evaluation of the set of n experimental points $\left(\sigma_{a_i}; 2N_{f_i}\right)_{i=1}^{n}$ leads to the form

$$\log(2N_f) = A_0 + B_0\log\sigma_a \tag{4}$$

The evaluated values of constants A_0, B_0 stands for the material at 50 % probability of occurrence for $\left(\sigma_{a_i}, 2N_{f_i}\right)_{i=1}^{n}$. The parameters of fatigue strength would be

$$\begin{aligned} A_0 &= -\frac{1}{b}\log\sigma_f^{'} \ \Rightarrow\ \sigma_f^{'} = 10^{-bA_0}, \\ B_0 &= \frac{1}{b} \ \Rightarrow\ b = \frac{1}{B_o} \end{aligned} \tag{5}$$

The equation of the regression line that was shifted is as follows:

$$\log(2N_f) = A_0 \pm d_{(P\%)}s_N + B_0\log\sigma_a \tag{6}$$

where $d_{(P\,\%)}$ is the constant expressing the regression line shift (in multiples of the s_N standard deviation) to the required value of P %. For P = 50 % the value $d_{(50\%)} = 0$. The signs (-) and (+) are valid for determination of the lower and upper interval limits, respectively, of the confidence interval.

For the set of n experimental points $\left(\sigma_{a_i}, 2N_{f_i}\right)_{i=1}^{n}$, the s_N is given by the following relation

$$s_N = \sqrt{\frac{\sum_{i=1}^{n}Y_i^2 - A_0\sum_{i=1}^{n}Y_i - B_0\sum_{i=1}^{n}X_iY_i}{n-2}} \tag{7}$$

where Y = $\log(2N_f)$, X = $\log\sigma_a$, and the constants A_0 , B_0 are the regression line parameters in Eq. (4). It follows the above-noted assumptions that the material constant b will remain unchanged for all values of P %. The value of $\sigma_f^{'}$ for P % is obtained using the evaluated constant A_0 and the relevant value of the $d_{(P\,\%)}$ constant. Let us set

$$A_0 \pm d_{(P\%)}s_N = A_1 \tag{8}$$

The value of constant A_1 with respect to Eq.(5) would be

$$A_1 = -\frac{1}{b}\log\sigma_{f(P\%)}^{'} \tag{9}$$

The value of the fatigue strength coefficient for P % would be

$$\sigma_f^{'} = 10^{-bA_1} = 10^{-b(A_0 \pm d_{(P\%)}s_N)} \tag{10}$$

By the same procedure, it is possible to obtain the set of material constants pertaining to Manson-Coffin's curve [14]. Let us also note that the assumption of log-normal distribution of fatigue-life, used in our derivations, is a widely accepted practice.

3. TEST SPECIFICATIONS OF DIS

The parameters of fatigue-life curve are evaluated by regression analysis method based on the experimental data of fatigue test. Focusing on DIs, International standard ISO 14801 [5] has specified a method of fatigue testing for signal post DIs of the pre-manufactured prosthetic components.

3.1 TESTING SCHEMA

Testing shall be carried out on the worst-case conditions within the recommended use if a part of the DI system is available in various dimensions and /or configuration. **Fig.2** shows the schema of testing setup for DIs. The numbers (1-6) in **Fig.2** indicate the loading device, the nominal bone level, the connecting part, the hemispherical loading member, the dental implant body and the specimen holder, respectively. The loading force (F) of the testing machine shall be applied in a way including no lateral constraints occurs and the loading center is well-defined, such that the moment y can be measured or calculated.

The loading force shall be applied to the hemispherical loading surface by a loading device with a plane surface normal to the loading direction of the machine. The loading device shall be unconstrained in the transverse direction, so as to not reduce the magnitude of the applied bending moment. The hemispherical loading surface and the surface of the loading device shall be examined visually after each test to ensure that permanent deformation has not occurred. For an DI body and /or connecting part that locks rotational symmetry around either the central longitudinal axis of the implant body or the axis of nominal prosthetic loading. The loading geometry shall be selected to test the worst case compatible with the intended use of the implant.

