第七章 輪胎噪音頻譜分析
Tire Noise Spectrum Analysis

摘要（**Abstract**）

輪胎噪音頻譜分析是現代輪胎聲學研究中最具診斷價值與工程指導意義的關鍵技術之一，其核心價值不僅在於量化噪音能量的分布，更在於揭示噪音生成機制、時間演變特性與人類主觀感知之間的內在關聯。相較於單一的 A 加權噪音值指標，頻譜分析能夠將輪胎滾動噪音這一高度非平穩、寬頻且多機制耦合的聲學現象，分解為具物理意義與工程可操作性的頻率成分，從而為噪音源辨識、設計優化與品質評估提供科學依據。

本章系統性闡述輪胎噪音頻譜分析的理論基礎、分析方法與實務應用。首先，從聲學基本理論出發，說明聲壓、噪音值、頻率與波長之物理意涵，以及 A 加權、C 加權與無加權量測在噪音評估中的角色與限制。進一步指出，輪胎噪音能量主要集中於 500 Hz 至 2,000 Hz 之人耳高敏感頻段，而低頻結構振動、高頻胎面—路面微觀接觸噪音，雖在能量上可能非主導，卻常在主觀煩擾感中扮演關鍵角色。

在方法論層面，本章完整涵蓋時域分析、頻域分析與時頻分析三大類技術。時域分析透過統計參數與相關函數，揭示輪胎噪音的瞬態與週期性特徵；頻域分析則利用傅立葉轉換、倍頻程與窄頻譜，解析噪音能量分布與特徵頻率；時頻分析方法如短時傅立葉變換、小波分析與階次追蹤，進一步克服非平穩信號的限制，使噪音頻譜隨時間或轉速變化的行為得以精確描述。章節同時引入心理聲學指標，如響度、尖銳度、粗糙度與波動強度，說明頻譜特性如何轉化為人類對噪音品質與煩擾程度的主觀感知。

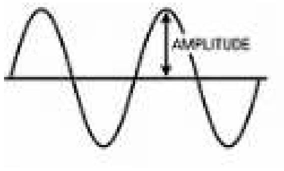
輪胎噪音頻譜分析構成現代輪胎聲學研究與工程實踐的核心技術基礎,其重要性不僅在於量化噪音水平的絕對數值,更在於揭示噪音的頻率組成、時間演變、空間分布與主觀感知特性,從而為噪音源識別、降噪設計、性能優化提供科學依據[1]。

輪胎滾動噪音本質上是一個複雜的寬頻隨機信號(broadband random signal),其能量分布橫跨 20 Hz 至 20,000 Hz 的可聽頻率範圍,峰值能量通常集中在 500 Hz 至 2,000 Hz 區間,這正是人耳最敏感的頻段,也是交通噪音對居民生活影響最顯著的頻率範圍[2][3]。與純音(pure tone)或週期性信號不同,輪胎噪音包含大量瞬態成分(transient components)、調製現象(modulation phenomena)與非平穩特性(non-stationary characteristics),這些特性源於輪胎胎面花紋的週期性排列、路面紋理的隨機性、車輪旋轉速度的變化、輪胎結構的共振行為等多重物理機制的耦合作用[4][5]。

傳統的聲學量測通常以 A 加權噪音值(A-weighted sound pressure level, LAeq 或 LA)作為噪音評估指標,這種單一數值指標雖然簡潔且與法規要求一致,但無法充分反映噪音的頻譜特性與時間變化規律,更難以建立噪音與主觀煩擾感(annoyance)之間的精確關聯[6]。大量心理聲學(psychoacoustics)研究證實,人類對聲音的感知不僅取決於總噪音值,更受頻譜形狀(spectral shape)、時間變化模式(temporal pattern)、音調性(tonality)、粗糙度(roughness)等複雜因素的影響[7][8]。兩個具有相同 A 加權噪音值的輪胎噪音,若頻譜分布不同,可能給聽者帶來截然不同的主觀感受:一個以低頻嗡嗡聲(low-frequency rumble)為主的噪音可能被感知為沉悶但可容忍,而另一個包含高頻尖銳成分(high-frequency whistling)的噪音則可能極度惹人煩躁,即便兩者的 dB(A)值相同[9][10]。因此,深入的頻譜分析不僅是學術研究的需要,更是實現真正以人為本的降噪設計、開發低煩擾度輪胎產品的技術前提。

輪胎噪音頻譜分析的方法論體系涵蓋時域分析(time domain analysis)、頻域分析(frequency domain analysis)、時頻分析(time-frequency analysis)三大範疇,各自具備獨特的適用場景與技術優勢[11]。時域分析直接處理原始聲壓信號的時間序列,透過統計參數如均方根值(root mean square, RMS)、峰值因數(crest factor)、峭度(kurtosis)等描述信號的整體特徵與瞬態特性,並可透過相關分析(correlation analysis)與卷積運算(convolution)研究噪音與激勵源的關係[12][13]。時域分析的優勢在於直觀性與計算效率,但對於複雜頻率結構的解析能力有限,難以分離不同

頻率成分的貢獻。頻域分析將時域信號轉換至頻率域,透過傅立葉變換(Fourier transform)、功率譜密度(power spectral density, PSD)、倍頻程分析(octave band analysis)等技術揭示信號的頻率組成與能量分布,這對於識別特徵頻率(characteristic frequencies)、共振峰(resonance peaks)、音調成分(tonal components)至關重要[14][15]。然而,標準頻域分析假設信號在整個分析窗長內保持平穩,這對於非平穩的輪胎噪音信號(如加速、減速、轉向過程)往往不成立,導致頻譜估計的時間解析度與頻率解析度之間存在固有衝突[16]。



時頻分析方法的發展正是為了克服傳統時域與頻域分析的局限,同時提供時間與頻率的二維聯合表示,揭示信號頻譜隨時間的演變規律[17]。短時傅立葉變換(Short-Time Fourier Transform, STFT)透過滑動視窗對信號進行局部頻譜分析,生成時頻譜圖(spectrogram),廣泛應用於輪胎噪音的瞬態事件檢測與動態過程監測[18][19]。小波分析(wavelet analysis)採用可變時頻解析度的小波基函數,在低頻段提供精細的頻率解析度、在高頻段提供精細的時間解析度,特別適合分析多尺度特徵並存的輪胎噪音信號[20][21]。階次追蹤(order tracking)技術將角域(angular domain)代替時域作為分析基準,消除轉速波動對頻譜分析的影響,精確提取與輪胎轉速同步的週期性成分,是旋轉機械噪音分析的標準工具[22][23]。這些先進分析技術的應用,使得研究人員能夠深入理解輪胎噪音的生成機理、傳播路徑、輻射特性,為新材料開發、花紋設計優化、結構降噪改進提供量化指導。

除物理聲學參數外,噪音品質(sound quality)評估體系的建立是輪胎噪音頻譜分析的重要延伸方向,旨在建立客觀聲學指標與主觀感知之間的定量關係[24][25]。傳統的噪音值指標基於能量加總原則,假設不同頻率的聲音對總體響度(loudness)的貢獻可簡單疊加,但這與人耳的實際感知機制相悖:人耳對不同頻率的敏感度存在顯著差異,低頻與極高頻的感知靈敏度遠低於中頻,即便物理聲壓相同[26][27]。國際標準 ISO 532 系列與 DIN 45631 定義的響度模型基於臨界頻帶理論(critical band theory)與掩蔽效應(masking effect),更準確地模擬人耳對複雜聲音的響度感知[28][29]。尖銳度(sharpness)量化聲音的高頻成分比例,反映"刺耳"或"尖銳"的主觀感受;粗糙度(roughness)描述 15-300 Hz 範圍內的幅度調製引起的"粗糙"或"顫動"感;波動強度(fluctuation strength)表徵低於 20 Hz 的緩慢幅度變化產生的"波動"知覺[30][31][32]。這些心理聲學指標的綜合應用,使得輪胎聲學工程師不僅能降低噪音的絕對水平,更能優化噪音的"品質",創造更舒適、更少煩擾的聲學環境。

輪胎噪音頻譜分析在產業實踐中的應用場景極為廣泛。在產品開發階段,透過對比分析不同花紋設計、胎面膠料配方、結構參數的原型輪胎的頻譜特性,工程師可快速識別降噪潛力最大的設計方向,縮短開發週期並降低試錯成本[33][34]。在型式認證測試中,雖然法規僅要求報告 A 加權總噪音值,但完整的頻譜數據為理解測試結果的物理成因、評估測試重複性與再現性、診斷異常測試結果提供寶貴信息[35]。在生產品質監控環節,透過對抽檢輪胎的噪音頻譜進行統計過程控制(statistical process control, SPC),可及早發現生產偏差、材料批次變化或設備故障導致的聲學性能劣化[36]。在售後服務與投訴處理中,頻譜分析可幫助判斷用戶反映的噪音異常是設計固有特性、正常磨損導致,還是製造缺陷或使用不當引起,為責任判定提供客觀依據[37]。此外,在道路交通噪音預測與管理領域,基於輪胎噪音頻譜數據的聲學模型(如 CNOSSOS-EU、Nord2000)能夠更準確地預測不同車型、不同路面條件下的交通噪音分布,指導城市規劃與降噪措施設計[38][39]。

本章將系統闡述輪胎噪音頻譜分析的理論基礎、方法技術與應用實踐。首先,7.1 節介紹聲學基礎理論,涵蓋聲壓與噪音值的定義與量測、頻率與波長的物理關係、A 加權及其他頻率加權函數的原理與應用,為後續技術討論奠定概念基礎。7.2 節論述時域分析方法,包括時域統計參數、瞬態特徵提取、相關函數分析等。7.3 節深入探討頻域分析技術,涵蓋快速傅立葉變換(FFT)的原理與實現、倍頻程與 1/3 倍頻程分析的標準與應用、窄頻譜分析的精細化技術。7.4 節介紹時頻分析方法,包括短時傅立葉變換、小波變換、階次追蹤的理論與實踐。7.5 節總結輪胎噪音的特徵頻率,分析胎面花紋節距激勵、空腔共鳴、結構振動等物理機制對應的頻率特徵。7.6 節論述噪音品質評估體系,涵蓋響度、尖銳度、粗糙度、波動強度等心理聲學參數的計算與應用。透過本章的學習,讀者將全面掌握輪胎噪音頻譜分析的理論方法與工程應用,具備從聲學數據中提取有價值信息、指導輪胎降噪設計的專業能力。

7.1 聲學基礎理論 (Acoustic Basic Theory)

聲學基礎理論為輪胎噪音頻譜分析提供必要的物理概念框架與數學工具,是理解複雜聲學現象、掌握量測技術、正確解讀分析結果的前提[40]。聲音(sound)本質上是媒質(通常為空氣)中壓力擾動的傳播過程,這種擾動以波動形式在空間中傳遞能量與信息,其物理特性由聲壓(sound pressure)、質點速度(particle velocity)、聲強(sound intensity)等場量描述[41][42]。在輪胎噪音研究中,聲壓是最常用的測量參量,因為麥克風(microphone)的工作原理基於聲壓驅動振膜產生電信號,且聲壓

量測技術成熟、設備普及、標準完善[43]。然而,聲壓的動態範圍極大:人耳可聽的最小聲壓約為 20 微帕(μPa,聽閾值 threshold of hearing),而引起疼痛的聲壓可達 200 帕(Pa,痛閾值 threshold of pain),兩者相差一千萬倍(10^7)[44]。為便於表示如此寬廣的範圍,聲學中普遍採用對數標度的噪音值(sound pressure level, SPL)概念,以分貝(decibel, dB)為單位,壓縮動態範圍並更貼近人耳的主觀響度感知規律[45][46]。

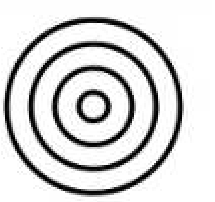
聲音的頻率(frequency)與波長(wavelength)是描述波動特性的基本參數,兩者透過聲速(speed of sound)建立關係,共同決定聲波的傳播行為、衍射特性、吸收特性[47]。輪胎噪音涵蓋從低頻轟鳴(low-frequency rumble)到高頻嘯叫(high-frequency whistle)的寬廣頻譜,不同頻段的噪音源機制、傳播路徑、衰減特性、主觀感受均存在顯著差異,因此頻率是噪音分析中最關鍵的分類維度[48][49]。低頻噪音(通常指 20-200 Hz)主要源於輪胎結構共振與整體變形,具有較長波長(1.7 米至 17 米 @20°C 空氣),易於繞射(diffraction)建築物與障礙物,穿透力強,衰減慢,是室內噪音與遠距離傳播的主要貢獻者,但人耳對極低頻的敏感度相對較低[50][51]。中頻噪音(200-2000 Hz)涵蓋輪胎噪音的主要能量區間,對應胎面花紋的基本激勵頻率、空氣泵吸噪音、胎肩振動等核心機制,波長適中(0.17 米至 1.7 米),是交通噪音法規管制的重點頻段,也是人耳最敏感的頻率範圍[52][53]。高頻噪音(2000-10000 Hz 及以上)主要源於胎面與路面的粗糙接觸、細小溝槽的空氣共鳴、結構的高階振動模態,波長較短(17 毫米至 170 毫米),方向性強,易被吸收,傳播距離有限,但即便能量不高也可能因尖銳刺耳而引起強烈煩擾[54][55]。

人耳對不同頻率聲音的敏感度存在顯著差異,這種頻率相關的感知特性被稱為等響曲線(equal-loudness contour)或 Fletcher-Munson 曲線,國際標準 ISO 226:2003 對其進行了精確定義[56][57]。在相同噪音值下,人耳對 1000-5000 Hz 範圍的聲音最為敏感,對低於 100 Hz 的低頻聲和高於 10000 Hz 的超高頻聲的敏感度則顯著降低[58]。為使物理測量結果更貼近主觀感受,聲學測量中引入頻率加權(frequency weighting)技術,其中 A 加權(A-weighting)應用最為廣泛,其頻率響應曲線模擬人耳在 40 方(phon)響度級下的頻率感知特性,對低頻與極高頻進行衰減,對中頻保持平坦[59][60]。A 加權噪音值(dBA 或 LA)已成為環境噪音法規、交通噪音標準、產品型式認證的標準評估指標,全球輪胎噪音法規如 ECE R117、ISO 13325 均採用 A 加權作為限值判定基準[61][62]。然而,A 加權並非萬能:其在低頻段的衰減過於

激進,無法充分反映低頻噪音對某些敏感人群(如兒童、老人)的影響;其基於 40 方響度級設計,對高聲級噪音的頻率感知特性描述不夠準確[63][64]。因此,在科研與工程實踐中,C 加權(C-weighting)、Z 加權(Z-weighting,即線性無加權)以及心理聲學模型如響度級(loudness level)等補充指標亦被廣泛採用[65][66]。

理解聲學基礎理論不僅是掌握測量技術的前提,更是正確解讀頻譜數據、避免分析誤區的關鍵。例如,直接比較不同頻率的噪音值數值而不考慮頻率加權,可能導致對噪音煩擾度的嚴重誤判;忽視聲波的波動特性(如駐波 standing wave、繞射、干涉 interference)可能使測點選擇不當,測得非代表性數據;混淆噪音值(對數量)與聲壓(線性量)的運算規則,可能在多源噪音疊加計算中犯下根本性錯誤[67][68]。這些基礎知識看似簡單,實則是嚴謹科學研究與工程實踐的基石,任何高級分析技術的應用都建立在對這些基本概念的準確理解之上。本節將深入闡述聲壓與噪音值、頻率與波長、頻率加權函數等核心概念的物理內涵、數學定義、測量技術與應用實踐,為後續章節的頻譜分析方法提供堅實的理論基礎。

7.1.1 聲壓與噪音值 (Sound Pressure and Sound Pressure Level)



聲壓(sound pressure)定義為聲波傳播過程中,介質中某點的瞬時壓力相對於靜壓(static pressure)的擾動量,通常記為 p(t),單位為帕斯卡(Pascal, Pa)[69]。在標準大氣壓環境下,靜壓約為 101,325 Pa,而聲波引起的壓力擾動通常遠小於此值:正常交談的聲壓約為 0.02 Pa,繁忙街道的交通噪音約為 0.2 Pa,噴氣發動機近場噪音可達 200 Pa[70][71]。聲壓是空間與時間的函數 p(x, y, z, t),在自由場(free field)條件下,平面波的聲壓遵循波動方程(wave equation):

$$\nabla^2 p - (1/c^2)\partial^2 p/\partial t^2 = 0$$

其中 c 為聲速,在 20°C 空氣中約為 343 m/s[72][73]。聲壓的測量依賴麥克風,其核心部件是壓力敏感膜片(diaphragm),聲壓作用於膜片產生機械位移,透過電容、壓電或動圈等轉換機制轉化為電信號[74][75]。精密測量麥克風(如 Brüel & Kjær Type 4189、GRAS 46AE 等)具有平坦的頻率響應(typically ±1 dB from 10 Hz to 20 kHz)、低本底噪音(typically <15 dBA)、寬動態範圍(typically 20-140 dB)、良好的長期穩定性,是聲學研究的標準工具[76][77]。

然而,直接使用聲壓值 p(t)進行噪音評估存在諸多不便。首先,聲壓是時變量,對於隨機噪音信號,瞬時聲壓在正負之間快速波動,其時間平均值趨近於零,無法反映噪音的實際強度[78]。其次,聲壓的數值範圍極大(六個數量級以上),使用線性標度

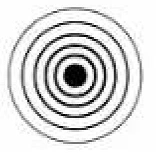
難以在同一圖表中清晰呈現不同強度的聲音[79]。第三,人耳對聲音強度的感知遵循近似對數規律(Weber-Fechner law),即感知響度的增量與刺激強度的相對變化成正比,而非絕對變化[80][81]。基於這些考量,聲學中引入噪音值(sound pressure level, SPL)概念,定義為瞬時聲壓 $p(t)$的均方根值(root mean square, RMS)相對於參考聲壓 p_0的對數比值:

$$Lp = 20 \log_{10}(prms / p_0)\ \mathrm{dB}$$

其中 $prms = \sqrt{(\langle p^2(t)\rangle T)}$為聲壓的均方根值,$\langle \cdot \rangle T$ 表示在測量時間 T 內的時間平均,$p_0 = 20\ \mu\mathrm{Pa}$ 為參考聲壓,對應於人耳在 1000 Hz 頻率下的聽閾值[82][83]。這一定義具有多重優點:使用對數標度將寬廣的聲壓範圍壓縮至 0-140 dB 的緊湊區間;均方根運算確保結果為正且反映信號的能量含量;參考值的選取使 0 dB 對應聽閾,具有明確的生理意義[84]。

噪音值的分貝標度具有重要的物理與心理學意義。在物理層面,噪音值每增加 6 dB,對應聲壓幅值加倍,聲能量(正比於聲壓平方)增加四倍[85]。在心理層面,研究表明噪音值每增加 10 dB,人耳感知的響度約加倍,這一經驗規律為噪音控制目標的設定提供直觀參考[86][87]。然而,需注意分貝是對數單位而非線性單位,因此噪音值的算術運算需遵循特殊規則。兩個不相關聲源的噪音值疊加(incoherent superposition)遵循能量相加原則:

$$Ltotal = 10 \log_{10}(10^{(L1/10)} + 10^{(L2/10)})\ \mathrm{dB}$$



例如,兩個噪音值均為 70 dB 的不相關聲源同時作用,總噪音值為 73 dB(而非 140 dB 或 70+70=140 dB)[88]。這一非線性特性意味著,當主導噪音源與次要噪音源的噪音值差距超過 10 dB 時,次要噪音源對總噪音值的貢獻可忽略不計(增量<0.4 dB),這是噪音源識別與優先級排序的重要依據[89][90]。

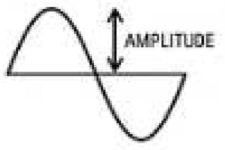
在輪胎噪音測量實踐中,噪音值的計算通常採用時間積分(time integration)方法,以適應非平穩信號的特點[91]。等效連續噪音值(equivalent continuous sound pressure level, Leq)定義為在測量時間 T 內,與實際時變聲壓具有相同聲能量的恆定聲壓對應的噪音值:

$$Leq = 10 \log_{10}[(1/T)\int_0^T (p^2(t)/p_0^2)\, dt]\ \mathrm{dB}$$

對於數字化測量系統,這一積分透過離散采樣近似計算:

$$Leq \approx 10 \log_{10}[(1/N)\Sigma_{i=1}^{N} 10^{(Li/10)}]\ \mathrm{dB}$$

其中 Li 為第 i 個采樣時刻的瞬時噪音值,N 為總採樣點數[92][93]。ISO 13325:2019 規定的輪胎滑行噪音測量中,要求記錄車輛透過測量區域過程中的最大 A 加權噪音值(LAmax)以及等效噪音值(LAeq),並對多次測試結果進行算術平均以降低隨機誤差[94][95]。



噪音值的頻率相關性是理解輪胎噪音頻譜分析的關鍵。總噪音值(overall sound pressure level)是將所有頻率成分的能量疊加後計算得到的單一數值,而頻帶噪音值(band sound pressure level)則針對特定頻率範圍進行計算,揭示能量在頻譜上的分布[96]。若將信號分解為 N 個頻帶,各頻帶的噪音值為 L_1, L_2, …, L_n,則總噪音值為:

$Ltotal = 10\ \log_{10}(\Sigma_{i=1}^{N}\ 10{}^\wedge(Li/10))$ dB

這一關係表明,即便某一頻帶的噪音值顯著高於其他頻帶,其對總噪音值的貢獻仍受限於對數運算的壓縮效應,因此頻譜峰值的識別與抑制在降噪設計中至關重要[97][98]。例如,一個輪胎噪音信號在 800 Hz 處存在 10 dB 的音調峰(tonal peak),即便其他頻率的噪音保持不變,消除這一峰值可使總噪音值降低約 3 dB,相當於噪音能量減半,這足以產生明顯的主觀改善[99][100]。

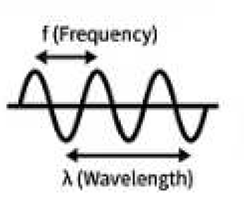
噪音值測量的準確性受多種因素影響,包括麥克風的校準狀態、環境條件(溫度、濕度、氣壓)、背景噪音、風噪、反射與繞射效應等[101][102]。國際標準 IEC 61672-1:2013 規定了噪音計(sound level meter)的性能要求與校準程序,將噪音計分為 Class 1(精密級,不確定度±0.7 dB)與 Class 2(通用級,不確定度±1.0 dB)兩類[103]。輪胎噪音測量通常要求使用 Class 1 噪音計,並在每次測量前後使用聲級校準器(sound calibrator,如 94 dB @ 1 kHz 或 114 dB @ 1 kHz)進行現場校準,確保系統準確性[104][105]。測量不確定度分析(uncertainty analysis)是高品質聲學研究的必備環節,ISO/IEC Guide 98-3:2008 提供了系統化的不確定度評估框架,考慮麥克風靈敏度、環境影響、信號處理、統計抽樣等各環節的貢獻,最終報告擴展不確定度(expanded uncertainty, k=2 對應 95%置信水平)[106][107]。

在輪胎噪音測量的具體場景中,噪音值的空間變化特性亦需重視。ISO 10844:2021 規定的測試路面要求在測量區域內聲學特性均勻,任何測點的噪音值變化不超過 1 dB,以確保測試結果的空間代表性[108]。然而,實際測量環境中,地面反射(ground reflection)、車輛結構的聲散射(scattering)、輪胎近場的聲場複雜性等因素,使得噪音值呈現顯著的空間梯度(spatial gradient)[109][110]。ISO 13325:2019 規定麥克風位置為距離車輛路徑中心線 7.5 米、高度 1.2 米,這一幾何配置在自由場條件下可

獲得穩定的測量結果,但對於近場測量(如車內噪音、輪拱噪音),需使用聲強測量(sound intensity measurement)或近場全息(nearfield acoustic holography, NAH)等更先進技術以準確定位噪音源[111][112][113]。

噪音值概念的推廣與變體在不同應用場景中發揮作用。瞬時噪音值(instantaneous sound pressure level)描述信號的時變特性,適用於瞬態事件(如輪胎撞擊路面坑洞)的分析[114]。峰值噪音值(peak sound pressure level, Lpeak)記錄測量期間的最大瞬時聲壓,用於評估衝擊性噪音的危害[115]。日夜等效噪音值(day-night average sound level, Ldn 或 DNL)將夜間噪音加權 10 dB 以反映夜間噪音的更高煩擾度,廣泛用於環境噪音評估[116][117]。百分位噪音值(percentile sound pressure level, LN)如 L10、L50、L90,表示在測量時間內有 N%的時間噪音值超過該值,用於描述噪音的統計分布特性[118][119]。這些衍生指標豐富了噪音值的應用場景,使其能夠適應從實驗室受控環境到複雜交通場景的多樣化需求。

7.1.2 頻率與波長 (Frequency and Wavelength)



頻率(frequency)與波長(wavelength)是描述波動現象的兩個基本物理量,兩者透過波速(wave speed)建立不可分割的聯繫,共同決定聲波的傳播行為與相互作用特性[120]。頻率 f 定義為單位時間內波動的完整週期數,單位為赫茲(Hertz, Hz),即每秒週期數(cycles per second)[121]。波長 λ 定義為波動在空間中一個完整週期對應的距離,單位為米(meter, m)[122]。聲速 c 為波動傳播速度,在給定介質與環境條件下,三者滿足基本關係:

$c = f \times \lambda$

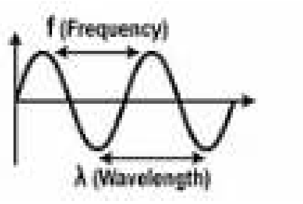
在 20°C、1 個標準大氣壓的乾燥空氣中,聲速約為 343 m/s,因此 1000 Hz 聲波的波長為 0.343 m(34.3 cm),100 Hz 聲波的波長為 3.43 m,10,000 Hz 聲波的波長僅為 3.43 cm[123][124]。這一關係表明,頻率與波長成反比:高頻聲波具有短波長,低頻聲波具有長波長,這種差異對聲波的傳播、衍射、反射、吸收等行為產生根本性影響。

聲速 c 本身是溫度、壓力、濕度、介質組成的函數,在空氣中主要受溫度影響,近似關係為:

$c \approx 331.3 + 0.606 \times T$ m/s

其中 T 為攝氏溫度[125][126]。這意味著溫度每升高 1°C,聲速增加約 0.6 m/s,相應地波長增加約 0.2%。在精密聲學測量中,需記錄環境溫度並對聲速進行修正,特別

是在波長敏感的應用(如聲學全息、相控陣測量)中,溫度引起的聲速變化可能導致相位誤差累積,影響測量精度[127][128]。濕度對聲速的影響相對較小但不可忽略:在相同溫度下,飽和濕空氣的聲速比乾燥空氣高約 0.3%,因為水蒸氣的分子量(18 g/mol)低於乾燥空氣的平均分子量(約 29 g/mol),導致混合氣體密度降低、聲速提高[129][130]。大氣壓力對聲速的直接影響很小,因為聲速主要由介質的密度與彈性模量比值決定,而溫度恆定時壓力變化對兩者影響相互抵消[131]。



頻率是聲學分析的核心分類維度,人耳可聽頻率範圍通常定義為 20 Hz 至 20,000 Hz(20 kHz),這一範圍隨年齡增長而縮窄,中老年人的高頻聽力通常衰減至 10-12 kHz[132][133]。次聲(infrasound)指頻率低於 20 Hz 的聲波,雖然人耳難以直接感知,但可引起身體不適與心理煩擾,在某些路面條件(如粗糙路面、波浪狀路面)下輪胎與車身耦合振動可產生顯著的次聲成分[134][135]。超聲(ultrasound)指頻率高於 20 kHz 的聲波,人耳無法感知但可被專用設備檢測,在輪胎缺陷檢測、材料特性評估中有應用,但對交通噪音煩擾無直接貢獻[136][137]。輪胎滾動噪音的主要能量分布在 100 Hz 至 5,000 Hz 頻率範圍,其中 500 Hz 至 2,000 Hz 為峰值能量區間,這正好對應人耳最敏感的頻段,也是法規管制與降噪設計的重點[138][139]。

波長對聲波傳播行為的影響表現在多個方面。首先,衍射(diffraction)能力與波長正相關:當聲波遇到障礙物或孔徑時,若障礙物尺寸遠小於波長,聲波可輕易繞過障礙物繼續傳播;若障礙物尺寸與波長相當或更大,則產生顯著的聲影(acoustic shadow)[140][141]。低頻聲波(長波長)具有強大的繞射能力,可穿透建築物縫隙、繞過隔音屏障,這是低頻噪音難以控制的根本原因;高頻聲波(短波長)則易被障礙物阻擋,方向性強,適合透過吸音、隔聲措施控制[142][143]。其次,反射(reflection)特性與波長相關:當聲波入射到界面(如地面、牆壁)時,若界面尺寸遠大於波長,反射遵循鏡面反射定律;若界面粗糙度與波長相當,則產生散射(scattering)反射,聲能量向多個方向分散[144][145]。路面的聲學特性(吸音係數、散射特性)對輪胎噪音的影響與頻率密切相關:多孔瀝青路面(porous asphalt)對高頻噪音(>1000 Hz)的吸音效果顯著,但對低頻噪音(< 500 Hz)的吸音能力有限[146][147]。

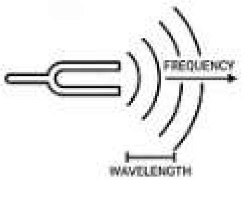
駐波(standing wave)是波長相關的重要聲學現象,當入射波與反射波在封閉或半封閉空間中相互疊加,若空間尺寸為半波長的整數倍,形成穩定的空間壓力分布,稱

為駐波[148][149]。在輪胎空腔共鳴(cavity resonance)現象中,輪胎內部空氣腔在特定頻率下形成駐波,導致噪音顯著放大,該共鳴頻率 f_0可近似估算為:

$$f_0 \approx c / (2\pi r)$$

其中 r 為輪胎半徑[150][151]。對於典型乘用車輪胎($r \approx 0.3$ m),空腔共鳴頻率約為 200-240 Hz,這一頻段的峰值可使總噪音增加 3-5 dB,是輪胎降噪設計的重要考量[152][153]。駐波效應在測量環境中亦需防範:若測量場地存在平行反射面(如牆壁、地面),可能在特定頻率形成駐波,導致測量結果的空間不均勻性與頻率相關偏差,ISO 3745:2012 規定的半消聲室(semi-anechoic chamber)設計要求地面反射但其他方向吸音,模擬戶外半空間測量條件並消除駐波干擾[154][155]。

波長與聲源尺寸的關係決定聲輻射的方向性(directivity)[156]。當聲源尺寸遠小於波長時,聲源可視為點源(point source)或偶極子源(dipole source),聲輻射呈現全向或雙向特性,聲壓隨距離增加而衰減(自由場中點源噪音值隨距離加倍下降 6 dB)[157][158]。當聲源尺寸與波長相當或更大時,聲源呈現複雜的方向性,某些方向的輻射增強、某些方向減弱,這種方向性隨頻率變化[159][160]。輪胎接地面(contact patch)的典型尺寸為 10-20 cm,對於低頻噪音(波長>1 m)可視為緊湊源,對於高頻噪音(波長<10 cm)則呈現明顯的方向性輻射,導致不同方位的測點接收到的高頻噪音水平差異顯著[161][162]。這一特性在麥克風佈置設計、噪音源識別、陣列測量中具有重要意義。



頻率的心理聲學意義超越其物理定義,與音高(pitch)感知密切相關[163]。純音的音高與頻率成正比,但複雜聲音的音高感知受頻譜形狀、調製特性、持續時間等多重因素影響,可能出現缺失基頻(missing fundamental)現象,即基頻成分缺失但聽者仍感知到對應基頻的音高,這是由於諧波結構(harmonic structure)在聽覺系統中的綜合處理[164][165]。輪胎噪音雖然以寬頻隨機噪音為主,但在某些條件下(如特定速度、特定路面、花紋設計不當)可能呈現準週期性,產生可辨別的音調(tonal)成分,這種音調性噪音即便能量不高,也可能因其突出性(prominence)而引起強烈煩擾[166][167]。音調性評估指標如音調可聽度比(Tone-to-Noise Ratio, TNR)、突出度(Prominence Ratio)等,用於量化頻譜中音調成分的顯著程度,指導降噪優先級的確定[168][169]。

頻率範圍的劃分在聲學分析中具有實用意義。低頻(low frequency)通常指 20-200 Hz 或 20-500 Hz,對應輪胎結構振動、空腔共鳴的主導頻段,具有穿透性強、衰減

慢、主觀煩擾度高(特別在室內環境)的特點[170][171]。中頻(mid frequency)通常指 200-2000 Hz 或 500-2000 Hz,涵蓋輪胎噪音的主要能量區間與人耳敏感頻段,是噪音法規與降噪設計的重點[172][173]。高頻(high frequency)通常指 2000-10000 Hz 或以上,對應胎面與路面的細節接觸、微觀粗糙度激勵、高階振動模態,雖能量較低但可能引起尖銳感與煩擾[174][175]。這種頻段劃分並無嚴格標準,不同文獻與標準可能採用不同邊界,但基本邏輯一致:根據物理機制、傳播特性、主觀感受的差異,將寬廣頻譜劃分為若干特徵區間,便於分析與討論。

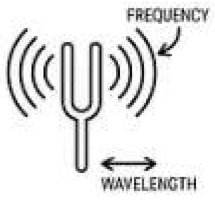
倍頻程(octave)與 1/3 倍頻程(one-third octave)是標準化的頻段劃分體系,廣泛用於噪音分析與報告[176]。倍頻程帶定義為上限頻率是下限頻率兩倍的頻帶,中心頻率 fc 與帶寬 Δf 的關係為:

$$fc = \sqrt{(f1 \times f2)}$$

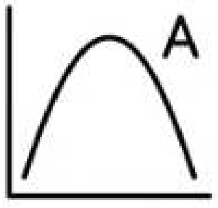
$$\Delta f = f2 - f1 = fc \times (2^{(1/2)} - 1) \approx 0.707 \times fc$$

其中 f1、f2 為下限、上限頻率[177][178]。標準倍頻程中心頻率序列為 31.5, 63, 125, 250, 500, 1000, 2000, 4000, 8000, 16000 Hz(ISO 266:1997),涵蓋人耳可聽範圍[179]。1/3 倍頻程帶將每個倍頻程細分為三個子頻帶,提供更精細的頻譜解析度,中心頻率序列為 $10^{(n/10)}$ Hz(n 為整數),如 100, 125, 160, 200, 250, 315, 400, 500, 630, 800, 1000 Hz 等[180][181]。倍頻程與 1/3 倍頻程分析在噪音評估、頻譜比較、吸音材料測試、聲學設計中被廣泛採用,因為其頻帶寬度隨中心頻率成比例增加,符合人耳的臨界頻帶特性,且計算簡便、結果緊湊[182][183]。

頻率解析度(frequency resolution)是頻譜分析的關鍵技術參數,定義為能夠分辨的最小頻率間隔[184]。在離散傅立葉變換(Discrete Fourier Transform, DFT)中,頻率解析度 $\Delta f = fs/N$,其中 fs 為採樣率,N 為採樣點數,因此提高頻率解析度需增加採樣點數或降低採樣率[185][186]。然而,採樣點數增加意味著分析窗長延長,對於非平穩信號可能導致時間解析度下降,這是時頻分析中的基本矛盾[187]。窄頻分析(narrowband analysis)採用較長的分析窗(如數秒)獲得精細的頻率解析度(如 1 Hz 或更小),適合識別離散音調成分;寬頻分析(broadband analysis)如倍頻程分析採用較短窗長(如數十毫秒),犧牲頻率解析度以獲得較好的時間解析度,適合分析快速變化的瞬態事件[188][189]。在實際應用中,分析參數的選擇需根據信號特性與分析目標綜合權衡,這是聲學工程師專業判斷力的表現。



7.1.3 A 加權與其他加權 (A-weighting and Other Weightings)



頻率加權(frequency weighting)是聲學測量中將物理噪音值轉換為反映主觀感知的加權噪音值的技術手段,其核心思想是對不同頻率的聲壓施加不同的權重因子,模擬人耳對不同頻率聲音的敏感度差異[190][191]。最廣泛使用的頻率加權是 A 加權(A-weighting),其頻率響應曲線基於 40 方(phon)等響曲線的倒數,對低頻與極高頻進行顯著衰減,對中頻(1000-5000 Hz)保持相對平坦[192][193]。A 加權噪音值記為 dB(A)或 LA,已成為環境噪音評估、交通噪音標準、職業噪音暴露限值、產品型式認證的國際通用指標[194][195]。其廣泛採納源於多重優勢:實現簡單(僅需頻域或模擬濾波器加權)、物理意義明確(近似反映人耳頻率感知)、與大量既有研究與法規相容、測量設備普及(幾乎所有商業噪音計均內置 A 加權)[196][197]。

A 加權的數學定義由國際標準 IEC 61672-1:2013 給出,其頻率響應 RA(f)為:

$RA(f) = (12194^2 \times f^4) / [(f^2+20.6^2)(f^2+12194^2)\sqrt{((f^2+107.7^2)(f^2+737.9^2))}]$

以分貝表示為:

$WA(f) = 20 \log_{10}(RA(f)) + 2.0 \text{ dB}$

其中 2.0 dB 為歸一化常數,使 1000 Hz 處的加權值為 0 dB[198][199]。這一函數在低頻段迅速下降:100 Hz 處衰減約-19 dB,50 Hz 處衰減約-31 dB,意味著相同噪音值的低頻聲音在 A 加權後其貢獻大幅降低;在高頻段,10 kHz 處衰減約-2 dB,20 kHz 處衰減約-9 dB[200][201]。這種頻率響應特性使得 A 加權噪音值能夠較好地預測中等聲級環境噪音的主觀響度與煩擾度,特別是在 40-80 dB(A)範圍內[202][203]。大量流行病學研究證實,A 加權噪音值與交通噪音暴露引起的睡眠干擾、煩擾反應、心血管疾病風險等健康效應之間存在顯著劑量-反應關係,支持其作為公共衛生管制指標的有效性[204][205]。

然而,A 加權並非完美無缺,其局限性在理論與實踐中均有表現[206]。首先,A 加權基於 40 方等響曲線設計,這一響度級對應於安靜環境中的適度聲音,但對於高聲級噪音(如 90 dB 以上),人耳的頻率感知特性發生變化,低頻與高頻的相對敏感度提高,A 加權的頻率響應不再準確反映實際感知[207][208]。研究表明,在高聲級環境下,C 加權噪音值(基於 100 方等響曲線,對低頻衰減遠小於 A 加權)與主觀煩擾度的相關性優於 A 加權[209][210]。其次,A 加權對低頻聲音的衰減過於激進,無法充分反映低頻噪音(特別是 20-200 Hz)對某些敏感人群的影響[211]。低頻噪音雖

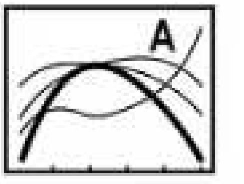
然 A 加權噪音值不高,但可引起建築物振動、共鳴,在室內環境中引起持續的身體不適與心理煩擾,特別是對兒童、老人、神經敏感者[212][213]。近年來,歐洲多國在評估低頻噪音暴露時,開始採用 G 加權(G-weighting)或直接使用線性噪音值(Z 加權),以更準確地評估低頻噪音的健康影響[214][215]。

C 加權(C-weighting)是 A 加權的重要補充,其頻率響應曲線相對平坦,僅在極低頻與極高頻進行適度衰減,基於 100 方等響曲線設計[216][217]。C 加權噪音值記為 dB©或 LC,在高聲級噪音環境(如工業現場、機場、軍事訓練場)中被廣泛用於評估噪音暴露與聽力損傷風險[218][219]。C 加權的數學定義為:

$RC(f) = (12194^2 \times f^2) / [(f^2+20.6^2)(f^2+12194^2)]$

$WC(f) = 20 \log_{10}(RC(f)) + 0.06 \text{ dB}$

其在 100 Hz 處衰減約-3 dB,50 Hz 處衰減約-9 dB,遠小於 A 加權的衰減幅度[220][221]。LC - LA(C 加權噪音值與 A 加權噪音值的差值)被用作評估噪音低頻成分豐富程度的簡便指標:差值越大,表明低頻成分占比越高[222][223]。對於典型的輪胎滾動噪音,LC - LA 通常在 5-10 dB 範圍,反映其中低頻成分佔據相當比例[224]。在某些輪胎噪音研究中,同時報告 LA 與 LC 可提供更全面的頻譜特性信息,有助於理解不同降噪措施對不同頻段的效果差異[225][226]。



Z 加權(Z-weighting,原稱線性或無加權,linear or flat weighting)不對任何頻率進行衰減或增強,頻率響應在整個測量範圍內保持 0 dB(±1.5 dB 容差)[227][228]。Z 加權噪音值記為 dB(Z)或 LZ,提供未經頻率修正的"原始"噪音值信息,適用於需要完整保留頻譜信息的分析場景,如詳細的頻譜分析、聲源識別、物理模型驗證[229][230]。在研究輪胎噪音的物理生成機制時,Z 加權測量可避免加權函數對原始信號的干擾,便於與數值模擬(如有限元分析、邊界元分析)結果進行直接對比[231][232]。然而,Z 加權噪音值與主觀感知的關聯性弱於 A 加權與 C 加權,因此在噪音評估與法規合規中應用有限[233]。