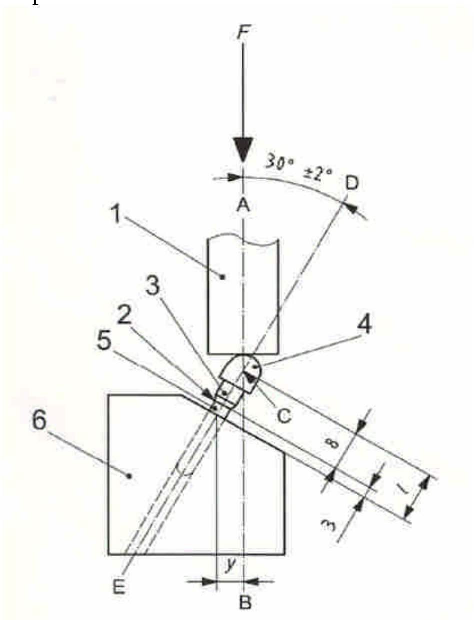


Figure 2. Schematic of test set-up for DIs

The bone-anchoring part of the specimen shall be fixed in a rigid clamping device. If an embedding material is used, it shall have a module of elasticity higher than 3 GPa. The geometry of the clamping device shall be such that the testing geometry is achieved. The clamping device shall be designed so as not to deform the test specimen. The dimension of the loading member shall be chosen to define a distance, l=11±0.5 mm from center of the hemisphere to the clamping plane. The moment arm y is defined as $l \times \sin 30^\circ$ for the standard configuration. As a result, the bending moment M is expressed as $M=y \times F$.

3.2 LOADING PROCESS

Moreover, fatigue testing shall be carried with a unidirectional load. The load shall vary sinusoidal between a normal peak value and 10 % of the value. The loading frequency shall be more than 15 Hz. The general principles for fatigue testing were described in ISO 1099. The data for a load-cycle diagram at a series of loads are generated until a lower limit (maximum endured load) is reached at which at least three specimens survive and none falls in the specified numbers of cycles. Plot the measured points in a load-cycle diagram. For testing conducted at frequency ≧ 2 Hz testing shall be conducted to 5×10^6 cycles.

A proper starting load is 80% of the load to failure in a static test performance using the same test geometry. Subsequent test typically are performed at lower loads resulting in a fatigue curve. At least two and preferably three specimens shall be tested at each of at least four loads.

Identify the critical failure point and the location of failure initiation. Failure is defined as material yielding, permanent deformation, loosing of the implant assembly or fracture of any component. Draw the load cycle curve to show the maximum load at which the DI system with withstand 5×10^6 cycles. Every specimen (at least three) rested at the maximum endured load shall reach the specified number or cycles with no failures. The fatigue properties of the test objects are determined by testing a number of specimens at different values of the peak load.

4. FAILURE ANALYSIS TO DIS

DIs must undergo dynamic fatigue test for commercializing to the market. To construct the model of life estimation, computer simulation analysis is performed to picture out the stress distributions and the fatigue lives of DIs. The purpose is to observe the possible fracture spots so that the fatigue test can be planned accordingly for reducing the test cost.

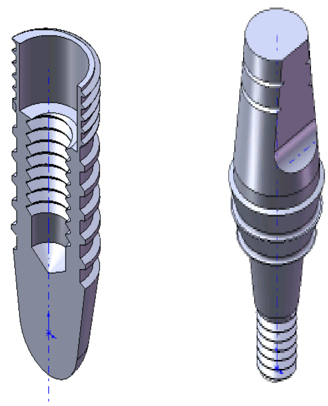


Figure 3. Geometry shapes of a DI

4.1 STRESS ANALYSIS

The DIs primary enclose two parts the fixture and the abutment (see **Fig.3)**. The fixture is screwed to the dental bone to sustain and fix the abutment. The geometry shapes and the dimensions of the DIs are given by a dental clinic which is mastered in designing and manufacturing to DIs. The CAD models are leaded to the software ANSYS for analyzing the deformation, stress and fatigue life of the DIs. The CAD models including the base and the hemisphere cup, and the test setting are shown in **Fig.4.** The material properties for these components including elastic module (E), yielding strength (σ_y), ultimate strength (σ_u) must be given. In this example, the hemisphere cup is designed with structural steel where the material properties {E, σ_y, σ_u} are {$2*10^5$, 250, 460} Mpa, the abutment and the fixture with pure titanium being {$1.24*10^5$, 657, 762} Mpa, and the holder with polyethylene being {$4.4*10^3$, 75, 100} Mpa.