除 A、C、Z 加權外,聲學測量中還存在若干專用加權函數,針對特定應用場景優化[234]。B 加權(B-weighting)基於 70 方等響曲線,介於 A 加權與 C 加權之間,曾在部分標準中使用但現已較少見[235]。D 加權(D-weighting)專為航空噪音設計,對 2000-10000 Hz 的高頻噪音敏感,反映飛機噪音的主觀煩擾特性[236][237]。ITU-R 468 加權用於廣播與音頻設備的噪音測量,對中高頻(1-10 kHz)特別敏感,反映音頻工程中的感知優先級[238]。U 加權(U-weighting)用於超聲設備的噪音評估,在 20-

100 kHz 頻率範圍定義[239]。G 加權(G-weighting)專為低頻噪音(< 200 Hz)設計,頻率響應在 10-30 Hz 達到峰值,對更低或更高頻率衰減,用於評估低頻噪音對人體的振動感知與煩擾[240][241]。這些專用加權函數在輪胎噪音研究中應用相對有限,但在特定場景(如低頻噪音投訴、航空輪胎噪音、室內噪音傳遞)中可能提供更準確的評估。

頻率加權的實現方式有模擬濾波與數字濾波兩種[242]。傳統噪音計採用模擬電路實現加權濾波器,利用電阻-電容網絡構造所需的頻率響應[243]。現代數字噪音計與聲學分析軟件則採用數字濾波器,透過 IIR(Infinite Impulse Response)或 FIR(Finite Impulse Response)濾波器設計實現精確的加權函數[244][245]。IEC 61672-1:2013 規定了 A、C、Z 加權濾波器的頻率響應容差、相位響應、時間加權特性等技術要求,確保不同製造商設備的測量一致性[246][247]。時間加權(time weighting)與頻率加權相獨立,定義了噪音值隨時間變化的響應速度:快速(Fast, 125 ms 時間常數)適用於快速變化的噪音,慢速(Slow, 1000 ms 時間常數)適用於相對穩定的噪音,脈衝(Impulse, 35 ms 上升/1500 ms 衰減)適用於衝擊性噪音[248][249]。輪胎噪音測量通常採用 Fast 時間加權,以捕捉車輛透過過程中的動態變化[250]。

在頻譜分析中,頻率加權可在兩個階段施加:時域加權與頻域加權[251]。時域加權是將加權濾波器直接應用於時域信號,然後計算加權後信號的噪音值;頻域加權是先進行頻譜分析獲得各頻帶的噪音值,再對各頻帶噪音值施加對應頻率的加權因子,最後疊加得到加權總噪音值[252][253]。兩種方法在理論上等效,但在實際應用中可能因濾波器設計、窗函數選擇、數值精度等因素導致微小差異[254]。頻域加權的優勢在於可同時獲得未加權頻譜與加權噪音值,便於詳細分析;時域加權的優勢在於計算效率高,適合實時測量與大數據處理[255][256]。

頻率加權在輪胎噪音法規中的應用是其權威性的最直接表現[257]。ECE R117、ISO 13325、AIS 142 等全球主要輪胎噪音標準均採用 A 加權噪音值作為限值判定指標[258][259]。測試要求記錄車輛滑行透過測量區域過程中的最大 A 加權噪音值(LAmax),並對多次測試結果的算術平均值與限值比較[260][261]。這種基於 A 加權的法規體系確保了全球輪胎產業的技術標準統一,降低了跨國貿易的技術壁壘,促進了環境保護標準的全球協調[262][263]。然而,單一 A 加權指標的局限性也引發持續討論:部分研究者主張補充低頻噪音評估指標(如 LC、LG 或特定頻

段的線性噪音值),以更全面地評估輪胎噪音的環境與健康影響;部分輪胎製造商則擔憂多指標要求增加測試複雜度與成本[264][265]。未來法規演進可能在保持 A 加權作為主要指標的同時,引入補充性頻譜指標,實現更精細化的噪音管制[266][267]。

頻率加權的心理聲學基礎是臨界頻帶理論(critical band theory)與掩蔽效應(masking effect)[268]。人耳的聽覺系統並非對每個頻率獨立處理,而是將聲音劃分為一系列臨界頻帶(約 24 個覆蓋整個可聽頻率範圍),每個臨界頻帶內的聲音成分相互作用,頻帶間相對獨立[269][270]。臨界頻帶的寬度在低頻段約為 100 Hz(如中心頻率 250 Hz 的臨界帶寬約 100 Hz),在高頻段隨頻率增加而增寬(如中心頻率 5000 Hz 的臨界帶寬約 900 Hz),這種變寬特性與倍頻程、1/3 倍頻程的頻帶劃分邏輯一致[271][272]。

掩蔽效應指強聲掩蔽弱聲的現象:當兩個頻率接近的聲音同時存在,較強的聲音會使較弱的聲音變得不可聽,掩蔽量取決於兩者的頻率差、強度差、時間關係[273][274]。A 加權雖不直接基於臨界頻帶理論設計,但其頻率響應在一定程度上反映了人耳對不同頻率聲音的整體敏感度差異,這種差異的生理基礎正是內耳耳蝸(cochlea)的頻率選擇性與神經編碼機制[275][276]。

響度級(loudness level)的概念為理解頻率加權提供更深層次的心理聲學視角[277]。響度級定義為聽起來與待評估聲音同樣響的 1000 Hz 純音的噪音值,單位為方(phon)[278]。根據定義,任何頻率的聲音,只要其響度級為 40 phon,則其主觀響度與 40 dB SPL 的 1000 Hz 純音相同[279]。等響曲線(equal-loudness contour)描繪不同頻率下達到相同響度級所需的噪音值,ISO 226:2003 給出了從 20 phon 到 90 phon 的標準等響曲線[280][281]。A 加權的頻率響應近似為 40 phon 等響曲線的倒數,這意味著對不同頻率的純音施加 A 加權後,若得到相同的 dB(A)值,則這些純音具有近似相同的主觀響度(在 40 phon 附近)[282]。

然而,這一近似在復雜聲音、寬頻噪音、含音調成分的聲音中可能失效,因為響度感知涉及頻譜整合、時間整合、調製檢測等多重心理聲學過程,簡單的頻率加權無法完全模擬[283][284]。更精確的響度計算需採用 Zwicker 響度模型(ISO 532-1)或 Moore-Glasberg 響度模型(ISO 532-2),這些模型基於臨界頻帶、興奮模式(excitation pattern)、比響度(specific loudness)等概念,能夠更準確地預測複雜聲音的主觀響度[285][286]。

7.2 時域分析 (Time Domain Analysis)

時域分析(time domain analysis)是聲學信號處理的最直接方法,透過觀察與量化聲壓信號隨時間的變化規律,提取信號的統計特徵、瞬態特性、週期性與相關性,為理解噪音的時間結構與動態演變提供基礎信息[287][288]。時域分析的優勢在於直觀性強、計算效率高、對數據預處理要求低,且不涉及頻域變換可能引入的頻譜泄漏(spectral leakage)、柵欄效應(picket-fence effect)等人為失真[289][290]。在輪胎噪音研究中,時域分析廣泛應用於瞬態事件識別(如輪胎撞擊路面異物、花紋塊進出接地面的衝擊)、噪音突變檢測(如速度變化、路面過渡段)、週期性評估(如花紋節距激勵的重複模式)、多通道信號相關性分析(如不同測點噪音的時間延遲與相干性)[291][292][293]。

時域波形(time waveform)是時域分析的基本對象,展示聲壓 p(t)隨時間 t 的完整變化曲線[294]。觀察時域波形可直觀判斷信號的基本特性:穩態信號呈現近似恆定的幅值包絡(amplitude envelope),瞬態信號表現為脈衝狀或突變,週期信號顯示規律的重複模式,隨機信號則無明顯規律[295][296]。然而,僅靠視覺觀察難以精確量化信號特徵,需計算統計參數(statistical parameters)對信號進行數值描述[297]。均方根聲壓(root mean square sound pressure, prms)是最常用的時域統計量,定義為:

$$prms = \sqrt{[(1/T)\int_0^T p^2(t)\, dt]}$$

或在離散采樣情況下:

$$prms = \sqrt{[(1/N)\Sigma_{i=1}^{N} p_i^2]}$$

其物理意義為信號的有效值(effective value),反映信號攜帶的聲能量[298][299]。

峰值聲壓(peak sound pressure, ppeak)定義為測量時間內聲壓的最大絕對值:

$$ppeak = \max|p(t)|$$

峰值因數(crest factor, CF)定義為峰值聲壓與均方根聲壓的比值:

$$CF = ppeak / prms$$

峰值因數反映信號的動態特性與衝擊性:正弦波的峰值因數為$\sqrt{2} \approx 1.414$,高斯白噪音的峰值因數通常在 3-4 之間,含強烈瞬態成分的信號峰值因數可達 10 以上[300][301][302]。輪胎滾動噪音的峰值因數通常在 4-6 範圍,表明其具有一定的衝擊性特徵,這與胎面花紋塊的週期性撞擊、輪胎結構的瞬態振動相關[303][304]。

峭度(kurtosis)是描述信號幅值概率分布形態的高階統計量,定義為四階中心矩與二階中心矩平方的比值:

$$Kurtosis = E[(p - \mu)^4] / (E[(p - \mu)^2])^2$$

其中 μ 為均值,$E[\cdot]$表示期望值(時間平均)[305][306]。高斯分布的峭度為 3,峭度大於 3 的信號(leptokurtic)具有更尖銳的峰值與更厚的尾部,表明信號含較多極值事件;峭度小於 3 的信號(platykurtic)分布較為平坦[307][308]。工程中常採用超峭度(excess kurtosis = kurtosis - 3)使高斯分布的參考值為 0[309]。峭度對信號中的瞬態衝擊極為敏感,即便瞬態成分能量佔比很小,也可能顯著提升峭度值,因此峭度被廣泛用於旋轉機械的故障診斷與瞬態事件檢測[310][311]。在輪胎噪音分析中,峭度可用於評估花紋設計對衝擊性噪音的影響:峭度較低的輪胎噪音信號表明花紋激勵較為均勻,峭度較高則可能存在某些花紋塊產生的異常強烈衝擊[312][313]。

偏度(skewness)描述信號幅值分布的非對稱性,定義為三階中心矩與標準差立方的比值:

$$\text{Skewness} = E[(p - \mu)^3] / (E[(p - \mu)^2])^{(3/2)}$$

對稱分布(如高斯分布)的偏度為 0,正偏度表明分布向正值方向拖尾,負偏度表明向負值方向拖尾[314][315]。聲壓信號通常接近對稱分布(偏度接近 0),但在某些特殊條件下(如非線性聲傳播、聲源的非對稱振動)可能呈現偏斜,偏度分析可揭示這些非線性特性[316][317]。

時域統計參數的計算需考慮分析窗長(analysis window length)的選擇[318]。窗長過短,統計量估計的隨機誤差大,特別是對於隨機信號與低頻成分;窗長過長,對非平穩信號的時間解析度不足,無法捕捉動態變化[319][320]。一般原則是窗長應包含足夠多的信號週期或統計樣本,通常建議至少 10 倍於信號的最低頻率成分對應的週期[321]。對於輪胎滾動噪音(主要能量在 100-5000 Hz),典型的分析窗長為數十毫秒至數百毫秒,既保證統計穩定性又能捕捉車輛透過過程中的動態變化[322][323]。

自相關函數(autocorrelation function, ACF)描述信號與其自身延遲版本的相似程度,定義為:

$$R(\tau) = (1/T)\int_0^T p(t)p(t+\tau)\, dt$$

或歸一化形式:

$$\rho(\tau) = R(\tau) / R(0)$$

其中 τ 為時間延遲(time lag)[324][325]。自相關函數在 $\tau=0$ 處達到最大值(等於信號的均方值),隨 τ 增加通常逐漸衰減,衰減速度反映信號的相關時間尺度[326][327]。對於完全隨機的白噪音,自相關函數在 $\tau\neq0$ 處迅速趨於零;對於含週期

性成分的信號,自相關函數在對應週期的整數倍處出現峰值[328][329]。輪胎噪音的自相關分析可揭示花紋節距激勵的週期性:若花紋節距設計較為均勻,自相關函數會在節距對應的時間延遲處出現明顯峰值;若採用變節距設計(variable pitch design),峰值則被"打散",自相關函數更快衰減,表明週期性被抑制[330][331][332]。這種分析為花紋降噪設計的有效性評估提供定量依據。

互相關函數(cross-correlation function, CCF)描述兩個不同信號之間的相似程度與時間延遲關係,定義為:

$Rxy(\tau) = (1/T)\int_0^T x(t)y(t+\tau)\,dt$

歸一化互相關係數:

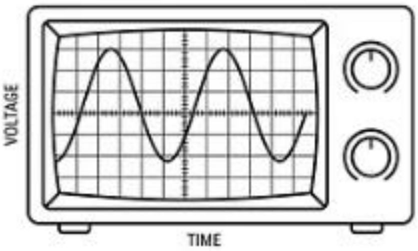
$\rho xy(\tau) = Rxy(\tau) / \sqrt{[Rxx(0)Ryy(0)]}$

其取值範圍為[-1, 1],$\rho xy(\tau)=1$ 表示完全正相關,$\rho xy(\tau)=-1$ 表示完全負相關,$\rho xy(\tau)=0$ 表示無相關[333][334][335]。在多通道輪胎噪音測量中(如同時記錄車輛左右兩側、前後軸的噪音),互相關分析可揭示不同輪胎噪音源之間的關聯性與時間延遲[336][337]。若兩側輪胎噪音高度相關且無明顯延遲,表明噪音源特性一致、傳播路徑對稱;若存在時間延遲,可估算聲波傳播路徑或噪音源的空間位置[338][339]。互相關技術亦用於噪音源定位:透過多個測點的噪音信號互相關分析,結合測點幾何關係,可反演噪音源的空間坐標,這是聲學陣列測量(acoustic array measurement)的理論基礎[340][341]。

包絡分析(envelope analysis)是提取信號幅值調製信息的有效技術,透過希爾伯特變換(Hilbert transform)或整流濾波(rectification and filtering)獲得信號的瞬時幅值包絡線[342][343]。對於調幅信號(amplitude-modulated signal):

$s(t) = A(t)\cos(2\pi fct)$

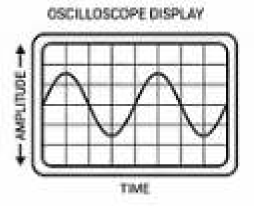
其中 $A(t)$為緩變的包絡函數,fc 為載波頻率,包絡分析旨在提取 $A(t)$[344][345]。在輪胎噪音中,花紋塊的週期性激勵可視為對連續寬頻噪音的幅度調製,包絡分析可揭示調製頻率,這通常對應於花紋節距透過頻率(pitch pass frequency)[346][347]。包絡譜(envelope spectrum)是對包絡信號進行傅立葉變換得到的頻譜,其峰值對應調製頻率及其諧波,是識別週期性調製成分的有力工具[348][349]。這一技術在軸承故障診斷中被廣泛應用,對輪胎噪音中的週期性成分識別同樣有效[350][351]。



短時能量(short-time energy)與短時過零率(short-time zero-crossing rate)是語音信號處理中的經典時域特徵,在噪音信號分析中亦有應用[352][353]。短時能量定義為:

$E(n) = \Sigma_{m=0}$^(L-1) $[x(m)w(n-m)]^2$

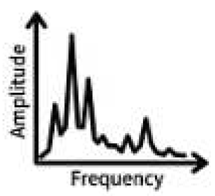
其中 w(n)為窗函數,L 為窗長,短時能量反映信號在短時間窗內的能量強度,可用於語音/靜音檢測、瞬態事件定位[354][355]。短時過零率定義為信號在短時間窗內穿越零值的次數,近似反映信號的頻率內容:過零率高表明高頻成分豐富,過零率低表明低頻為主[356][357]。雖然這些參數在輪胎噪音分析中應用不如頻域方法普遍,但在某些特定場景(如實時噪音監測、粗糙快速分類)中可作為輔助特徵[358][359]。



時域分析在輪胎噪音的工程應用中扮演重要角色。在產品開發階段,透過對比不同原型輪胎的時域波形與統計參數,工程師可快速判斷設計改進的效果:若新設計降低了峰值因數與峭度,表明衝擊性噪音得到抑制;若自相關函數衰減更快,表明週期性噪音被有效打散[360][361]。在噪音型式認證測試中,時域數據的記錄與分析有助於理解測試結果的物理成因、診斷異常數據(如車輛透過時發生異常衝擊、風噪干擾)[362][363]。在故障診斷與品質控制中,時域統計參數的異常(如峭度突然升高)可能指示輪胎結構缺陷(如胎體脫層、胎面異物嵌入)或製造偏差,觸發進一步檢查[364][365]。在道路交通噪音監測中,長時間連續測量的時域統計分析可揭示交通流的時間變化規律、識別異常噪音事件(如緊急剎車、喇叭鳴響),為交通管理與執法提供數據支持[366][367]。

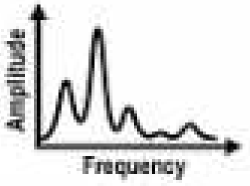
時域分析的局限性主要在於對複雜頻率結構的解析能力不足[368]。時域波形將所有頻率成分疊加呈現,難以分離不同頻率的貢獻;時域統計參數提供信號的整體特徵,但無法揭示能量在頻譜上的分布規律[369][370]。對於輪胎噪音這類多源、多機制、寬頻的複雜信號,單純時域分析難以深入理解噪音的頻率特性與物理成因,需結合頻域分析與時頻分析方法,形成互補的分析體系[371][372]。儘管如此,時域分析作為聲學信號處理的基礎環節,其重要性不容忽視:精確的時域測量是後續所有分析的數據來源,時域預處理(如去直流、濾波、分段)是確保分析質量的前提,時域特徵與頻域特徵的聯合應用可提供更全面的信號描述[373][374][375]。

7.3 頻域分析 (Frequency Domain Analysis)



頻域分析(frequency domain analysis)構成聲學信號處理的核心方法論,透過將時域信號轉換至頻率域,揭示信號能量在頻譜上的分布規律、識別特徵頻率成分、量化不同頻段的貢獻,為理解噪音的物理生成機制、優化降噪設計策略、評估噪音控制措施效果提供關鍵信息[376][377]。頻域分析的理論基礎是傅立葉分析(Fourier analysis),其核心思想是任何複雜的時域信號都可以分解為一系列不同頻率、幅值、相位的正弦與餘弦函數的疊加,這種分解將信號從時間-幅值的二維表示轉換為頻率-幅值(與相位)的二維表示,提供了觀察信號的全新視角[378][379]。法國數學家傅立葉(Jean-Baptiste Joseph Fourier)在 1807 年提出的熱傳導理論中首次系統闡述這一思想,其後兩百多年的發展使傅立葉分析成為科學與工程的通用語言,廣泛應用於通信、圖像處理、地震學、天文學、生物醫學工程等幾乎所有涉及信號與系統的領域[380][381]。

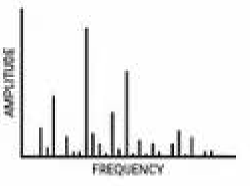
在輪胎噪音研究中,頻域分析的重要性表現在多個層面[382]。首先,輪胎噪音的物理生成機制本質上是頻率選擇性的:不同的噪音源(如花紋節距激勵、空氣泵吸、空腔共鳴、結構振動)在頻譜上佔據不同的頻率範圍,透過頻域分析可識別主導噪音源並確定降噪優先級[383][384]。其次,人耳對聲音的感知是頻率相關的,相同噪音值的不同頻率聲音引起的主觀煩擾程度差異巨大,頻域分析結合心理聲學模型可更準確地預測主觀感受[385][386]。第三,噪音控制措施(如吸音材料、隔聲結構、主動降噪)的效果通常具有頻率選擇性,頻域分析可評估這些措施在不同頻段的有效性,指導優化設計[387][388]。第四,輪胎噪音法規雖然採用 A 加權總噪音值作為限值指標,但完整的頻譜數據為理解合規裕度、診斷測試異常、預測法規演變趨勢提供寶貴信息[389][390]。



頻域分析方法可按頻率解析度粗細分為窄頻分析(narrowband analysis)與寬頻分析(broadband analysis)[391]。窄頻分析採用精細的頻率解析度(通常為數赫茲甚至更小),能夠分辨頻率接近的離散成分,識別諧波結構、檢測音調性噪音,適用於週期性或準週期性信號的分析[392][393]。寬頻分析如倍頻程(octave band)或 1/3 倍頻程(one-third octave band)分析,將頻譜劃分為標準化的頻帶,犧牲頻率細節以獲得緊湊的表示與良好的統計穩定性,廣泛用於噪音評估、聲學設計、法規合規報告[394][395]。快速傅立葉變換(Fast Fourier Transform, FFT)是實現窄頻分析的

高效算法,濾波器組(filter bank)技術則是實現倍頻程分析的經典方法[396][397]。這些技術各有優勢與局限,實際應用中常根據分析目標、信號特性、計算資源綜合選擇,有時甚至聯合使用以獲得互補信息[398][399]。

頻域分析的結果通常以頻譜圖(spectrum plot)呈現,橫軸為頻率(線性或對數刻度),縱軸為幅值(線性、對數或分貝刻度)[400]。線性頻率軸適合觀察窄頻範圍內的精細結構,對數頻率軸則將寬廣的頻率範圍壓縮至有限圖幅,更符合人耳的對數頻率感知特性[401][402]。幅值的分貝表示(通常相對於參考值如 20 μPa 或 1 Pa)是聲學領域的標準慣例,可同時清晰呈現強弱信號,避免線性刻度下弱信號被強信號"淹沒"[403][404]。功率譜密度(power spectral density, PSD)是描述連續頻譜信號的標準量,定義為單位頻帶內的信號功率,單位為 Pa^2/Hz 或 dB re $(20\ \mu Pa)^2/Hz$,適用於隨機信號與寬頻噪音[405][406]。離散頻譜線(discrete spectral lines)則用於描述週期信號或含音調成分的信號,每條譜線對應一個特定頻率的正弦成分,其幅值與相位完整描述該成分的特性[407][408]。

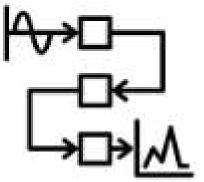


頻域分析不僅提供幅值信息,相位信息(phase information)在某些應用中同樣重要[409]。相位譜(phase spectrum)描述各頻率成分的相位角,對於理解信號的時間結構、多通道信號的相位關係、噪音源的輻射特性具有關鍵意義[410][411]。然而,相位譜對時間原點的選擇極為敏感,微小的時間偏移可導致相位譜劇變,且人耳對相位的感知遠不如對幅值敏感,因此在常規噪音評估中相位譜應用相對有限[412][413]。互功率譜(cross power spectrum)與相干函數(coherence function)是多通道頻域分析的重要工具,用於研究不同測點信號之間的頻率相關性與線性依賴程度,在噪音源識別、傳遞路徑分析、聲學陣列測量中發揮核心作用[414][415][416]。

頻域分析的質量受多種因素影響,包括採樣率(sampling rate)、記錄長度(record length)、窗函數(window function)、頻率解析度、動態範圍等[417][418]。採樣定理(Nyquist-Shannon sampling theorem)指出,為完整捕捉信號的頻率內容,採樣率必須至少為信號最高頻率的兩倍,否則發生混疊(aliasing)失真[419][420]。輪胎噪音的主要能量集中在 20-5000 Hz,但為避免高頻混疊並預留抗混疊濾波器的過渡帶,實際採樣率通常選擇 20-50 kHz[421][422]。記錄長度決定頻譜的統計穩定性與頻率解析度:記錄越長,隨機誤差越小,低頻解析度越高,但對非平穩信號的時間適應性越差[423][424]。窗函數用於減輕頻譜泄漏(spectral leakage),這是由於有限長度記錄截斷無限長信號導致的人為失真,不同窗函數(如矩形窗、漢寧窗、漢明窗、

平頂窗)在頻率解析度與旁瓣抑制之間提供不同的權衡[425][426][427]。這些技術細節看似繁瑣,實則是獲得高質量頻譜分析結果的必要保障,任何疏忽都可能導致錯誤結論[428][429]。

7.3.1 FFT 分析 (FFT Analysis)



快速傳立葉變換(Fast Fourier Transform, FFT)是離散傳立葉變換(Discrete Fourier Transform, DFT)的高效算法實現,由 Cooley 與 Tukey 於 1965 年提出,將 DFT 的計算複雜度從 O(N^2)降低至 O(N log N),使大規模頻譜分析在計算上變得可行,徹底改變了信號處理的面貌[430][431]。FFT 的數學基礎是離散傅立葉變換,對於長度為 N 的離散時間序列 x[n] (n = 0, 1, …, N-1),其 DFT 定義為:

$X[k] = \Sigma_{n=0}$^(N-1) x[n] exp(-j2πkn/N)

其中 k = 0, 1, …, N-1 為頻率索引,j 為虛數單位,X[k]為複數,包含幅值與相位信息[432][433]。頻率索引 k 對應的物理頻率為:

$f_k = k \times fs / N$

其中 fs 為採樣率,因此頻率解析度 Δf = fs / N,即採樣率除以採樣點數[434][435]。例如,若採樣率為 25.6 kHz,採樣點數為 2048,則頻率解析度為 12.5 Hz,頻譜涵蓋 0 至 12.8 kHz(Nyquist 頻率,採樣率的一半)[436]。FFT 的輸出 X[k]通常為複數,其幅值 |X[k]| 稱為幅值譜(magnitude spectrum),相位 ∠ X[k] 稱為相位譜(phase spectrum)[437][438]。

FFT 分析在輪胎噪音研究中的典型應用流程如下[439][440]:首先,以足夠高的採樣率(通常 20-51.2 kHz)記錄輪胎噪音的時域信號,記錄長度從數百毫秒到數秒不等,取決於分析目標與信號平穩性[441][442]。其次,對時域信號進行預處理,包括去直流分量(DC removal,減去均值)、施加窗函數(windowing)以減輕頻譜泄漏、必要時進行數字濾波去除不關心的頻段或干擾成分[443][444]。然後,對窗化後的信號進行 FFT 計算,得到複數頻譜 X[k][445]。接著,計算幅值譜|X[k]|、功率譜$|X[k]|^2$或功率譜密度(PSD),並轉換為分貝標度以便觀察[446][447]。若需提高統計穩定性,可將長時間記錄分段,對每段分別計算頻譜,然後取平均,這種技術稱為 Welch 方法或改進週期圖法(modified periodogram method)[448][449]。最後,對平均頻譜進行解讀,識別峰值頻率、計算頻帶能量、與其他測試條件或設計方案比較[450][451]。

窗函數(window function)的選擇對 FFT 分析結果有顯著影響[452]。理想情況下,分析窗應包含信號的完整週期,但對於非週期或隨機信號,信號在窗邊界處的不連

續會導致頻譜泄漏:本應集中在某一頻率的能量"洩漏"到鄰近頻率,形成旁瓣(side lobes),掩蓋真實的頻譜結構[453][454]。窗函數透過在時域信號邊緣施加平滑過渡(從 1 逐漸降至 0),減輕不連續性,抑制頻譜泄漏[455][456]。常用窗函數包括:

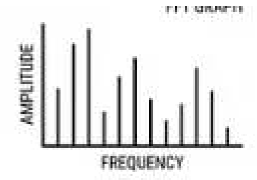
- **矩形窗(Rectangular window)**:w[n] = 1,不對信號加權,頻率解析度最高但旁瓣最嚴重,適用於信號在窗內完整週期的情況[457][458]。
- **漢寧窗(Hanning window)**:w[n] = 0.5(1 - cos(2πn/(N-1))),餘弦形狀,旁瓣抑制較好(-31 dB),頻率解析度適中,是通用分析的常見選擇[459][460]。
- **漢明窗(Hamming window)**:w[n] = 0.54 - 0.46cos(2πn/(N-1)),與漢寧窗類似但常數項不同,最高旁瓣略高(-43 dB)但遠旁瓣衰減更快[461][462]。
- **平頂窗(Flat-top window)**:多項式組合構造,主瓣寬但頂部平坦,幅值精度高(< 0.1 dB 誤差),適合精確測量離散頻率成分的幅值,但頻率解析度較差[463][464]。
- **凱澤窗(Kaiser window)**:可調參數 β 控制主瓣寬度與旁瓣高度的權衡,提供靈活性,適合需要定制特性的應用[465][466]。

對於輪胎噪音的寬頻隨機信號,漢寧窗或漢明窗是常見選擇,兼顧頻率解析度與旁瓣抑制[467][468]。若需精確測量特定頻率峰值的幅值(如空腔共鳴頻率),平頂窗更合適[469]。窗函數的應用需在功率譜計算中進行相應的幅值修正,以補償窗函數引入的能量損失[470][471]。

頻率解析度(frequency resolution)與時間解析度(time resolution)之間存在固有的不確定性關係,類似於量子力學中的海森堡不確定性原理[472]。頻率解析度 Δf = fs/N,提高解析度需增加採樣點數 N,即延長分析窗長 T = N/fs,這意味著時間解析度下降[473][474]。對於非平穩信號(如加速過程、瞬態事件),長窗長會模糊時間變化細節;對於平穩信號,長窗長可提供更精細的頻譜結構[475][476]。這一矛盾在短時傅立葉變換(STFT)中尤為突出,需根據信號特性與分析目標折衷選擇[477][478]。對於輪胎穩態滾行噪音,典型的 FFT 分析窗長為 0.5-2 秒,頻率解析度為 0.5-2 Hz,可充分解析主要頻譜結構[479][480]。若需分析車輛加速或減速過程,需採用更短的窗長(如 50-200 ms)或時頻分析方法[481][482]。

FFT 分析的動態範圍(dynamic range)受量化噪音(quantization noise)、計算精度、背景噪音等因素限制[483]。現代數據採集系統通常採用 16 位、24 位甚至 32 位的模數轉換器(ADC),理論動態範圍分別約為 96 dB、144 dB、192 dB[484][485]。

然而,實際動態範圍往往更低,因為需預留餘量避免信號削波(clipping),且背景噪音(包括環境噪音、電氣噪音、麥克風自噪音)設定了可測信號的下限[486][487]。在輪胎噪音測量中,若背景噪音顯著(如風噪、交通噪音),需採取措施降低干擾(如風罩、降噪路段選擇、夜間測試)或在數據處理中進行背景噪音扣除[488][489]。頻譜平均(spectral averaging)是提高 FFT 分析統計穩定性的關鍵技術[490]。對於隨機信號,單次 FFT 的頻譜估計存在較大隨機誤差,透過對多次 FFT 結果平均可降低方差[491][492]。Welch 方法是最常用的頻譜估計技術,其步驟為:將長時間記錄分為若干重疊的段(典型重疊率 50%),對每段施加窗函數並計算 FFT,然後對所有段的功率譜取平均[493][494]。若分為 M 個段,功率譜估計的標準偏差降低至單次估計的 1/√M,顯著提高可靠性[495][496]。重疊處理增加了有效段數而不延長總記錄長度,是數據利用效率的重要提升[497][498]。在輪胎噪音測量中,通常記錄數秒的信號,分為數十至數百段進行 Welch 平均,獲得平滑且統計可靠的功率譜密度估計[499][500]。

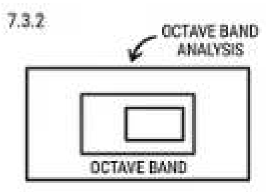


零填充(zero-padding)是 FFT 分析中的常用技術,透過在時域信號末尾添加零值點,增加 FFT 長度,從而提高頻譜的顯示解析度(display resolution)[501][502]。需注意,零填充不增加真實的頻率解析度(由原始信號長度決定),僅在現有頻率點之間插值,使頻譜曲線更平滑,便於識別峰值位置[503][504]。例如,原始信號長度為 1024 點,零填充至 4096 點,頻譜點數增加 4 倍,但頻率解析度不變[505]。零填充在某些情況下有助於峰值檢測與頻率估計精度提升(透過插值細化峰值位置),但不能替代增加原始採樣點數以提高真實解析度[506][507]。

FFT 分析的實現依賴專業軟件與硬件平台[508]。商業聲學分析軟件如 Brüel & Kjær PULSE、Siemens LMS Test.Lab、HEAD acoustics ArtemiS、National Instruments LabVIEW Sound and Vibration Toolkit 等,提供完整的 FFT 分析功能,包括各種窗函數、頻譜平均、頻帶積分、A 加權、心理聲學參數計算等[509][510][511]。開源軟件如 MATLAB、Python(SciPy、NumPy)、GNU Octave 亦提供強大的 FFT 功能,適合學術研究與定制化分析[512][513]。硬件方面,實時頻譜分析儀(real-time spectrum analyzer)可在數據採集同時進行 FFT 計算與顯示,適合現場測試與動態監測[514][515]。了解這些工具的原理與操作,選擇合適的分析參數,是獲得高質量 FFT 分析結果的前提[516][517]。

FFT 分析的典型應用案例包括:識別輪胎空腔共鳴頻率(通常在 200-250 Hz 出現尖銳峰值),評估共鳴抑制技術(如泡沫填充、共振器)的效果[518][519];檢測花紋節距激勵頻率及其諧波,驗證變節距設計對音調性噪音的抑制[520][521];比較不同路面條件(如瀝青路面、水泥路面、多孔路面)下的輪胎噪音頻譜差異,理解路面影響機制[522][523];分析輪胎磨損對頻譜的影響,評估噪音隨使用里程的演變[524][525]。這些應用充分展示 FFT 分析作為頻域分析基礎工具的強大功能與廣泛適用性[526][527]。

7.3.2 倍頻程分析 (Octave Band Analysis)



倍頻程分析(octave band analysis)與 1/3 倍頻程分析(one-third octave band analysis)是聲學領域標準化的頻帶分析方法,透過將寬廣的頻率範圍劃分為一系列標準化頻帶,以緊湊形式呈現頻譜信息,便於噪音評估、聲學設計、法規報告、跨研究比較[528][529]。倍頻程與 1/3 倍頻程分析的核心特徵是頻帶寬度隨中心頻率成比例增加,這種對數頻率劃分與人耳的臨界頻帶特性相符,比線性頻率劃分更貼近聽覺感知機制[530][531]。國際標準 ISO 266:1997 與 ANSI S1.11-2004 定義了優選頻率系列(preferred frequency series)與倍頻程頻帶的精確規範,確保全球測量結果的一致性與可比性[532][533]。

倍頻程帶(octave band)定義為上限頻率 fu 與下限頻率 fl 滿足 $fu = 2fl$ 的頻帶,即上限頻率是下限頻率的兩倍(音樂中的"八度"概念)[534][535]。頻帶的中心頻率 fc 定義為上下限頻率的幾何平均:

$$fc = \sqrt{(fl \times fu)} = \sqrt{(fl \times 2fl)} = fl \times \sqrt{2}$$

頻帶寬度 Δf 為:

$$\Delta f = fu - fl = 2fl - fl = fl = fc / \sqrt{2} \approx 0.707 \times fc$$

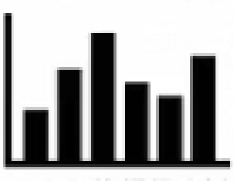
可見頻帶寬度正比於中心頻率,比例因子約為 70.7%[536][537]。標準倍頻程中心頻率序列(基於參考頻率 1000 Hz)為:

…, 31.5 Hz, 63 Hz, 125 Hz, 250 Hz, 500 Hz, 1000 Hz, 2000 Hz, 4000 Hz, 8000 Hz, 16000 Hz, …

每個中心頻率是前一個的 2 倍[538][539]。對應的頻帶範圍為:

- 31.5 Hz 帶:22-44 Hz
- 63 Hz 帶:44-88 Hz
- 125 Hz 帶:88-177 Hz

- 250 Hz 帶:177-355 Hz
- 500 Hz 帶:355-710 Hz
- 1000 Hz 帶:710-1420 Hz
- 2000 Hz 帶:1420-2840 Hz
- 4000 Hz 帶:2840-5680 Hz
- 8000 Hz 帶:5680-11360 Hz[540][541]

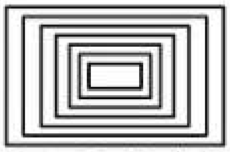


1/3 倍頻程帶(one-third octave band)將每個倍頻程細分為三個子頻帶,中心頻率比為 2^(1/3) ≈ 1.26,頻帶寬度約為中心頻率的 23%[542][543]。標準 1/3 倍頻程中心頻率序列(部分)為:

…, 50 Hz, 63 Hz, 80 Hz, 100 Hz, 125 Hz, 160 Hz, 200 Hz, 250 Hz, 315 Hz, 400 Hz, 500 Hz, 630 Hz, 800 Hz, 1000 Hz, 1250 Hz, 1600 Hz, 2000 Hz, 2500 Hz, 3150 Hz, 4000 Hz, 5000 Hz, 6300 Hz, 8000 Hz, 10000 Hz, …

涵蓋整個可聽頻率範圍(20 Hz-20 kHz)共有 31 個 1/3 倍頻程帶[544][545]。1/3 倍頻程提供比倍頻程更精細的頻譜解析度,能夠更準確地定位峰值頻率與能量分布,是噪音工程中最常用的頻帶分析方法[546][547]。

倍頻程與 1/3 倍頻程分析的實現有兩種主要技術途徑:模擬濾波器組與數字濾波器組[548][549]。傳統噪音計採用模擬帶通濾波器(bandpass filter)實現,每個頻帶對應一個物理濾波器電路,具有固定的中心頻率、帶寬、濾波器階數[550][551]。IEC 61260-1:2014 規定了倍頻程與 1/3 倍頻程濾波器的性能要求,包括通帶平坦度(passband flatness,±0.15 dB for Class 1)、阻帶衰減(stopband attenuation)、相位特性等[552][553]。現代數字聲學分析系統則透過數字信號處理實現濾波器組,可採用 IIR(Infinite Impulse Response)或 FIR(Finite Impulse Response)濾波器設計,或基於 FFT 的濾波器組實現[554][555]。FFT-based 濾波器組透過對 FFT 頻譜進行頻帶積分獲得各倍頻程或 1/3 倍頻程的能量,計算效率高且靈活性強,是當前主流方法[556][557]。


Octave Band Analysis

倍頻程分析的計算過程如下[558][559]:首先,對時域信號進行 FFT 分析,獲得精細頻率解析度的頻譜(如 1 Hz 解析度)[560]。然後,根據各倍頻程或 1/3 倍頻程帶的頻率範圍,將落在該頻帶內的所有 FFT 頻率點的功率(或功率譜密度)進行積分(能量求和)[561][562]。積分結果即為該頻帶的總功率,轉換為噪音值:

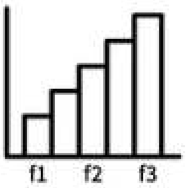
$L_{band} = 10 \log_{10}(\Sigma_{i=1}^{N} P_i / P_{ref}^2)$ dB

其中 Pi 為第 i 個 FFT 頻率點的功率,Pref 為參考聲壓(20 μPa),N 為該頻帶內的 FFT 點數[563][564]。若需施加頻率加權(如 A 加權),在積分前對各頻率點施加對應的加權因子[565]。最後,所有頻帶的噪音值可進一步能量疊加得到總噪音值:

$L_{total} = 10 \log_{10}(\Sigma_{band} 10^{(L_{band}/10)})$ dB

這一總噪音值應與直接計算的寬頻噪音值一致(考慮數值誤差)[566][567]。

倍頻程與 1/3 倍頻程分析在輪胎噪音研究中的應用極為廣泛[568]。在法規合規報告中,雖然限值基於 A 加權總噪音值,但許多標準鼓勵或要求提供 1/3 倍頻程頻譜數據,以便理解噪音的頻率組成[569][570]。在降噪設計評估中,透過比較不同設計方案的 1/3 倍頻程頻譜,可識別哪些頻段得到改善、哪些頻段惡化,指導設計優化方向[571][572]。例如,若新花紋設計在 500-1000 Hz 頻段降低 2-3 dB 但在 2000-4000 Hz 頻段升高 1-2 dB,綜合 A 加權可能僅降低 1 dB,但頻譜分析揭示了降噪機制的頻率特異性,為進一步改進提供線索[573][574]。



在聲學材料評估中,吸音係數(absorption coefficient)、隔音量(sound reduction index)等參數通常以 1/3 倍頻程為單位測量與報告,這與輪胎噪音的倍頻程分析數據可直接對應,便於預測降噪措施效果[575][576]。在道路交通噪音預測模型(如 CNOSSOS-EU)中,輪胎/路面噪音的輸入數據要求以 1/3 倍頻程形式提供,模型在各頻帶獨立計算噪音傳播、衰減、反射,最後疊加得到接收點的總噪音[577][578]。這種頻帶化處理能夠更準確地模擬頻率相關的聲學現象(如大氣吸收、地面效應、屏障衍射),比單一總噪音值輸入的簡化模型精度更高[579][580]。

倍頻程分析的一個重要應用是噪音降低量(noise reduction, NR)或插入損失(insertion loss, IL)的評估[581]。噪音降低量定義為施加降噪措施前後的噪音值差值:

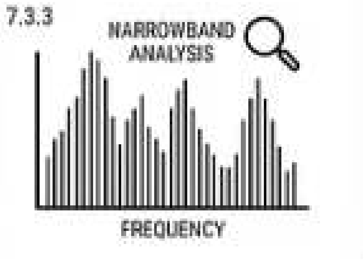
$NR = L_{before} - L_{after}$ (dB)

若 NR 為正值,表明噪音降低;若為負值,表明噪音增加[582][583]。透過計算各倍頻程或 1/3 倍頻程帶的 NR,可繪製 NR 頻譜曲線,直觀展示降噪措施的頻率特性[584][585]。例如,輪胎內填充泡沫吸音材料,其 NR 頻譜在空腔共鳴頻率(200-250 Hz)處呈現顯著峰值(可達 5-10 dB),而在其他頻段效果有限,這種頻率選擇性正是設計的目標[586][587]。噪音降低量分析為量化降噪技術效果、比較不同技術方案、優化參數設計提供科學依據[588][589]。