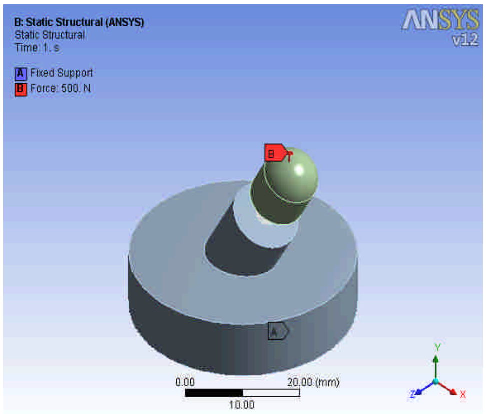
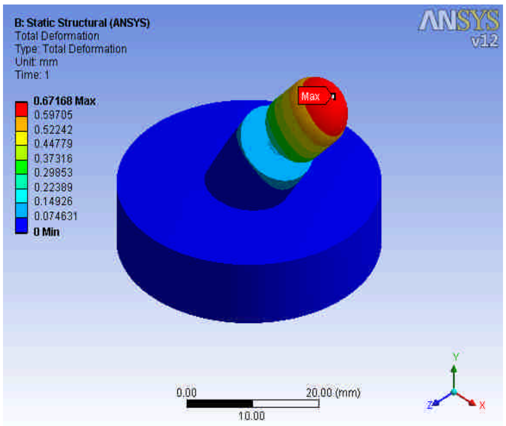
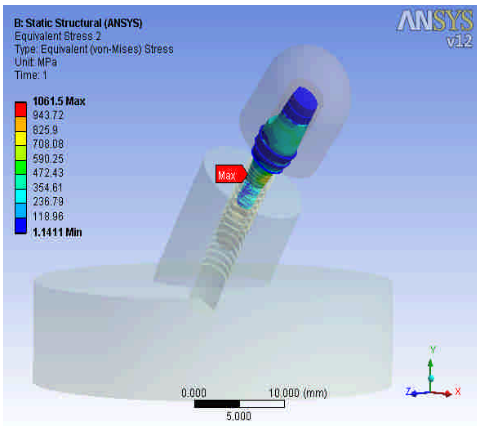


Figure 4. Combination diagram and testing loads



(a)



(b)

Figure 5. (a) Total deformations of system, (b) Stresses of the abutment

The contact types of these components in analyzing are set to no separation. The meshes of these components are automatically generated by the system. The analyzed results about the total deformation of system and the von-Mises stress for these components are shown in **Fig.5**. The results in **Fig.5** show that the total deformation of the system is 0.672 mm. Meanwhile, the maximum stress of the system is 1061.5 Mpa which occurs at the neck of the screw of the abutment. The results show that the abutment would fail prior to the fixture because its maximum stress is bigger than the fixture's maximum stress. Therefore, the abutment was regarded as the key component of the DIs as well as its life is used to represent the life of the DIs. The life investigation is done by the fatigue model in ANSYS.