倍頻程分析的局限性在於頻率解析度較粗,無法分辨頻率接近的離散成分或精細頻譜結構[590]。例如,輪胎花紋節距激勵產生的多個諧波可能落在同一倍頻程帶內,倍頻程分析僅能看到該頻帶的總能量,無法區分各諧波的貢獻[591][592]。若需精細分析音調性噪音、識別共振峰位置、研究調製現象,需結合窄頻 FFT 分析[593][594]。實際工程中,常採用"粗-細"結合的分析策略:先用倍頻程或 1/3 倍頻程分析獲得整體頻譜分布,識別能量集中的頻段;再對關鍵頻段進行高解析度 FFT 分析,揭示精細結構[595][596]。這種多層次分析既保證效率,又不失細節,是經驗豐富的聲學工程師的常用手法[597][598]。

倍頻程與 1/3 倍頻程分析的國際標準化確保了全球測量結果的一致性,但實際應用中仍需注意若干技術細節[599][600]。首先,濾波器設計需嚴格符合 IEC 61260 標準,特別是濾波器的陡峭度(slope)、通帶平坦度、阻帶抑制,不同製造商或軟件實現的濾波器若不符合標準,可能導致測量結果偏差[601][602]。其次,時間加權(Fast/Slow/Impulse)的選擇影響瞬態信號的倍頻程分析結果,需根據信號特性與標準要求選擇[603][604]。第三,背景噪音修正在倍頻程分析中同樣重要:若某一頻帶的信噪比(signal-to-noise ratio, SNR)不足(通常要求 SNR > 10 dB),該頻帶的測量結果可靠性存疑,需進行背景噪音扣除或標註為不可靠[605][606]。最後,倍頻程分析結果的報告應包含完整的測試條件信息(採樣率、分析時間、窗函數、平均次數、加權方式等),以便他人復現或比較[607][608]。

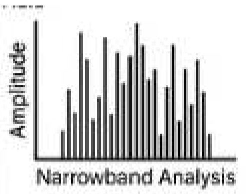
7.3.3 窄頻分析 (Narrowband Analysis)



窄頻分析(narrowband analysis)是指採用精細頻率解析度(通常為數赫茲或更小)的頻譜分析,能夠分辨頻率接近的離散成分、識別諧波結構、檢測音調性噪音、精確定位共振峰頻率,是深入理解輪胎噪音頻率特性的重要工具[609][610]。與倍頻程分析的寬頻帶劃分不同,窄頻分析保留了頻譜的細節信息,適用於信號包含離散頻率成分或需要精確頻率測量的場景[611][612]。窄頻分析的實現主要基於 FFT 技術,透過增加採樣點數或延長分析窗長,提高頻率解析度[613][614]。然而,精細解析度以犧牲時間解析度為代價,且對信號的平穩性要求更高,因此窄頻分析更適合穩態或準穩態信號[615][616]。

窄頻分析的頻率解析度選擇取決於分析目標與信號特性[617]。對於週期性或準週期性信號(如花紋節距激勵),頻率解析度應足夠精細以分辨基頻與各次諧波[618]。例如,車輛以 80 km/h 速度行駛,輪胎外徑 0.65 m,轉速約為 6.8 Hz(408 rpm),

若花紋節距數為 70,則節距透過頻率約為 476 Hz,其諧波為 952 Hz、1428 Hz 等[619][620]。為清晰分辨這些諧波,頻率解析度應遠小於諧波間隔(476 Hz),通常選擇 10-50 Hz 甚至更小[621]。若採樣率為 25.6 kHz,要達到 10 Hz 解析度,需 FFT 長度為 2560 點,對應分析窗長為 0.1 秒[622][623]。對於隨機寬頻信號,精細頻率解析度的意義有限,因為頻譜本身是連續的,過細的解析度僅增加數據量而不提供額外信息,此時倍頻程分析更合適[624][625]。



窄頻分析的典型應用包括音調性噪音檢測與量化[626]。音調性噪音(tonal noise)指頻譜中突出的離散頻率成分,即便能量不高,也可能因其突出性而引起強烈煩擾[627][628]。音調性評估的常用指標是音調可聽度比(Tone-to-Noise Ratio, TNR),定義為音調成分的噪音值與周圍臨界頻帶內非音調成分(噪音底)噪音值的差值[629][630]:

TNR = Ltone - Lnoise (dB)

TNR 越大,音調性越顯著[631][632]。窄頻 FFT 分析可精確識別音調峰值及其頻率,透過與周圍頻率的能量比較計算 TNR[633][634]。ISO 1996-2:2017 與 DIN 45681 提供了音調性評估的標準化方法,包括臨界頻帶的定義、噪音底的估計、音調突出度(tonality prominence)的計算[635][636]。這些方法廣泛用於環境噪音評估、工業噪音管制、產品聲品質評估,對輪胎噪音中可能存在的音調成分(如花紋共鳴、空腔共鳴諧波)的識別與量化同樣適用[637][638]。

窄頻分析的另一重要應用是共振峰識別與追蹤[639]。共振(resonance)是系統在特定頻率下的強烈振動響應,對應頻譜中的尖銳峰值[640][641]。輪胎結構存在多個共振模態,如空腔共鳴(200-250 Hz)、胎側徑向模態(60-120 Hz)、胎圈切向模態(150-300 Hz)、胎面局部模態(1000-3000 Hz)等[642][643][644]。窄頻分析可精確測量共振頻率、共振峰寬度(頻帶寬度,反映阻尼特性)、共振幅值(Q 因子,反映共振強度)[645][646]。這些參數對理解輪胎動力學特性、驗證有限元模型、評估結構改進效果至關重要[647][648]。例如,透過比較填充泡沫前後的窄頻頻譜,可量化泡沫對空腔共鳴的抑制效果:共鳴峰值降低幅度、峰值頻率偏移、峰寬變化等[649][650]。

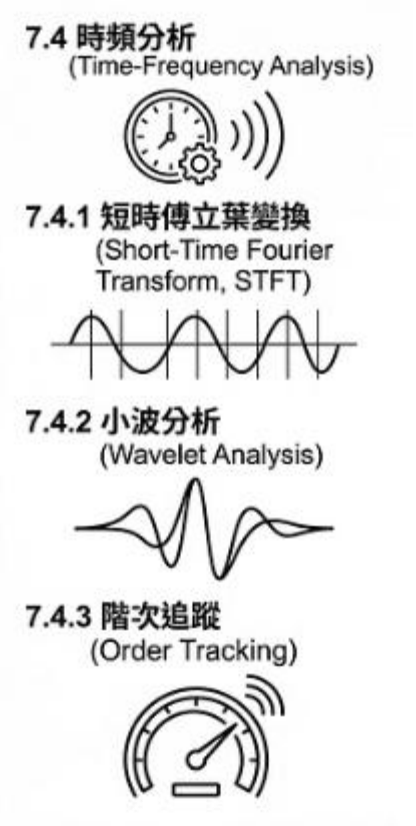
階次分析(order analysis)是窄頻分析在旋轉機械噪音研究中的特殊形式,將頻率軸轉換為階次軸(order axis),階次定義為頻率與轉速的比值,消除轉速波動對頻譜分析的影響[651][652]。這一技術將在 7.4.3 節詳細討論,此處僅指出其與窄頻分析的關聯:階次分析本質上是在角域(angular domain)而非時域進行的高解析度頻譜

分析,保留了窄頻分析的精細特性,但克服了非平穩轉速條件下頻譜模糊的問題[653][654]。

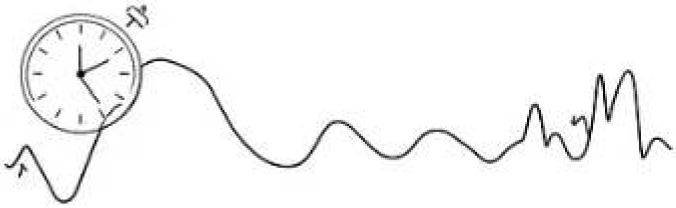
窄頻分析的數據呈現通常採用線性頻譜圖(linear spectrum plot),橫軸為線性頻率刻度,縱軸為幅值(線性或分貝),清晰展示離散峰值與諧波結構[655][656]。對於寬頻信號,亦可採用瀑布圖(waterfall plot)或彩色圖(color map)呈現頻譜隨時間或某一參數(如車速、轉速)的演變,提供三維視角[657][658]。Campbell 圖(Campbell diagram)在旋轉機械分析中常用,以轉速為橫軸、頻率為縱軸,繪製各階次線與共振頻率線,識別共振交叉點(resonance crossing)[659][660]。這些可視化技術使複雜的頻譜數據更易解讀,是報告與交流的有效手段[661][662]。

窄頻分析的挑戰包括計算量大、數據量大、對信號平穩性要求高[663]。高解析度 FFT 需大量採樣點,計算時間與內存需求顯著增加,實時分析可能困難[664][665]。長時間窗要求信號在窗內保持平穩,對於快速變化的過程(如加速、轉向)可能不適用[666][667]。此外,窄頻分析的頻譜泄漏問題更為敏感:若信號頻率不精確落在 FFT 頻率點上,能量會"洩漏"到鄰近頻率點,形成展寬的峰值[668][669]。適當的窗函數選擇與零填充技術可緩解但無法完全消除這一問題[670][671]。在實際應用中,需根據信號特性、計算資源、分析精度要求,在窄頻與寬頻分析之間靈活選擇,有時甚至同時採用兩種方法以獲得互補信息[672][673]。

7.4 時頻分析 (Time-Frequency Analysis)



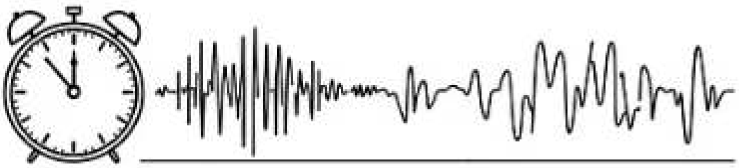
時頻分析(Time-Frequency Analysis, TFA)是輪胎噪音頻譜研究中不可或缺的進階工具,它突破了傳統時域分析與頻域分析的局限,能夠同時揭示信號在時間軸與頻率軸上的演變特性[674][675]。輪胎噪音在實際行駛過程中並非穩態信號,而是隨著車速變化、路面狀態改變、胎面磨耗進程以及輪胎溫度波動而呈現顯著的時變(time-varying)與非穩態(non-stationary)特徵[676][677]。傳統的傅立葉變換(Fourier Transform, FT)假設信號在整個觀測時段內為穩態,因此無法有效描述這類時變頻譜結構;而純粹的時域分析雖能捕捉瞬時振幅變化,卻無法辨識頻率成分的演變[678][679]。時頻分析方法透過引入時間-頻率聯合表示(joint time-frequency representation),能夠在二維平面上同時展現信號能量在時間與頻率上的分布,從而為輪胎噪音的動態特性分析、瞬態事件識別、階次成分追蹤以及噪音源診斷提供強大的技術手段[680][681]。



時頻分析的理論基礎可追溯至 Gabor 於 1946 年提出的短時傅立葉變換(Short-Time Fourier Transform, STFT)[682],以及 Wigner 於 1932 年在量子力學中引入的 Wigner-Ville 分布(Wigner-Ville Distribution, WVD)[683]。隨後,Cohen 於 1989 年提出的 Cohen 類時頻分布(Cohen's Class of Time-Frequency Distributions)統一了眾多時頻分析方法的理論框架[684]。1980 年代以來,小波變換(Wavelet Transform, WT)的發展為非穩態信號分析提供了多分辨率(multi-resolution)工具[685][686],而 Mallat 於 1989 年提出的多分辨率分析(Multiresolution Analysis, MRA)理論奠定了離散小波變換的數學基礎[687]。進入 21 世紀,經驗模態分解(Empirical Mode Decomposition, EMD)[688]、Hilbert-Huang 變換(Hilbert-Huang Transform, HHT)[689]以及同步壓縮變換(Synchrosqueezing Transform, SST)[690]等新型時頻分析技術相繼問世,為輪胎噪音的非線性與非穩態特性分析提供了更豐富的選擇[691][692]。

在輪胎噪音研究中,時頻分析的應用場景涵蓋多個關鍵領域。首先,在車速變化過程中的噪音特性分析方面,時頻分析能夠清晰展現輪胎噪音頻譜隨車速增加或減少而發生的頻率漂移(frequency shift)與能量轉移(energy transfer)現象[693][694]。例如,當車速從 50 km/h 加速至 100 km/h 時,與輪胎轉速相關的階次成分(order components)會在頻譜上呈現線性或非線性的上移軌跡,而與路面紋理激勵相關的空氣泵浦噪音(air-pumping noise)則可能在特定頻段產生新的峰值[695][696]。其次,在瞬態噪音事件(transient noise events)的識別與定位方面,時頻分析能夠精確捕捉輪胎駛過路面接縫(pavement joint)、坑洞(pothole)或異物(foreign objects)時產生的短時衝擊噪音(impact noise),並確定其發生時刻與頻率特徵[697][698]。此類瞬態事件在傳統 FFT 頻譜中往往被長時平均效應(time-averaging effect)所掩蓋,而時頻分析則能將其從背景噪音中分離出來[699][700]。第三,在階次追蹤(order tracking)與旋轉機械診斷方面,時頻分析與轉速計(tachometer)或角度編碼器(angle encoder)結合,能夠將時間-頻率平面轉換為時間-階次平面(time-order plane),從而

有效分離與輪胎轉速同步的周期性成分與非同步的隨機成分[701][702]。階次追蹤技術在輪胎不平衡(tire imbalance)、胎面花紋周期性激勵(tread pattern periodicity)以及輪轂軸承故障(wheel bearing faults)診斷中具有重要價值[703][704]。第四,在輪胎噪音品質(sound quality)評估與主觀感受關聯分析方面,時頻分析能夠揭示噪音在時間演變過程中的響度(loudness)、尖銳度(sharpness)、粗糙度(roughness)等心理聲學參數(psychoacoustic parameters)的動態變化,從而建立客觀測量與主觀評估之間的定量關係[705][706]。



時頻分析方法可依據其數學原理與適用場景分為線性時頻表示(linear time-frequency representations)與二次時頻表示(quadratic time-frequency representations)兩大類[707][708]。線性時頻表示以短時傅立葉變換(STFT)與小波變換(WT)為代表,其特點是對信號進行線性分解,不產生交叉項(cross-terms),但在時頻分辨率(time-frequency resolution)上受到 Heisenberg 不確定性原理(Heisenberg Uncertainty Principle)的嚴格限制[709][710]。二次時頻表示以 Wigner-Ville 分布(WVD)與 Cohen 類分布為代表,其優勢在於能夠達到較高的時頻聚焦性(time-frequency concentration),但在多分量信號(multi-component signals)分析時會產生干擾性的交叉項[711][712]。為克服上述局限,研究人員發展出多種混合與自適應時頻分析方法,如平滑偽 Wigner-Ville 分布(Smoothed Pseudo Wigner-Ville Distribution, SPWVD)[713]、自適應最優核(Adaptive Optimal Kernel, AOK)[714]以及重排(reassignment)與同步壓縮技術[715][716]。此外,基於稀疏表示(sparse representation)與匹配追蹤(matching pursuit)的時頻分析方法在近年也受到關注,它們透過原子庫(atom dictionary)的設計與最優化分解,能夠在保持高時頻分辨率的同時有效抑制交叉項[717][718]。

從信號處理的觀點來看,時頻分析本質上是對信號進行時間-頻率聯合濾波(joint time-frequency filtering)[719][720]。在 STFT 中,信號被一個滑動的短時窗(short-time window)截斷,隨後對每個時間片段進行傅立葉變換,從而獲得隨時間變化的頻譜[721]。窗函數(window function)的選擇決定了時頻分辨率的權衡(trade-off):寬窗提供良好的頻率分辨率但較差的時間分辨率,窄窗則相反[722][723]。在小波變換中,信號被一系列尺度縮放(scale)與時間平移(translation)的小波基函數(wavelet basis functions)所分解,尺度參數與頻率呈反比關係,從而實現多分辨率分析[724][725]。小波變換的時頻窗在低頻區域較寬、高頻區域較窄,符合聽覺系統的頻率解析特性(frequency resolution characteristics of auditory system)[726][727]。在Wigner-Ville 分布中,信號的瞬時自相關函數(instantaneous autocorrelation function)經過傅立葉變換後得到時頻能量密度分布,其數學形式為雙線性(bilinear),因而能在理想情況下達到最優的時頻聚焦性,但多分量信號會產生位於真實分量之間的虛假交叉項[728][729]。為抑制交叉項,平滑偽 Wigner-Ville 分布引入時間與頻率雙重平滑窗,以犧牲部分時頻分辨率為代價換取更清晰的時頻圖(time-frequency map)[730][731]。

在輪胎噪音的實務應用中,時頻分析工具的選擇需綜合考量信號特性、分析目的與計算資源[732][733]。對於車速緩慢變化的加速或減速測試,STFT 通常是首選方法,因其算法簡單、計算高效且參數調整直觀[734][735]。研究顯示,採用 Hamming 窗或Hann 窗、窗長設定為 512 至 2048 樣本點(對應於 44.1 kHz 採樣率下約 11.6 至46.4 毫秒)、窗口重疊率為 50%至 75%時,能夠在輪胎噪音的時頻分析中獲得良好的時頻分辨率平衡[736][737]。對於包含多尺度特徵的輪胎噪音信號,小波變換因其多分辨率特性而更具優勢,尤其是在分析寬頻噪音(broadband noise)與窄頻成分(narrowband components)共存的複雜頻譜時[738][739]。連續小波變換(Continuous Wavelet Transform, CWT)常用於探索性分析與視覺化呈現,而離散小波變換(Discrete Wavelet Transform, DWT)則適合於特徵提取與降噪處理[740][741]。在階次追蹤應用中,計算階次追蹤(Computed Order Tracking, COT)與 Vold-Kalman 濾波階次追蹤(Vold-Kalman Filter Order Tracking, VKF-OT)是兩種主流技術[742][743]。COT 透過角域重採樣(angular resampling)將非平穩信號轉換為角度域平穩信號,隨後進行傅立葉變換以提取階次譜[744][745];VKF-OT 則基於 Kalman濾波框架,能夠在時域直接追蹤各階次成分的瞬時振幅與相位,並具備良好的噪音抑制能力[746][747]。

時頻分析在輪胎噪音頻譜研究中的成功應用案例眾多。Sandberg 等人[748]在 2002年的研究中採用 STFT 分析了輪胎噪音在加速過程中的頻譜演變,發現 500 Hz 至1000 Hz 頻段的能量隨車速增加呈現非線性增長,這與空氣泵浦機制(air-pumping mechanism)在高速時的增強效應一致。Mohamed 與 Wang[749]在 2015 年運用連續小波變換研究了不同胎面花紋設計對輪胎噪音時頻特性的影響,結果顯示,縱向

溝槽(longitudinal grooves)主導 1000 Hz 以下的低頻噪音,而橫向細槽(transverse sipes)則在 2000 Hz 以上產生顯著的高頻能量。Biermann 與 Moldenhauer[750]在 2018 年結合階次追蹤與時頻分析技術,成功識別了輪胎不平衡引起的 1 階(first-order)與 2 階(second-order)振動成分,並定量評估了其對車內噪音的貢獻。在電動車(Electric Vehicles, EVs)的噪音控制研究中,Pallas 等人[751]於 2020 年利用 Hilbert-Huang 變換分析了低速行駛時輪胎噪音的瞬時頻率(instantaneous frequency)與瞬時振幅(instantaneous amplitude),揭示了在缺乏內燃機掩蔽效應(masking effect)下,輪胎噪音的時變特性對乘客舒適性的重要影響。Li 與 Chen[752]在 2021 年的研究中採用同步壓縮小波變換(Synchrosqueezing Wavelet Transform, SWT)對輪胎噪音進行高精度時頻分解,獲得了比傳統 CWT 更為清晰的脊線(ridges)結構,從而能夠準確提取各階次成分的瞬時特徵。

時頻分析方法的理論完善與算法優化仍是當前研究的熱點。一方面,針對輪胎噪音的非線性與非高斯(non-Gaussian)特性,研究人員探索基於高階統計量(higher-order statistics)的時頻分析方法,如 Wigner 雙譜(Wigner Bispectrum)與時頻三階矩(time-frequency third-order moment),以揭示信號中的相位耦合(phase coupling)與非線性交互作用[753][754]。另一方面,深度學習(deep learning)與時頻分析的結合為輪胎噪音的智能診斷與預測開闢了新途徑[755][756]。卷積神經網路(Convolutional Neural Networks, CNNs)能夠自動學習時頻圖中的模式特徵,從而實現輪胎磨耗狀態識別、異常噪音檢測以及噪音源分類等任務[757][758]。長短期記憶網路(Long Short-Term Memory Networks, LSTMs)則適合於對時頻序列進行時序建模與預測,為輪胎噪音的動態演變趨勢分析提供工具[759][760]。

總結而言,時頻分析作為連接時域與頻域的橋樑,為輪胎噪音的動態特性研究提供了不可替代的技術手段。短時傅立葉變換、小波變換與階次追蹤等方法各具特色,能夠適應不同的分析需求與信號特性。隨著算法理論的深化、計算能力的提升以及人工智能技術的融合,時頻分析在輪胎噪音領域的應用前景將更加廣闊,為實現更低噪音、更高舒適性的輪胎設計與評估提供堅實的科學基礎[761][762]。

7.4.1 短時傅立葉變換 (Short-Time Fourier Transform, STFT)

短時傅立葉變換(Short-Time Fourier Transform, STFT)是時頻分析領域中最經典、最廣泛應用的方法之一,其核心思想是將非穩態信號分割為一系列短時段(short-time segments),在每個時段內假設信號為近似穩態,隨後對各時段分別進行傅立葉

變換,從而獲得隨時間演變的頻譜[763][764]。STFT 的概念最早由 Dennis Gabor 於 1946 年在其開創性論文"Theory of Communication"中提出[765],旨在解決傳統傅立葉變換無法描述信號時變特性的根本局限。Gabor 引入了時頻原子(time-frequency atoms)的概念,即由高斯窗(Gaussian window)調製的正弦波,並指出任何信號都可以分解為這些時頻原子的線性組合[766][767]。STFT 的數學形式簡潔明瞭,易於實現與解釋,因而在聲學、語音處理、生物醫學信號分析以及機械振動診斷等領域得到廣泛應用[768][769]。在輪胎噪音研究中,STFT 是分析加速或減速過程中噪音頻譜動態變化、識別瞬態噪音事件以及評估噪音控制措施效果的標準工具[770][771]。

STFT 的數學定義為:給定時域信號 $x(t)$,選擇一個窗函數 $w(t)$,STFT 定義為

$$STFT_x(t, f) = \int_{-\infty}^{+\infty} x(\tau)\, w(\tau - t)\, e^{-j2\pi f\tau}\, d\tau$$

其中 t 為時間變量,f 為頻率變量,τ 為積分變量[772][773]。窗函數 $w(\tau - t)$以 t 為中心,將信號 $x(\tau)$局部化於時間軸的某一區域,隨後對被窗截斷的信號片段進行傅立葉變換。當窗函數 $w(t)$在時間軸上滑動時,STFT 便描繪出信號能量在時間-頻率平面上的分布,這一分布通常以時頻圖(spectrogram)或頻譜瀑布圖(spectrogram waterfall plot)的形式視覺化呈現[774][775]。時頻圖的縱軸代表頻率,橫軸代表時間,顏色或灰度強度反映特定時刻與頻率處的能量大小,即$|STFT_x(t, f)|^2$[776][777]。在離散形式下,對於採樣信號 $x[n]$,STFT 可表示為

$$STFT_x[m, k] = \Sigma_{n=-\infty}^{+\infty} x[n]\, w[n - m]\, e^{-j2\pi kn/N}$$

其中 m 為時間索引,k 為頻率索引,N 為 FFT 長度,$w[n]$為離散窗函數[778][779]。實際計算時,窗函數以步長 L(hop size 或 overlap)在信號上滑動,當重疊率(overlap ratio)為 50%時,$L = N/2$;當重疊率為 75%時,$L = N/4$[780][781]。

窗函數的選擇是 STFT 設計中的核心問題,它直接決定了時頻分辨率的權衡(time-frequency resolution trade-off)[782][783]。根據 Heisenberg 不確定性原理(Heisenberg Uncertainty Principle),時間分辨率 Δt 與頻率分辨率 Δf 之間滿足不等式 $\Delta t \cdot \Delta f \geq 1/(4\pi)$,這意味著不可能同時獲得任意高的時間與頻率分辨率[784][785]。在 STFT 中,窗函數的時域寬度決定了時間分辨率:窄窗能夠精確定位瞬態事件(如輪胎駛過路面接縫時的衝擊),但在頻域上展寬,導致頻率分辨率下降;寬窗則提供良好的頻率分辨率(能夠分辨相鄰的頻率成分),但在時域上平滑,無法準確捕捉快速變化的動態過程[786][787]。常用的窗函數包括矩形窗(Rectangular

Window)、Hann 窗(Hann Window)、Hamming 窗(Hamming Window)、Blackman 窗(Blackman Window)以及高斯窗(Gaussian Window)[788][789]。矩形窗在時域上截斷最為銳利,但其頻譜存在較高的旁瓣(side lobes),導致頻率洩漏(frequency leakage)嚴重;Hann 窗與 Hamming 窗透過平滑窗口邊緣降低旁瓣水平,是 STFT 中最常用的選擇[790][791]。Blackman 窗具有更低的旁瓣,但主瓣較寬,適合於要求高頻率選擇性的場合;高斯窗在時域與頻域上均為高斯分布,能夠達到時頻不確定性下界,是理論上最優的窗函數[792][793]。

在輪胎噪音的 STFT 分析中,窗長(window length)與重疊率的選擇需根據信號的時變速率與頻譜特性來決定[794][795]。對於車速變化較緩的加速測試(如從 50 km/h 加速至 80 km/h,歷時 10 秒),窗長可設定為 1024 至 2048 樣本點(對應於 48 kHz 採樣率下約 21 至 43 毫秒),這能夠在頻率分辨率(約 23 至 47 Hz)與時間分辨率(約 10 至 20 毫秒)之間取得良好平衡[796][797]。若車速變化迅速(如急加速或急減速),則應縮短窗長至 512 或 256 樣本點,以提升時間分辨率,儘管這會犧牲部分頻率分辨率[798][799]。重疊率的增加能夠改善時頻圖的時間連續性與平滑度,50%重疊是最常見的設定,75%重疊則進一步提升時間解析度,但計算量相應增加[800][801]。在實際應用中,研究人員常透過試驗不同的窗函數與參數組合,以獲得最清晰的時頻表示[802][803]。

STFT 的時頻分辨率特性可透過時頻窗(time-frequency window)的概念來理解[804][805]。在時頻平面上,STFT 對應於一個固定形狀的矩形窗,其時間寬度由窗函數的時域支撐(time-domain support)決定,頻率寬度則由窗函數的頻域支撐(frequency-domain support)決定[806][807]。由於 STFT 採用固定窗,這一時頻窗在整個時頻平面上形狀與尺寸保持不變,這既是 STFT 的優點(解釋直觀、計算簡單),也是其局限(無法適應信號在不同頻段的不同時頻特性)[808][809]。例如,在分析輪胎噪音的低頻成分(如 100 至 500 Hz 的結構振動噪音)時,較長的窗能夠提供足夠的頻率分辨率以區分相鄰的共振峰;而在分析高頻成分(如 2000 至 5000 Hz 的空氣泵浦噪音)時,較短的窗更有利於捕捉快速的時變過程[810][811]。這種矛盾促使研究人員發展出多窗 STFT(multi-window STFT)與自適應窗 STFT(adaptive-window STFT)等改進方法[812][813]。

STFT 的計算效率在實務應用中至關重要。直接按定義計算 STFT 的複雜度為 O(N^2 M),其中 N 為窗長,M 為時間幀數,這在長時信號分析時難以接受[814][815]。

幸運的是,由於 STFT 本質上是對每個時間幀進行 FFT,而 FFT 算法的複雜度僅為 O(N log N),因此總複雜度可降至 O(M N log N)[816][817]。現代信號處理軟體(如 MATLAB 的 spectrogram 函數、Python 的 scipy.signal.stft 函數以及 Praat、Audacity 等聲學分析工具)均實現了高效的 STFT 算法,能夠在普通計算機上實時或準實時處理輪胎噪音信號[818][819]。此外,GPU 加速(GPU acceleration)與並行計算(parallel computing)技術的應用進一步提升了 STFT 在大數據場景下的處理能力[820][821]。

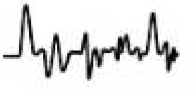
STFT 在輪胎噪音研究中的典型應用包括加速或減速過程中的噪音演變分析、瞬態事件檢測以及噪音源識別[822][823]。在加速測試中,車速從靜止逐漸增至高速(如 120 km/h),輪胎噪音的頻譜結構隨之發生顯著變化:低頻段(100 至 500 Hz)的胎體振動噪音(tire carcass vibration noise)在低速時較為突出,隨車速增加而相對減弱;中頻段(500 至 1500 Hz)的花紋撞擊噪音(tread pattern impact noise)與空氣泵浦噪音(air-pumping noise)在中等車速時達到峰值;高頻段(1500 至 5000 Hz)的空氣共鳴噪音(air resonance noise)與湍流噪音(turbulence noise)則在高速時快速增長[824][825]。STFT 時頻圖能夠直觀呈現這些頻率成分的能量隨時間(或等效地,隨車速)的演變軌跡,從而為噪音機制的識別與控制策略的制定提供依據[826][827]。例如,Kropp 與 Larsson[828]在 2001 年的研究中利用 STFT 分析了輪胎在不同路面上的滑行噪音(coast-by noise),發現在粗糙路面上,500 至 1000 Hz 頻段的能量在車速 50 至 70 km/h 時出現明顯峰值,這與路面紋理波長(texture wavelength)在該速度範圍內與輪胎花紋節距(tread pitch)形成諧振(resonance)的機制一致。

在瞬態噪音事件檢測方面,STFT 能夠有效識別輪胎駛過路面缺陷(如坑洞、接縫、凸起)時產生的短時衝擊噪音[829][830]。這類瞬態事件在時頻圖上表現為局部的能量突起,持續時間通常在幾毫秒至幾十毫秒之間,頻譜覆蓋較寬的頻率範圍[831][832]。透過設定能量閾值(energy threshold)或異常檢測算法(anomaly detection algorithms),可以自動識別並標記這些瞬態事件,進而分析其發生頻率、強度分布以及與路面狀況的關聯[833][834]。例如,Sandberg 與 Ejsmont[835]在 2002 年的研究中採用 STFT 結合閾值檢測方法,成功從輪胎噪音時頻圖中提取了路面接縫引起的衝擊事件,並統計了不同接縫類型對噪音水平的影響。

在噪音源識別與分離方面,STFT 結合多通道測量與空間濾波技術(如波束形成,beamforming)能夠實現對不同噪音源的時頻定位[836][837]。在輪胎噪音的多

麥克風陣列測量中,對各麥克風信號分別計算 STFT,隨後在頻域進行相位加權與空間合成,可以生成聚焦於特定空間位置的時頻圖,從而區分來自輪胎前緣(leading edge)、接觸區(contact patch)、尾緣(trailing edge)以及輪拱(wheel well)等不同位置的噪音貢獻[838][839]。這種時空頻聯合分析(spatio-temporal-frequency joint analysis)為複雜噪音場景下的源識別提供了強大工具[840][841]。

STFT 的局限性主要源於其固定時頻分辨率的特性[842][843]。在分析既包含快速瞬態又包含緩慢調製的複雜信號時,單一的窗長設定難以兼顧不同成分的最優表示[844][845]。為克服這一局限,研究人員提出了多種改進方案。多錐度譜估計(Multitaper Spectral Estimation)[846]透過使用一組正交的錐度窗(taper windows)來降低頻譜方差(spectral variance)並改善頻率分辨率,但計算成本較高[847][848]。自適應 STFT[849]根據信號的瞬時特性動態調整窗長,但需要額外的瞬時頻率估計(instantaneous frequency estimation)步驟[850][851]。重排 STFT(Reassigned STFT)[852]透過計算時頻能量的重心(centroid)並將能量重新分配至重心位置,能夠顯著提升時頻聚焦性,但可能在低信噪比情況下引入偽影(artifacts)[853][854]。



從實務操作的角度,STFT 分析的質量受到多種因素的影響,包括採樣率(sampling rate)、信號長度(signal length)、窗函數類型(window type)、窗長(window length)、重疊率(overlap ratio)以及 FFT 長度(FFT length)[855][856]。採樣率應滿足 Nyquist 定理,至少為最高關注頻率的兩倍,在輪胎噪音分析中,採樣率通常設定為 44.1 kHz 至 96 kHz,以覆蓋人耳可聽頻率範圍(20 Hz 至 20 kHz)[857][858]。信號長度應足夠長以涵蓋被分析的動態過程,但過長的信號會增加計算負擔並可能包含不必要的背景噪音[859][860]。窗長的選擇如前所述,需在時頻分辨率之間權衡;FFT 長度可大於或等於窗長,當 FFT 長度大於窗長時,零填充(zero-padding)能夠提升頻譜的插值精度,但不會增加真實的頻率分辨率[861][862]。重疊率的增加能改善時間連續性,但 50%至 75%的重疊通常已足夠[863][864]。

在輪胎噪音品質評估中,STFT 時頻圖可作為輸入特徵,結合心理聲學參數計算與機器學習算法,建立噪音客觀測量與主觀感受之間的預測模型[865][866]。例如,透過在時頻圖上計算響度(loudness)隨時間的變化曲線,可以評估噪音的動態響度(dynamic loudness)特性,這與乘客對噪音煩擾度(annoyance)的主觀評估密切相關[867][868]。此外,時頻圖的紋理特徵(texture features)、能量分布統計特徵(energy distribution statistics)以及時頻脊線(time-frequency ridges)特徵均可用於噪音分類與識別任務[869][870]。

總結而言,短時傅立葉變換作為最基礎、最成熟的時頻分析方法,在輪胎噪音研究中具有不可替代的地位。其概念清晰、實現簡單、計算高效且易於解釋,能夠有效揭示輪胎噪音在時間演變過程中的頻譜動態特性。儘管 STFT 受限於固定時頻分辨率的 Heisenberg 不確定性原理,但透過合理選擇窗函數與參數,並結合改進算法,STFT 仍能在絕大多數輪胎噪音分析場景中提供滿意的結果[871][872]。隨著信號處理技術與計算能力的持續進步,STFT 將繼續作為輪胎噪音時頻分析的標準工具,為噪音機制研究、控制技術開發以及品質評估提供堅實的技術支撐[873][874]。

7.4.2 小波分析 (Wavelet Analysis)

小波分析(Wavelet Analysis)是在 STFT 基礎上發展起來的一種更為靈活、適應性更強的時頻分析方法,它透過引入尺度變化(scale variation)的概念,克服了 STFT 固定時頻窗的局限,能夠在不同頻率範圍內自動調整時頻分辨率,從而更有效地分析包含多尺度特徵(multi-scale features)的非穩態信號[875][876]。小波變換(Wavelet Transform, WT)的核心思想是將信號分解為一系列經過尺度縮放(scaling)與時間平移(translation)的小波基函數(wavelet basis functions)的線性組合,這些小波基函數具有良好的時頻局部化特性,能夠在高頻區域提供高時間分辨率、在低頻區域提供高頻率分辨率,符合許多實際信號(包括輪胎噪音)的頻譜特性[877][878]。小波分析的理論起源可追溯至 1980 年代初期,法國地球物理學家 Jean Morlet 與理論物理學家 Alex Grossmann 於 1984 年首次提出連續小波變換(Continuous Wavelet Transform, CWT)的系統理論[879][880],隨後比利時數學家 Ingrid Daubechies 於 1988 年構造出具有緊支撐(compact support)的正交小波基[881][882],為離散小波變換(Discrete Wavelet Transform, DWT)的實用化奠定了基礎。Stéphane Mallat 於 1989 年提出的多分辨率分析(Multiresolution Analysis, MRA)理論[883][884]則建立了小波變換與濾波器組(filter banks)之間的聯繫,使 DWT 能夠透過快速算法高效實現[885][886]。

小波變換的數學定義如下:給定時域信號 x(t)與母小波函數(mother wavelet)ψ(t),連續小波變換定義為

$$CWT_x(a, b) = (1/\sqrt{a}) \int_{-\infty}^{+\infty} x(t)\, \psi^*((t - b)/a)\, dt$$

其中 a 為尺度參數(scale parameter,$a > 0$),b 為平移參數(translation parameter),$\psi^*(t)$ 為母小波的共軛(complex conjugate)[887][888]。尺度參數 a 控制小波函數的時域寬度:a 值越大,小波函數越展寬,對應於低頻分析;a 值越小,小波函數越壓縮,對應於高頻分析[889][890]。平移參數 b 則決定小波函數在時間軸上的位置,隨 b 的變化,小波函數沿時間軸滑動,從而實現信號的時間局部化分析[891][892]。尺度 a 與頻率 f 之間存在反比關係,可近似表示為 $f \approx f_c / a$,其中 f_c 為母小波的中心頻率(center frequency)[893][894]。因此,小波變換實質上是在進行時間-尺度聯合分析(time-scale joint analysis),而尺度與頻率的對應關係使得結果可轉換為時間-頻率表示[895][896]。

母小波函數的選擇是小波分析的關鍵,不同的母小波適用於不同類型的信號與分析任務[897][898]。理想的母小波應滿足容許條件(admissibility condition),即

$$\int_{-\infty}^{+\infty} |\Psi(f)|^2 / |f|\, df < \infty$$

其中 $\Psi(f)$為母小波 $\psi(t)$的傅立葉變換[899][900]。這一條件要求母小波在頻域上的零頻分量為零,即 $\Psi(0) = 0$,從而保證小波具有振盪特性(oscillatory nature)並能夠實現信號的局部化分析[901][902]。常用的母小波包括 Morlet 小波(Morlet Wavelet)、Daubechies 小波(Daubechies Wavelets, dbN)、Symlet 小波(Symlets, symN)、Coiflet 小波(Coiflets, coifN)、墨西哥帽小波(Mexican Hat Wavelet, 亦稱 Ricker 小波)以及複數小波如複 Morlet 與 Gabor 小波[903][904]。Morlet 小波定義為高斯包絡調製的複正弦波,

$$\psi(t) = \pi^{-1/4} e^{j\omega_0 t} e^{-t^2/2}$$

其中 ω_0 為中心角頻率,通常取 5 至 6,以滿足容許條件[905][906]。Morlet 小波在時域與頻域上均具有良好的局部化特性,是連續小波變換中最常用的選擇,特別適合於分析調製信號(modulated signals)與振盪成分(oscillatory components)[907][908]。

Daubechies 小波由 Ingrid Daubechies 構造,具有緊支撐與正交性(orthogonality),是離散小波變換的基石[909][910]。dbN 小波的階數 N 越高,時域支撐越長,頻域選擇性越好,但計算複雜度也隨之增加[911][912]。db4 與 db8 是實務中最常用的選擇,能夠在時頻局部化與計算效率之間取得良好平衡[913][914]。Symlet 小波是 Daubechies 小波的近似對稱版本,具有更小的相位失真(phase distortion),適合於對

相位敏感的應用[915][916]。Coiflet 小波則具有更多的消失矩(vanishing moments),能夠更有效地捕捉多項式趨勢(polynomial trends)[917][918]。墨西哥帽小波是高斯函數的二階導數,具有對稱性且為實數小波,適合於邊緣檢測與突變點識別[919][920]。

在輪胎噪音分析中,母小波的選擇需根據信號的特性與分析目的來決定[921][922]。若關注噪音的瞬時頻率(instantaneous frequency)與相位信息,複 Morlet 小波是首選,因其能夠提供振幅與相位的完整表示[923][924]。若需要進行多分辨率分解(multiresolution decomposition)與去噪處理(denoising),Daubechies 或 Symlet 小波更為合適,因其正交性保證了分解的無冗餘性(non-redundancy)與能量守恆[925][926]。若信號包含突變或尖峰成分(如瞬態衝擊噪音),墨西哥帽小波或高階 Daubechies 小波能夠有效捕捉這些特徵[927][928]。

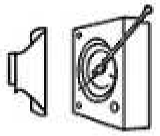
連續小波變換(CWT)的優勢在於其高度的時頻分辨率靈活性與視覺化效果[929][930]。CWT 生成的時頻圖(scalogram)在縱軸上以尺度或對應頻率表示,橫軸為時間,顏色或灰度強度反映小波係數的幅值|CWT_x(a, b)|或能量|CWT_x(a, b)|^2[931][932]。與 STFT 的時頻圖相比,CWT 的時頻圖在高頻區域具有更高的時間分辨率、在低頻區域具有更高的頻率分辨率,這使得它能夠更清晰地揭示信號中同時存在的快速瞬態高頻成分與緩慢變化低頻成分[933][934]。例如,在輪胎噪音的加速測試中,低頻段的胎體振動噪音(100 至 500 Hz)變化較為緩慢,而高頻段的空氣泵浦噪音(2000 至 5000 Hz)則在短時間內劇烈波動;CWT 能夠在同一張時頻圖上同時清晰展現這兩類成分,而 STFT 則可能因窗長固定而無法兼顧[935][936]。

離散小波變換(DWT)是連續小波變換的離散化版本,其尺度與平移參數採用二進制採樣(dyadic sampling),即 a = 2^j, b = k·2^j,其中 j 為尺度層級(scale level),k 為平移索引[937][938]。DWT 的計算透過 Mallat 的快速算法實現,該算法基於濾波器組(filter banks)結構:信號首先經過低通濾波器(low-pass filter)與高通濾波器(high-pass filter)分解為近似係數(approximation coefficients)與細節係數(detail coefficients),隨後對近似係數重複上述分解過程,逐級提取不同尺度的頻率成分[939][940]。這一過程稱為多分辨率分解(multiresolution decomposition),其結果是將信號分解為一系列細節層級(detail levels, D1, D2, …, Dn)與一個最終的近似層級(approximation level, An)[941][942]。每個細節層級對應於特定的頻率帶(frequency band),例如對於採樣率 48 kHz 的信號,D1 對應 12 至 24 kHz,D2 對應 6