4.2 FATIGUE ANALYSIS

There are two variables need to be set for implementing fatigue life analysis. One is to determine the load types. The other is to set the analyzed models for considering the effects of mean stress[16]. According to the specifications, the loads are to vary between the maximum load and the 10% of the maximum load. The load setting and the fracture line, for example, Goodman model are shown in **Fig.6.**

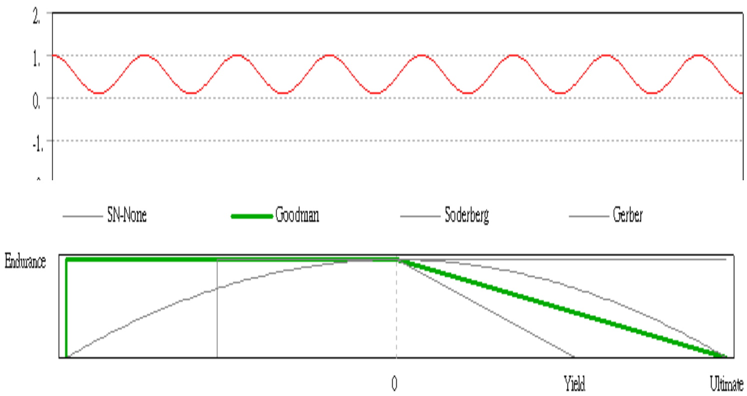
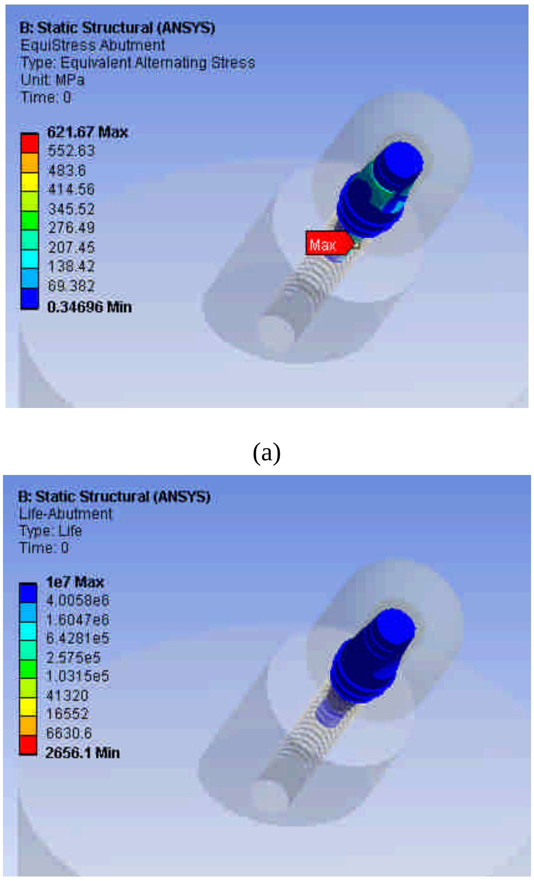


Figure 6. The loading setting and the fracture models



(a)

(b)

Figure 7. (a) the equivalent stresses, (b) the fatigue lives

The maximum load is set to 547 N which is determined according to the results of the static test. The fatigue limits of pure Titatium used as the materials of the abutment and the fixture are set to 350 Mpa subjected to life 10^7 cycles which is obtained from the atlas of the fatigue curves[15]. The estimated lives of the abutment at different fracture models are compared with the test data obtained at 80 % the maximum load. The results show that the life estimation by Gerber model is the most closed one to the life values of real test. As a result, the fatigue lives under various loads are estimated using Gerber facture model. For example, the maximum equivalent stress at loads between 43.8 N and 438 N (80% of the maximum load) was 621.7 Mpa, and the corresponding fatigue life at the point was 2656.1 cycles (see **Fig.7**). The safety factor of fatigue failure related to $5*10^6$ cycles was 0.7389. The sensitivity of the fatigue lives related to the load varieties from 80% to 120% of the load 438 N is shown in **Fig.8**.