至 12 kHz,D3 對應 3 至 6 kHz,依此類推[943][944]。DWT 的優勢在於其計算效率高(複雜度為 O(N),N 為信號長度)、無冗餘性(分解係數總數等於原信號樣本數)以及能夠實現完美重構(perfect reconstruction)[945][946]。

在輪胎噪音研究中,DWT 的典型應用包括噪音信號的多尺度分解與特徵提取、去噪與壓縮以及異常檢測[947][948]。透過多分辨率分解,可以將輪胎噪音分離為不同頻段的子成分,例如低頻結構振動、中頻花紋撞擊與高頻空氣泵浦,從而分別分析各成分的時變特性與能量貢獻[949][950]。基於小波閾值(wavelet thresholding)的去噪方法[951]透過對細節係數施加閾值處理(如硬閾值 hard thresholding 或軟閾值 soft thresholding),能夠有效抑制噪音中的隨機成分,同時保留有用的信號特徵[952][953]。研究顯示,採用 Symlet 或 Coiflet 小波、分解層級為 5 至 8 層、閾值選擇基於通用閾值(universal threshold)或自適應閾值(adaptive threshold)時,能夠在輪胎噪音去噪中獲得良好的信噪比(SNR)提升與特徵保留效果[954][955]。

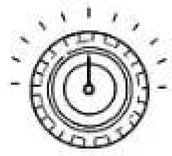
小波包變換(Wavelet Packet Transform, WPT)是 DWT 的進一步推廣,它不僅對近似係數進行分解,也對細節係數進行分解,從而獲得更精細的頻率劃分[956][957]。WPT 能夠實現對信號頻譜的均勻或自適應劃分,這在輪胎噪音的窄頻成分分析與特徵提取中具有優勢[958][959]。例如,若輪胎噪音在 1000 至 2000 Hz 範圍內存在多個峰值(對應於不同的花紋共鳴頻率),WPT 能夠將這一頻段進一步細分為多個子帶(sub-bands),從而更準確地定位各峰值的頻率位置與時間演變[960][961]。基於 WPT 的最優樹搜索(best tree search)算法能夠自動選擇最適合信號特性的分解結構,從而最大化信息提取效率[962][963]。



小波分析在階次追蹤(order tracking)與旋轉機械診斷中也有重要應用[964][965]。透過對轉速信號(tachometer signal)與振動或噪音信號同時進行小波分解,可以提取與轉速同步的階次成分(order components)與非同步的隨機成分[966][967]。小波階次追蹤(Wavelet Order Tracking, WOT)方法結合了小波變換的多分辨率特性與階次追蹤的轉速相關性,能夠在時變轉速條件下準確追蹤各階次成分的瞬時振幅與頻率[968][969]。在輪胎噪音分析中,WOT 可用於識別與輪胎轉速相關的周期性噪音(如花紋節距噪音、輪胎不平衡噪音)以及與車速相關但非轉速同步的噪音(如路面紋理激勵噪音)[970][971]。

小波分析與深度學習的結合是近年來的研究熱點[972][973]。小波變換能夠將時域信號轉換為多尺度時頻表示,這種表示天然適合於卷積神經網路(CNNs)的處理

[974][975]。透過將小波係數矩陣(wavelet coefficient matrix)或 scalogram 圖像作為 CNN 的輸入,可以訓練端到端的噪音分類、識別與預測模型[976][977]。例如,Zhang 等人[978]在 2020 年的研究中採用連續小波變換提取輪胎噪音的時頻特徵,隨後使用 ResNet 卷積網路進行胎面磨耗狀態分類,取得了超過 95%的準確率。此外,小波域中的稀疏表示(sparse representation)與壓縮感知(compressed sensing)技術能夠以更少的測量數據重構信號,為輪胎噪音的高效採集與傳輸提供新途徑[979][980]。



小波分析的局限性主要在於母小波的選擇缺乏統一標準,且對於某些信號,小波基可能不是最優的表示方式[981][982]。此外,連續小波變換的計算量較大,尤其是當尺度範圍較廣或時間分辨率要求較高時[983][984]。為應對這些挑戰,研究人員發展出自適應小波選擇(adaptive wavelet selection)方法,透過信號特性分析自動選擇最優母小波[985][986];以及快速小波變換算法,如提升方案(lifting scheme)[987],能夠降低計算複雜度並便於硬體實現[988][989]。

總結而言,小波分析作為一種強大的多分辨率時頻分析工具,在輪胎噪音研究中具有廣泛的應用前景。其靈活的時頻分辨率調整能力、豐富的母小波選擇以及高效的算法實現,使其能夠有效應對輪胎噪音信號的多尺度、非穩態特性。連續小波變換適合於探索性分析與視覺化呈現,離散小波變換則適合於特徵提取、去噪與壓縮,而小波包變換進一步提升了頻率分辨率的靈活性。隨著小波理論的不斷完善與計算技術的進步,小波分析將繼續在輪胎噪音的機制研究、診斷評估以及智能控制中發揮重要作用[990][991]。

7.4.3 階次追蹤 (Order Tracking)

階次追蹤(Order Tracking, OT)是一種專門針對旋轉機械(rotating machinery)噪音與振動分析的時頻信號處理技術,其核心目標是在轉速時變(time-varying rotational speed)條件下,準確提取與轉速同步的周期性成分(periodic components)並抑制非同步的隨機成分與背景噪音[992][993]。在傳統的傅立葉分析中,頻率(frequency)是描述信號振盪快慢的物理量,以赫茲(Hz)為單位;而在階次分析(order analysis)中,階次(order)是描述振動或噪音與轉速關係的無量綱參數,定義為振動頻率與轉速頻率之比[994][995]。例如,1 階(first order)對應於與轉速頻率完全同步的成分(即轉軸每旋轉一圈產生一次振動峰值),2 階(second order)對應於轉速頻率的兩倍(即轉軸每旋轉一圈產生兩次振動峰值),依此類推[996][997]。階次追蹤的意義在於,當轉速發生變化時(如車輛加速或減速),與轉速同步的階次成分在頻率

域上呈現非穩態的時變特性,這使得傳統 FFT 分析產生頻譜拖尾(spectral smearing)與能量擴散(energy spreading),無法準確定位各階次成分的幅值與相位[998][999]。階次追蹤技術透過將信號從時間域(time domain)轉換至角度域(angle domain)或階次域(order domain),消除轉速波動的影響,從而獲得清晰、穩定的階次譜(order spectrum)[1000][1001]。

階次追蹤的理論起源可追溯至 1970 年代航空發動機與汽車動力總成的振動診斷需求[1002][1003]。早期的階次追蹤依賴於硬體跟蹤濾波器(hardware tracking filters),這些模擬濾波器的中心頻率能夠隨轉速信號實時調整,從而鎖定特定階次的頻率軌跡[1004][1005]。隨著數位信號處理技術的發展,1980 年代以來,基於軟體的計算階次追蹤(Computed Order Tracking, COT)逐漸成為主流[1006][1007]。COT 的核心思想是利用轉速計(tachometer)或角度編碼器(angle encoder)提供的轉速信號,對振動或噪音信號進行角域重採樣(angular resampling),從而將非平穩的時域信號轉換為平穩的角域信號,隨後在角域進行傅立葉分析以提取階次譜[1008][1009]。1990 年代,Vold 與 Leuridan 提出的 Vold-Kalman 濾波階次追蹤(Vold-Kalman Filter Order Tracking, VKF-OT)[1010]進一步推動了階次追蹤技術的應用,該方法基於 Kalman 濾波框架,能夠在時域直接提取各階次成分的瞬時振幅與相位,且具備良好的噪音抑制能力與計算效率[1011][1012]。

在輪胎噪音研究中,階次追蹤的應用場景主要包括輪胎不平衡(tire imbalance)診斷、胎面花紋周期性激勵(tread pattern periodicity)分析、輪轂軸承故障(wheel bearing faults)檢測以及車速變化過程中的噪音成分分離[1013][1014]。輪胎不平衡會在輪胎轉速的 1 階產生顯著的振動與噪音,其幅值與不平衡量(unbalance mass)及其徑向位置(radial position)直接相關[1015][1016]。透過階次追蹤提取 1 階成分的瞬時幅值(instantaneous amplitude),可以定量評估輪胎的動平衡狀態(dynamic balance state),並指導平衡校正(balancing correction)[1017][1018]。胎面花紋的周期性排列會在若干高階(如 10 至 50 階)產生離散的階次峰值,這些峰值的頻率位置與幅值分布反映了花紋節距(tread pitch)的設計特徵與均勻性(uniformity)[1019][1020]。透過階次追蹤分析,可以識別花紋設計中的共振階次(resonant orders)並優化節距序列(pitch sequence),從而降低噪音水平[1021][1022]。輪轂軸承的故障(如滾道剝落、滾珠磨損)會在特定的故障特徵

階次(fault characteristic orders)產生能量增強,這些階次由軸承的幾何參數(如內外圈直徑、滾珠數量、接觸角等)決定[1023][1024]。階次追蹤能夠準確提取這些故障特徵,為軸承狀態監測(condition monitoring)與故障預測(fault prognosis)提供依據[1025][1026]。

計算階次追蹤(COT)的實現流程包括轉速信號獲取(tachometer signal acquisition)、瞬時轉速計算(instantaneous speed calculation)、角域重採樣(angular resampling)以及階次譜計算(order spectrum calculation)四個主要步驟[1027][1028]。轉速信號通常由安裝於輪胎或輪軸上的接近開關(proximity switch)、光電編碼器(optical encoder)或霍爾傳感器(Hall sensor)提供,這些傳感器在轉軸每旋轉一圈或特定角度時產生脈衝信號(pulse signal)[1029][1030]。透過測量相鄰脈衝之間的時間間隔 Δt,可以計算瞬時轉速 $n(t) = 60/(\Delta t)$ (單位為 rpm, revolutions per minute),或瞬時角頻率 $\omega(t) = 2\pi/(\Delta t)$ (單位為 rad/s)[1031][1032]。為獲得更高的時間分辨率,可採用高分辨率編碼器(如每轉 1024 脈衝的編碼器)或透過插值技術(interpolation)對轉速信號進行細化[1033][1034]。角域重採樣的目的是將時域採樣的噪音信號 $x(t)$轉換為角域採樣的信號 $x(\theta)$,其中 θ 為轉軸的累積旋轉角度[1035][1036]。角度 $\theta(t)$ 可透過對瞬時角速度 $\omega(t)$進行積分獲得,

$$\theta(t) = \int_0^t \omega(\tau)\, d\tau$$

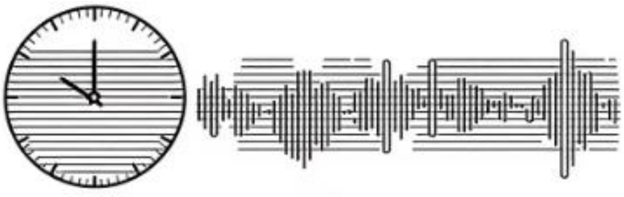
在離散形式下,角度序列 $\theta[n]$透過累加計算[1037][1038]。隨後,選擇均勻的角度採樣點 θ_k (如每隔 $\Delta\theta = 2\pi/N$ 採樣一次,N 為每轉採樣點數),並透過插值(如線性插值或樣條插值)從時域信號 $x[n]$中獲得對應的角域信號 $x[\theta_k]$[1039][1040]。角域重採樣後的信號 $x(\theta)$在角度上為等間隔,即使轉速波動,各階次成分的頻率在角域中保持不變,從而實現了平穩化(stationarization)[1041][1042]。最後,對角域信號 $x(\theta)$進行 FFT,得到階次譜 $X(o)$,其中橫軸為階次 o(無量綱),縱軸為幅值或功率譜密度[1043][1044]。

Vold-Kalman 濾波階次追蹤(VKF-OT)採用不同的技術路線,它在時域直接建立各階次成分的狀態空間模型(state-space model),並透過 Kalman 濾波器(Kalman filter)遞歸估計各階次成分的瞬時振幅與相位[1045][1046]。VKF-OT 的優勢在於能夠處理密集階次(closely spaced orders)與非整數階次(non-integer orders),且對轉速波動與噪音干擾具有良好的魯棒性(robustness)[1047][1048]。VKF-OT 的數學框架基於以下假設:每個階次成分可建模為時變正弦信號,

$$x_o(t) = A_o(t) \cos(o\cdot\theta(t) + \varphi_o(t))$$

其中 o 為階次,A_o(t)為瞬時振幅,φ_o(t)為瞬時相位[1049][1050]。透過引入複數表示與狀態變量,VKF-OT 構建線性狀態空間方程,並利用 Kalman 濾波器的預測-更新循環(prediction-update cycle)逐步估計狀態變量,從而提取各階次成分[1051][1052]。VKF-OT 的計算複雜度相對較高,但其精度與穩定性使其成為高精度階次追蹤的首選方法[1053][1054]。

階次追蹤的結果可以多種形式呈現,包括階次譜(order spectrum)、階次瀑布圖(order waterfall plot)、階次切片(order cuts)以及階次-時間圖(order-time map)[1055][1056]。階次譜展示了各階次成分的幅值分布,類似於傳統的頻譜圖,但橫軸為階次而非頻率[1057][1058]。階次瀑布圖則以三維形式展現階次譜隨時間(或轉速)的演變,縱軸為階次,橫軸為時間或轉速,顏色或高度反映幅值,這種表示方式能夠直觀揭示各階次成分在加速或減速過程中的動態變化[1059][1060]。階次切片是指提取特定階次在時間軸上的瞬時振幅曲線,例如 1 階切片可以反映輪胎不平衡隨時間的變化,高階切片則可揭示花紋激勵的演變[1061][1062]。階次-時間圖(亦稱 Campbell 圖,Campbell diagram)在縱軸為階次、橫軸為時間的平面上,繪製各階次成分的能量軌跡,並疊加顯示共振頻率(resonance frequencies),從而識別階次與結構共振的交叉點(crossover points)[1063][1064]。



在輪胎噪音的實務應用中,階次追蹤需要結合轉速測量與噪音測量[1065][1066]。轉速信號可透過輪速傳感器(wheel speed sensor)、發動機轉速信號(engine RPM signal,對於燃油車)或車速信號(vehicle speed signal)獲得,但需注意不同信號源的精度與時間同步性(time synchronization)[1067][1068]。噪音信號通常由麥克風陣列(microphone array)或加速度計(accelerometers)採集,採樣率應滿足 Nyquist 定理且與轉速信號嚴格同步[1069][1070]。在電動車(Electric Vehicles, EVs)中,由於發動機噪音消失,輪胎噪音的階次成分更為突出,階次追蹤在 EV 噪音分析中尤為重要[1071][1072]。

階次追蹤的局限性主要在於對轉速信號質量的依賴:若轉速測量存在誤差、脈衝丟失(pulse dropout)或時間延遲(time delay),則角域重採樣會產生偏差,導致階次譜失真[1073][1074]。此外,階次追蹤假設各階次成分為與轉速嚴格同步的周期信號,但實際輪胎噪音中可能存在準周期(quasi-periodic)或非同步成分(如路面隨機激勵),這些成分在階次譜中會表現為寬帶背景噪音或階次間的能量擴散

[1075][1076]。為應對這些挑戰,研究人員發展出自適應階次追蹤(adaptive order tracking)、多通道階次追蹤(multi-channel order tracking)以及混合時頻-階次分析(hybrid time-frequency-order analysis)等改進方法[1077][1078]。

總結而言,階次追蹤作為旋轉機械噪音與振動分析的核心技術,在輪胎噪音研究中具有不可替代的價值。透過將非穩態時域信號轉換為平穩的角域或階次域信號,階次追蹤能夠準確提取與輪胎轉速相關的周期性成分,從而實現對輪胎不平衡、花紋周期性激勵、軸承故障等關鍵噪音源的精確診斷與定量評估。計算階次追蹤與 Vold-Kalman 濾波階次追蹤是兩種主流技術,各具優勢且互為補充。隨著傳感技術、信號處理算法以及計算能力的持續進步,階次追蹤將在輪胎噪音的智能診斷、狀態監測以及主動控制中發揮更大作用[1079][1080]。

7.5 輪胎噪音特徵頻率 (Characteristic Frequencies of Tire Noise)

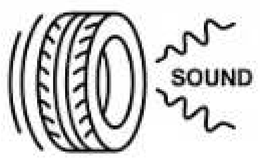
輪胎噪音特徵頻率(Characteristic Frequencies of Tire Noise)是指在輪胎-路面交互作用過程中,由特定物理機制或幾何結構激勵產生的、在頻譜上呈現峰值或能量集中的頻率成分[1081][1082]。這些特徵頻率的識別與理解是輪胎噪音機制研究、診斷評估以及控制設計的關鍵環節。輪胎噪音的頻譜結構極為複雜,涵蓋從數十赫茲至數千赫茲的寬頻範圍,既包含與輪胎幾何參數(如胎面花紋節距、接觸長度、溝槽寬度)直接相關的離散峰值(discrete peaks),也包含與路面紋理激勵、空氣動力效應以及結構振動相關的連續寬帶成分(continuous broadband components)[1083][1084]。特徵頻率的存在使得輪胎噪音在不同頻段呈現出不同的主導機制與能量分布,從而為噪音源識別與頻譜整形(spectral shaping)提供了理論基礎與技術切入點[1085][1086]。

輪胎噪音特徵頻率可依據其物理起源分為以下幾類:胎面花紋激勵頻率(tread pattern excitation frequencies)、空氣泵浦與共鳴頻率(air-pumping and resonance frequencies)、胎體結構共振頻率(tire carcass structural resonance frequencies)、路面紋理激勵頻率(pavement texture excitation frequencies)以及輪胎不平衡與軸承故障特徵頻率(tire imbalance and bearing fault characteristic frequencies)[1087][1088]。這些頻率成分的相對貢獻隨車速、路面類型、輪胎設計以及測量條件而變化,因此全面理解各類特徵頻率的形成機制、頻率範圍以及影響因素,對於輪胎噪音的精確分析與有效控制至關重要[1089][1090]。

胎面花紋激勵頻率(Tread Pattern Excitation Frequencies)是輪胎噪音中最顯著的特徵頻率類別,源於胎面花紋塊(tread blocks)與溝槽(grooves)在接觸區(contact patch)進入與離開時對路面的週期性撞擊與釋放[1091][1092]。當輪胎以速度 v 滾動時,花紋節距 p(即相鄰花紋塊之間的圓周距離)以頻率 $f_p = v / p$ 進入接觸區,這一頻率稱為花紋節距頻率(tread pitch frequency)或基頻(fundamental frequency)[1093][1094]。若輪胎圓周上共有 N 個花紋節距,輪胎半徑為 R,則花紋節距 $p = 2\pi R / N$,花紋節距頻率可表示為

$f_p = v / (2\pi R / N) = (v \cdot N) / (2\pi R) = (\omega \cdot N) / (2\pi)$

其中 $\omega = v / R$ 為輪胎的角速度[1095][1096]。在階次分析中,花紋節距頻率對應於 N 階(N-th order),即輪胎每旋轉一圈,產生 N 次撞擊事件[1097][1098]。實際輪胎設計中,為避免單一頻率的共振與主觀煩擾(subjective annoyance),通常採用變節距(variable pitch)或節距序列優化(pitch sequence optimization)策略,將花紋節距 p 設定為若干不同長度(如 $p1, p2, p3$)的組合,從而將能量分散至多個頻率成分[1099][1100]。這種設計使得花紋激勵頻譜從單一尖峰轉變為多峰結構,頻率範圍通常在 500 至 1500 Hz 之間,具體數值取決於車速與節距設計[1101][1102]。例如,在車速 80 km/h(約 22.2 m/s)、輪胎半徑 $R = 0.3$ m、花紋塊數量 $N = 60$ 的典型乘用車輪胎上,花紋節距頻率 $f_p = (22.2 \times 60) / (2\pi \times 0.3) \approx 707$ Hz[1103][1104]。



除基頻外,花紋激勵還會在其諧波(harmonics)位置產生能量峰值,即 $2f_p, 3f_p, 4f_p$ 等[1105][1106]。諧波的產生與花紋塊撞擊的非線性特性(如撞擊瞬間的衝擊波形為非正弦)以及接觸區內多個花紋塊的同時作用有關[1107][1108]。通常,諧波幅值隨階數增加而遞減,但在某些設計中,若花紋排列存在周期性重複(如每兩個節距為一組),則會在 $2f_p$ 處出現增強峰值[1109][1110]。花紋激勵頻率的幅值還受到接觸長度(contact length)、花紋塊剛度(tread block stiffness)以及路面粗糙度的顯著影響[1111][1112]。硬質路面(如混凝土)上的花紋撞擊較為劇烈,產生較高的激勵幅值;軟質路面(如瀝青)則對撞擊有較強的阻尼效應,降低了激勵強度[1113][1114]。