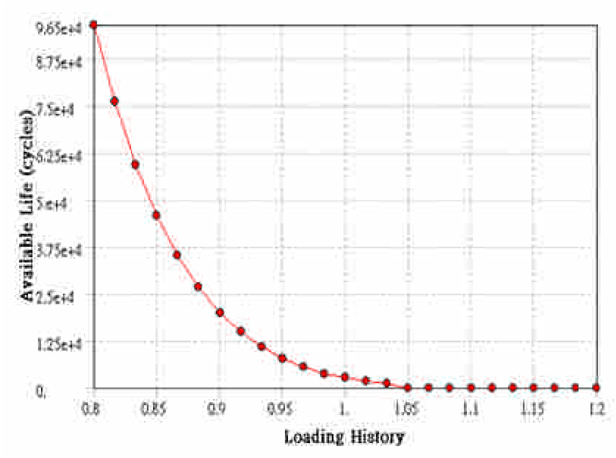


Figure 8. Sensitivity of fatigue lives to loading

The results show the estimated life, for example, at 80% of load 438 N, i.e. 350N, 64% of the maximum load, would be $9.65*10^4$ cycles. The analyzed results at different load scales of the maximum load in this example were listed as follows.

Scales	80%	75%	73%	70%	64%	60%	59%	50%
F_{max}	438	410	397	383	350	328	323	274
N	2.7E+03	1.0E+04	1.8E+04	3.1E+04	9.7E+04	2.6E+06	5.4E+06	1.0E+07

Based on the analyzed results, the lowest maximum load to satisfy the expected life need would be 323 N (59% of the maximum load).

5. LIFE ESTIMATION

To fulfill reliability analysis, the distributions of fatigue life must be established. The life distributions can be decided by distribution identification to the experimental data. In this paper, the dynamic fatigue tests on high stress were done to acquire the experimental data. The experimental data are combined with the simulated data to construct the life model of the DIs.

5.1 FATIGUE TESTING

The maximum load in testing is determined by the static test. In static test, a preload of 5 N was applied first, then applied compressive load at a rate of 1 mm/min on the loading cap until the implant failure or the force decreased below 20 % of the maximum load. The data were generated the load versus displacement diagram. The maximum force of the static test was determined by the load versus displacement diagram. The maximum load (547 N) is the mean of the test values of four samples.

The fatigue tests were then carried on according to the maximum load of the static test. The cyclic loading is set to 80% of the maximum load. R value (F_{max}/F_{min}) was set to 10. The test frequency was 10 Hz. The implant which endures over $5*10^6$ cycles of load was regarded as pass. Fracture of any component of achieve material yielding was determined as failure. The material yielding which is defined as the applied load at the intersection point of the stress-strain curve with the slope line of the offset displacement of 0.2% the strain. The yielding displacement was 0.525 mm which is calculated from the load-displacement diagram of the static test. The tested results of fatigue life at the stress level for 3 specimens were

N={834, 2255, 4522}

The magnitude of the scatter of cyclic properties is an important parameter in fatigue-reliability evaluation because it affects the shape of the distribution function. The magnitude of the scatter about the experimental data may be generated mainly depended on

(1). the method of DIs fixed in testing,
(2). inhomogeneity of the mechanical properties of the basic and added materials,
(3). compliance with requirements of the testing process, and other factors.

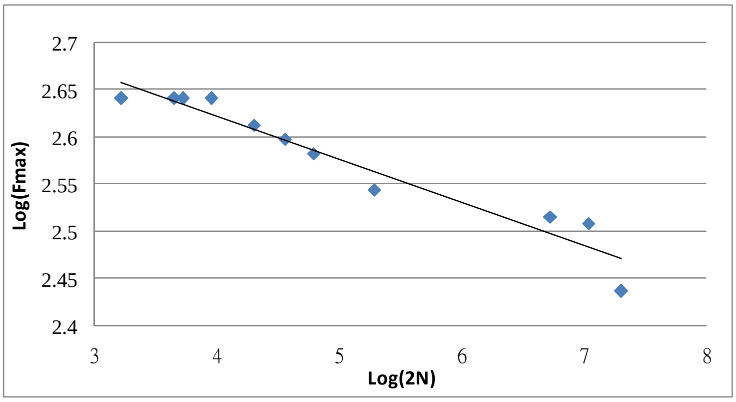


Figure 9. Fatigue life curve of the DI

5.2 RELIABILITY ANALYSIS

The fatigue-life model of DIs is constructed based on Basquin equation Eq.(3). The parameters are determined by linear regression method based on the combined data acquired by fatigue testing and by fatigue life analyzing in ANSYS. The estimated results of trend line of fatigue lives with respect to loads were shown in **Fig.9**. The fatigue life model would be

$$\log(2N_f) = 57.33\text{-}20.31\log(F_{\max})$$

The correlated coefficient for the life curve and the data is R^2=0.932. The result indicates that the life model possesses high correlation with the life data which is suitable to be used in estimating the fatigue life of the DIs. According to the results, the fatigue strength coefficient and the fatigue strength exponent could be evaluated by Eq.(5) as $\sigma_f^{'} = 665$ and b= -0.049, respectively. The estimated lives at various maximum loads based on this model were listed as follows.

Fmax	438	410	397	383	350	328	323	274
N	2.5E+03	9.1E+03	1.8E+04	3.7E+04	2.3E+05	8.5E+05	1.2E+06	3.4E+07

Particularly, the maximum load for the expected life $5*10^6$ cycles would be 301 Mpa. Meanwhile, the estimated values of 95% confidence intervals for $\sigma_f^{'}$ *and b* based on the model would be (81.8, 15657.2) and (-0.041, -0.062), respectively. Based on the estimated values of $\sigma_f^{'}$, The probabilistic lives of the DI at different P % of confidence intervals can then be estimated by Eq.(6). Here, the evaluated standard deviation for the fatigue life was calculated by Eq.(7) which is $s_{\log(N)}$=0.395.

Considering the variation of materials, the fatigue lives at each load can be regarded as lognormal distribution. Combining the estimated values of P=50 % life and the standard deviation of estimation, the reliabilities of fatigue life at different loads can be further computed. Define the characteristic parameters of life for lognormal distribution as (μ_Y，σ_Y). The parameters can be estimated by

$$\mu_Y = \log \mu_X - \frac{1}{2}\sigma_Y^2$$

$$\sigma_Y^2 = \log\left[\left(\frac{\sigma_X}{\mu_X}\right)^2 + 1\right]$$

where the parameters (μ_X, σ_X) are the mean and the standard deviation of life at normal distribution. The standard deviation of fatigue life can be set as σ_Y = $s_{\log(N)}$ as well as the mean representing the 50% life as μ_Y =log(N). As a result, the reliabilities of DIs at different loads can be evaluated by

$$R(t) = \Phi\left(\frac{\log(N) - \mu_Y}{\sigma_Y}\right)$$

The evaluated results of reliability, for example, loads at 80%, 70% and 60% of the maximum load were shown in **Fig. 10**. The results are useful to provide an index of risk assessments for DIs in use.

Normally, the magnitude of standard deviation of the DIs material's cyclic properties greatly affect the fatigue reliability. Based on the proposed methodology, the maximum permissible value of load was determined to ensure that the probability of premature fatigue failure occurrence would not exceed the allowable value.

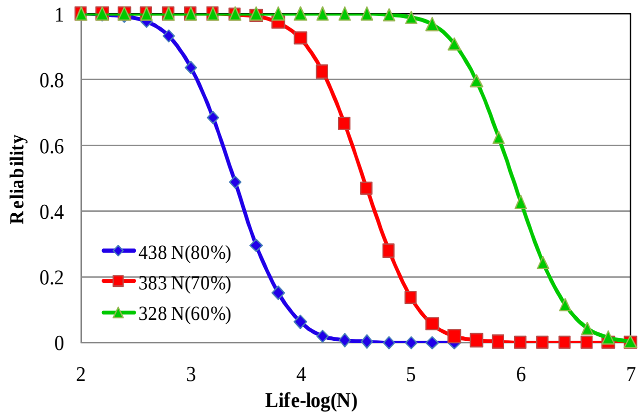


Figure 10. Reliabilities of the DI at different loads

6. CONCLUSION

The common problems of fatigue testing are long time-consuming and large cost burdening, especially at low stress fatigue testing. To reduce the testing time and expenditure, an effective approach is to acquire the related information by CAE analysis for decreasing quantity of testing. This paper proposed a fatigue-life estimation method based on the combined information of CAE analysis and fatigue life testing for DIs. The probabilistic forms of fatigue-life were reported to provide an estimation of fatigue life. The results of computer simulation analysis show that the abutment is the key component of DIs since it fails prior to the fixture. The fatigue lives of the abutment at different loads are further analyzed to construct the life estimation model of DIs

The analyzed results show that the lives at high stress evaluated by the Gerber model are the most closed to the lives obtained by fatigue testing. As a result, the life data of DIs as well as the equivalent stresses, safety factor and the sensitivity of lives to loads are estimated based on the model. The life estimation model was then determined by linear regression method based on the combined data obtained by both computer simulation and fatigue testing. The properties of using the combined information are able to ensure the accuracy of the life estimation model, meanwhile, reduce the quantity of testing Moreover, the method of reliability evaluation of fatigue life for DIs was proposed in cooperation with the standard deviation of the estimated lives. The studied results may be helpful to program fatigue testing and provide a risk index of DIs under various cyclic loads.

ACKNOWLEDGEMENTS

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國科會補助計畫衍生研發成果推廣資料表

日期:2012/08/17

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<tr><td>計畫編號: 100-2221-E-237-001-　　　　學門領域: 自動化工業工程技術</td></tr>
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100 年度專題研究計畫研究成果彙整表

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計畫名稱：機電系統可靠度設計分析與預兆式健康管理作業	

成果項目			量化：實際已達成數（被接受或已發表）	量化：預期總達成數(含實際已達成數)	量化：本計畫實際貢獻百分比	單位	備註（質化說明：如數個計畫共同成果、成果列為該期刊之封面故事...等）
國內	論文著作	期刊論文	0	0	50%	篇	
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		研討會論文	1	1	100%		
		專書	0	0	100%		
	專利	申請中件數	0	0	100%	件	
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	技術移轉	件數	0	0	100%	件	
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	參與計畫人力（本國籍）	碩士生	2	0	20%	人次	
		博士生	0	0	100%		
		博士後研究員	0	0	100%		
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		專任助理	0	0	100%		

其他成果 (無法以量化表達之成果如辦理學術活動、獲得獎項、重要國際合作、研究成果國際影響力及其他協助產業技術發展之具體效益事項等，請以文字敘述填列。)	無

	成果項目	量化	名稱或內容性質簡述
科教處計畫加填項目	測驗工具(含質性與量性)	0	
	課程/模組	0	
	電腦及網路系統或工具	0	
	教材	0	
	舉辦之活動/競賽	0	
	研討會/工作坊	0	
	電子報、網站	0	
	計畫成果推廣之參與（閱聽）人數	0	

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請就研究內容與原計畫相符程度、達成預期目標情況、研究成果之學術或應用價值（簡要敘述成果所代表之意義、價值、影響或進一步發展之可能性）、是否適合在學術期刊發表或申請專利、主要發現或其他有關價值等，作一綜合評估。

1. 請就研究內容與原計畫相符程度、達成預期目標情況作一綜合評估 ■達成目標 □未達成目標（請說明，以 100 字為限） 　　□實驗失敗 　　□因故實驗中斷 　　□其他原因 說明：
2. 研究成果在學術期刊發表或申請專利等情形： 論文：■已發表 □未發表之文稿 □撰寫中 □無 專利：□已獲得 □申請中 ■無 技轉：□已技轉 □洽談中 ■無 其他：（以 100 字為限） 發表國際研討會 1 篇、國內研討會 1 篇、國際期刊 1 篇（SCI）。
3. 請依學術成就、技術創新、社會影響等方面，評估研究成果之學術或應用價值（簡要敘述成果所代表之意義、價值、影響或進一步發展之可能性）（以 500 字為限） 本計畫研究內容與原計畫目標大致相符，其中系統設計、可靠度最佳化、設計維護等已初具規模，相關成果已發表於國內外研討會及期刊，由於疲勞失效常是設計考慮因素，為預估疲勞壽命，本計畫亦對機件建立疲勞壽命預估方法，結合試驗與模擬數據，建立疲勞壽命可靠度預估法，進一步擬結合信號量測、性能衰退預測模式，發展預兆式健康管理作業。