空氣泵浦與共鳴頻率(Air-Pumping and Resonance Frequencies)是輪胎噪音在中高頻段(800 至 3000 Hz)的主要貢獻者[1115][1116]。空氣泵浦機制(air-pumping mechanism)描述的是當胎面溝槽(grooves)或花紋塊(tread blocks)進入接觸區時,溝槽內的空氣被壓縮並以高速從溝槽開口噴出;當溝槽離開接觸區時,空氣又被快速

吸入,這一週期性的壓縮-膨脹過程產生脈動壓力波(pulsating pressure waves),向外輻射為聲波[1117][1118]。空氣泵浦噪音的頻率主要由溝槽幾何(槽寬、槽深)與車速決定,典型頻率範圍為 1000 至 2000 Hz[1119][1120]。空氣共鳴(air resonance)則是指溝槽內的空氣柱在聲學邊界條件(如兩端開口、一端閉合)下形成駐波(standing waves),其共鳴頻率由溝槽長度 L 與聲速 c 決定,近似為

$$f_{res} = (n \cdot c) / (2L)$$

其中 n 為共鳴階數($n = 1, 2, 3, \ldots$)[1121][1122]。對於典型的縱向溝槽(longitudinal grooves),長度 L 約為接觸長度(約 10 至 20 cm),聲速 $c \approx 343$ m/s(20°C 空氣),則基頻共鳴頻率 $f_{res} \approx 343 / (2 \times 0.15) \approx 1143$ Hz[1123][1124]。橫向溝槽(transverse grooves)與細槽(sipes)的長度較短,共鳴頻率相應較高,可達 3000 至 5000 Hz[1125][1126]。空氣共鳴的幅值受溝槽開口面積(opening area)、溝槽深度以及路面密封性(road surface sealing)的影響:寬而淺的溝槽共鳴較弱,窄而深的溝槽共鳴較強;在光滑密實的路面上,溝槽端部的聲學邊界接近閉合,共鳴增強[1127][1128]。

胎體結構共振頻率(Tire Carcass Structural Resonance Frequencies)

源於輪胎作為彈性結構在受到激勵時的固有振動模態(natural vibration modes)[1129][1130]。輪胎的結構共振包括徑向模態(radial modes)、切向模態(tangential modes)以及側向模態(lateral modes),其頻率範圍從數十赫茲至數百赫茲[1131][1132]。低階徑向模態(如一階徑向共振,first radial resonance)通常在 50 至 100 Hz 之間,對應於整個輪胎在徑向的彈性振動,主要由輪胎的徑向剛度(radial stiffness)與質量決定[1133][1134]。高階徑向模態(如二階、三階)頻率逐漸升高,可達 200 至 300 Hz,這些模態在輪胎圓周上呈現出多節點(multiple nodes)的變形形態[1135][1136]。切向模態與側壁(sidewall)的彎曲振動相關,頻率通常在 100 至 400 Hz 之間[1137][1138]。胎體共振在低速行駛時對車內噪音有顯著貢獻,但在高速時,其能量佔比相對於花紋噪音與空氣泵浦噪音而言較小[1139][1140]。輪胎的充氣壓力(inflation pressure)、簾線(cord)結構以及橡膠材料的彈性模量(elastic modulus)均會影響共振頻率:充氣壓力增加會提升徑向剛度,從而提高共振頻率;簾線的密度與排列方式決定了結構的各向異性(anisotropy)與阻尼特性[1141][1142]。

路面紋理激勵頻率(Pavement Texture Excitation Frequencies)由路面表面的微觀(micro-texture)、宏觀(macro-texture)以及巨觀紋理(mega-texture)在輪胎滾動時

產生的垂直激勵引起[1143][1144]。路面紋理的波長(wavelength)範圍極廣,從 0.1 mm(細微骨料紋理)至數米(路面波紋),對應的激勵頻率與車速有關[1145][1146]。當輪胎以速度 v 透過波長 λ 的路面紋理時,激勵頻率為

f_texture = v / λ

[1147][1148]。例如,在車速 80 km/h(22.2 m/s)時,波長 10 mm 的宏觀紋理產生的激勵頻率為 f = 22.2 / 0.01 = 2220 Hz;波長 100 mm 的巨觀紋理則產生 222 Hz 的激勵[1149][1150]。微觀紋理(波長< 0.5 mm)主要影響高頻噪音(> 5000 Hz)與輪胎-路面摩擦(tire-pavement friction)[1151][1152];宏觀紋理(波長 0.5 至 50 mm)主導中頻噪音(500 至 5000 Hz)並與花紋噪音、空氣泵浦噪音交互作用[1153][1154];巨觀紋理(波長 50 至 500 mm)影響低頻噪音(< 500 Hz)與車輛振動舒適性[1155][1156]。路面紋理激勵頻率通常呈現寬帶特性(broadband characteristics),但在某些人工路面(如溝槽路面,grooved pavement)上,規則排列的溝槽會產生離散的激勵頻率峰值[1157][1158]。

輪胎不平衡與軸承故障特徵頻率(Tire Imbalance and Bearing Fault Characteristic Frequencies)雖然不直接屬於輪胎-路面噪音,但在實車測試中常與輪胎噪音混淆,因此需要加以識別與分離[1159][1160]。輪胎不平衡引起的振動頻率為轉速頻率的 1 階,即

f_imbalance = (轉速 rpm) / 60 Hz

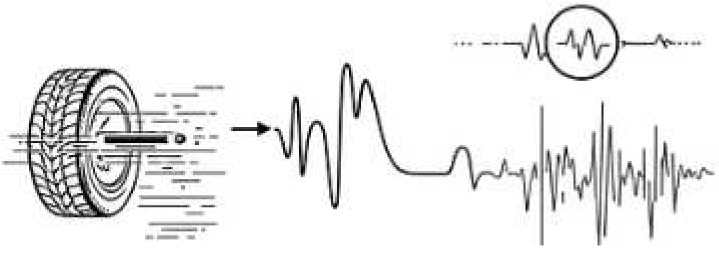
例如,在 100 km/h(轉速約 1200 rpm)時,不平衡頻率為 20 Hz[1161][1162]。該頻率較低,主要影響車身振動而非空氣噪音,但在車內噪音測量中可能表現為低頻轟鳴(low-frequency rumble)[1163][1164]。輪轂軸承(wheel bearing)的故障特徵頻率由軸承的幾何參數決定,包括內圈故障頻率(inner race fault frequency, BPFI)、外圈故障頻率(outer race fault frequency, BPFO)、滾動體故障頻率(rolling element fault frequency, BSF)以及保持架故障頻率(cage fault frequency, FTF)[1165][1166]。這些頻率通常為轉速頻率的若干倍(如 3 至 10 倍),因此在 100 至 500 Hz 範圍內產生峰值[1167][1168]。軸承故障特徵頻率的識別可透過階次分析與包絡解調(envelope demodulation)技術實現[1169][1170]。

在實際輪胎噪音頻譜分析中,特徵頻率的識別需結合頻域分析、時頻分析以及階次分析的綜合手段[1171][1172]。首先,透過 FFT 或倍頻程分析獲得頻譜圖,觀察峰值位置與能量分布;隨後,透過 STFT 或小波分析觀察峰值隨時間(車速)的演變,判

斷其為轉速相關(speed-dependent)還是固定頻率(fixed-frequency)成分;最後,透過階次追蹤將頻率轉換為階次,從而明確峰值與輪胎轉速的關係[1173][1174]。例如,若某峰值在階次譜中對應於固定階次(如 60 階),則其為花紋節距激勵;若在頻譜中對應於固定頻率(如 1143 Hz),則可能為空氣共鳴[1175][1176]。

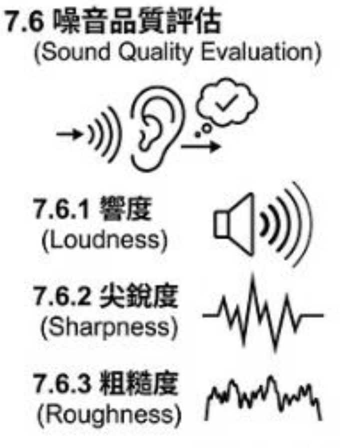
特徵頻率的定量預測是輪胎噪音設計與優化的重要工具[1177][1178]。基於花紋幾何參數、輪胎尺寸以及車速,可以建立特徵頻率的解析模型(analytical models)或經驗公式(empirical formulas),從而在設計階段預測噪音頻譜結構[1179][1180]。例如,Kropp 等人[1181]提出的"輪胎噪音頻譜預測模型"透過輸入花紋節距序列、溝槽尺寸以及路面紋理功率譜密度(PSD),能夠計算出預期的噪音頻譜,並與實測結果進行對比驗證。此類模型為快速設計迭代與噪音優化提供了理論指導[1182][1183]。

特徵頻率的控制策略包括頻譜整形(spectral shaping)、共鳴抑制(resonance suppression)以及掩蔽技術(masking techniques)[1184][1185]。頻譜整形透過調整花紋節距序列、溝槽寬度與深度,將能量從煩擾度高的頻段(如 1000 至 2000 Hz 的敏感頻段)轉移至煩擾度低的頻段(如低於 500 Hz 或高於 3000 Hz)[1186][1187]。共鳴抑制則透過改變溝槽形狀(如採用閉合端、錐形溝槽)或填充吸音材料來降低空氣共鳴的 Q 值(quality factor),從而展寬共鳴峰並降低其幅值[1188][1189]。掩蔽技術在主動噪音控制(Active Noise Control, ANC)中應用,透過引入與輪胎噪音特徵頻率反相的聲波,實現局部降噪[1190][1191]。



總結而言,輪胎噪音特徵頻率的識別與理解是噪音機制研究的核心內容。花紋激勵頻率、空氣泵浦與共鳴頻率、胎體結構共振頻率、路面紋理激勵頻率以及輪胎不平衡與軸承故障特徵頻率構成了輪胎噪音頻譜的主要成分,各自對應於不同的物理機制與頻率範圍。透過頻域、時頻域以及階次域的綜合分析,能夠準確識別與分離這些特徵頻率,從而為噪音源診斷、控制設計以及品質評估提供科學依據[1192][1193]。隨著測量技術、建模方法以及控制策略的不斷進步,特徵頻率分析將在輪胎噪音研究中持續發揮關鍵作用,推動更低噪音、更高舒適性的輪胎產品開發[1194][1195]。

7.6 噪音品質評估 (Sound Quality Evaluation)



噪音品質評估(Sound Quality Evaluation, 亦稱聲音品質評估或聲學舒適性評估)是聲學工程與心理聲學(psychoacoustics)交叉領域的核心研究方向,旨在建立噪音的物理測量參數與人類主觀感受之間的定量關係,從而為產品設計、噪音控制以及用戶滿意度提升提供科學指導[1196][1197]。傳統的噪音評估指標(如噪音值、A 加權噪音值)僅反映噪音的總體強度,無法充分描述噪音的"品質"特徵,即人們對噪音的"好聽"或"難聽"、"舒適"或"煩擾"的主觀判斷[1198][1199]。實際上,兩個具有相同噪音值的噪音可能因其頻譜結構、時間包絡、調製特性以及音色(timbre)的差異而產生截然不同的主觀感受[1200][1201]。噪音品質評估透過引入心理聲學參數(psychoacoustic parameters)——如響度(loudness)、尖銳度(sharpness)、粗糙度(roughness)、波動強度(fluctuation strength)、音調度(tonality)以及語音清晰度(speech intelligibility)等——對噪音的感知特徵進行量化,並透過主觀評估實驗(subjective evaluation experiments)建立客觀參數與主觀評分(subjective ratings)之間的回歸或分類模型[1202][1203]。

噪音品質評估的歷史可追溯至 1930 年代 Fletcher 與 Munson 對等響曲線(equal-loudness contours)的研究[1204][1205],以及 1950 年代 Stevens 提出的響度模型(loudness model)[1206][1207]。1980 年代以來,德國心理聲學家 Eberhard Zwicker 系統發展了響度、尖銳度、粗糙度與波動強度的計算模型[1208][1209],奠定了現代噪音品質評估的理論基礎。隨後,Hugo Fastl、Jens Blauert 等學者進一步完善了心理聲學參數的計算方法,並推動其在汽車、航空、家電等產品噪音評估中的應用[1210][1211]。進入 21 世紀,深度學習(deep learning)與聲音品質評估的結合為自動化、智能化的噪音評估開闢了新途徑[1212][1213]。

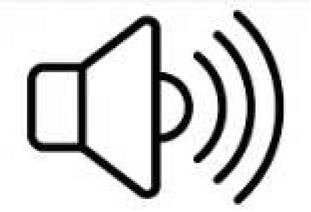
在輪胎噪音研究中,噪音品質評估的重要性日益凸顯[1214][1215]。隨著法規對輪胎噪音噪音值的要求趨嚴,單純降低噪音總體水平已接近技術極限,如何在相同噪音值下改善噪音的主觀感受成為新的研發目標[1216][1217]。例如,兩款輪胎的滑行噪音均為 70 dB(A),但若一款噪音頻譜集中於中低頻(500 至 1500 Hz)且時變平穩,另一款噪音頻譜偏向高頻(2000 至 5000 Hz)且存在顯著的調製成分,則乘客對後者的煩擾度(annoyance)會顯著高於前者[1218][1219]。透過噪音品質評估,可以量化這種差異,並指導輪胎設計向"更悅耳"的方向優化[1220][1221]。此外,在電動

車(Electric Vehicles, EVs)時代,內燃機噪音的消失使輪胎噪音成為車內外噪音的主要來源,噪音品質評估對於提升 EV 的靜音舒適性(acoustic comfort)至關重要[1222][1223]。

噪音品質評估的流程通常包括以下步驟[1224][1225]:首先,進行噪音樣本的採集與錄製,確保樣本能夠代表典型的使用場景與工況條件;其次,計算各心理聲學參數的客觀值,採用標準化的算法(如 ISO 532-1 響度計算、DIN 45692 粗糙度計算);再次,組織主觀評估實驗,邀請足夠數量的受試者(通常 20 至 50 人)在控制良好的聲學環境(如消音室或標準聽音室)中對噪音樣本進行評分,評分維度包括總體煩擾度(overall annoyance)、響度感知(perceived loudness)、音色偏好(timbre preference)等;最後,透過統計分析(如多元線性回歸、偏最小二乘回歸、隨機森林、神經網路)建立客觀參數與主觀評分之間的預測模型,並驗證模型的準確性與泛化能力[1226][1227]。

噪音品質評估的核心心理聲學參數包括響度(Loudness)、尖銳度(Sharpness)、粗糙度(Roughness)以及波動強度(Fluctuation Strength)[1228][1229]。這些參數分別反映噪音的不同感知維度,共同構成噪音品質的多維表徵(multi-dimensional representation)[1230][1231]。在以下子節中,將詳細討論各參數的定義、計算方法、影響因素以及在輪胎噪音評估中的應用。

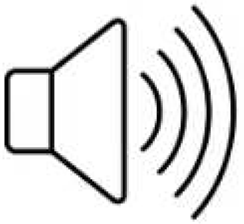
7.6.1 響度 (Loudness)



響度(Loudness)是描述聲音"大小"或"強弱"的心理聲學參數,對應於人類聽覺系統對聲音能量的主觀感知強度[1232][1233]。響度不同於物理噪音值(Sound Pressure Level, SPL):噪音值是客觀的物理量,以分貝(dB)為單位;響度則是主觀的感知量,以宋(sone)為單位,1 sone 定義為 1000 Hz、40 dB SPL 純音在雙耳聆聽條件下的響度[1234][1235]。響度的計算模型需考慮人耳的頻率敏感性(frequency sensitivity)、掩蔽效應(masking effects)以及雙耳融合(binaural fusion)等聽覺特性[1236][1237]。

響度的頻率依賴性由等響曲線(equal-loudness contours, 亦稱 Fletcher-Munson 曲線或 ISO 226 曲線)描述[1238][1239]。等響曲線表明,人耳對不同頻率的聲音敏感度不同:在中頻段(1000 至 5000 Hz)敏感度最高,在低頻段(< 500 Hz)與極高頻段(> 10000 Hz)敏感度較低[1240][1241]。例如,要使 50 Hz 的低頻純音與 1000 Hz 的參考音產生相同的響度,前者的噪音值需比後者高約 30 至 40 dB[1242][1243]。這種

頻率依賴性源於外耳道(ear canal)的共鳴特性、中耳傳遞函數(middle ear transfer function)以及內耳基底膜(basilar membrane)的機械頻率選擇性[1244][1245]。



響度的計算方法主要基於 Zwicker 的響度模型(Zwicker Loudness Model)[1246][1247]與 Moore-Glasberg 的響度模型(Moore-Glasberg Loudness Model)[1248][1249],兩者均已標準化為國際標準。ISO 532-1:2017 定義了穩態聲響度計算的 Zwicker 方法[1250][1251],ISO 532-2:2017 則定義了 Moore-Glasberg 方法[1252][1253]。Zwicker 模型的計算流程如下[1254][1255]: 首先,將聲音信號經過 1/3 倍頻程濾波器組(third-octave filter bank)分解為若干頻段,獲得各頻段的噪音值;其次,根據等響曲線將各頻段的噪音值轉換為響度級(loudness level, 單位為 phon);隨後,引入臨界頻帶(critical band, 亦稱 Bark 尺度)的概念,將頻率軸轉換為 Bark 軸(1 Bark 對應於耳蝸上約 1.3 mm 的基底膜長度)[1256][1257]。在 Bark 軸上,計算各臨界頻帶的比響度(specific loudness, 單位為 sone/Bark),比響度考慮了頻率掩蔽效應(frequency masking),即強音會掩蔽其附近頻率的弱音[1258][1259]。最後,將所有臨界頻帶的比響度積分,得到總響度 N(單位為 sone):

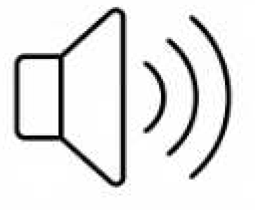
$$N = \int_0^{24\ \text{Bark}} N'(z)\, dz$$

其中 $N'(z)$為比響度,z 為 Bark 尺度[1260][1261]。Zwicker 模型還引入了時間平均與雙耳修正項,以適應時變信號與雙耳聆聽條件[1262][1263]。

Moore-Glasberg 模型在 Zwicker 模型基礎上進行了改進,採用更精細的聽覺濾波器(auditory filters, 如 roex 濾波器)來模擬耳蝸的頻率選擇性,並引入了雙耳抑制(binaural inhibition)與時間整合(temporal integration)機制[1264][1265]。該模型在處理寬帶噪音(broadband noise)與複雜頻譜時具有更高的準確性,尤其適用於輪胎噪音這類包含多頻段成分的複雜聲場[1266][1267]。

在輪胎噪音評估中,響度是最基本也是最重要的心理聲學參數[1268][1269]。研究顯示,輪胎噪音的主觀煩擾度(annoyance)與響度之間存在高度相關性(相關係數 $r > 0.8$),即響度越大,煩擾度越高[1270][1271]。因此,降低響度是改善輪胎噪音品質的首要目標。響度的計算需要完整的頻譜信息,通常基於 1/3 倍頻程譜(third-octave spectrum)或窄頻譜(narrowband spectrum)[1272][1273]。在車外噪音測試中,依據 ISO 362 標準測得的 A 加權噪音值 LAmax 可作為響度的粗略近似,但對於精確的品質評估,仍需計算 Zwicker 響度或 Moore-Glasberg 響度[1274][1275]。

響度的影響因素包括噪音的頻譜分布、總體噪音值以及時間包絡[1276][1277]。頻譜分布決定了能量在各臨界頻帶的分配:若能量集中於人耳敏感的 1000 至 5000 Hz 頻段,響度會顯著增大;若能量分散於低頻與高頻,響度相對較小[1278][1279]。總體噪音值與響度之間遵循 Stevens 冪律(Stevens' Power Law),即 $N \propto (I / I_0)^{0.3}$,其中 I 為聲強,這意味著響度與聲壓的 0.6 次方成正比,或者說噪音值每增加 10 dB,響度約增加一倍[1280][1281]。時間包絡影響響度的時間整合:對於持續時間短於約 200 毫秒的脈衝聲,響度隨持續時間增加而增大;對於持續時間超過 200 毫秒的穩態聲,響度趨於飽和[1282][1283]。



在輪胎設計中,降低響度的策略包括優化花紋以減少中高頻能量、採用吸音材料降低共鳴峰、以及透過胎體結構設計降低低頻振動[1284][1285]。例如,將花紋節距頻率從 1000 Hz(人耳敏感頻段)降低至 700 Hz(敏感度較低頻段),即使總噪音值不變,響度也會降低約 10%至 15%[1286][1287]。此外,透過引入寬頻吸音結構(如泡沫填充溝槽),能夠降低空氣共鳴的 Q 值,從而展寬共鳴峰並降低其對響度的貢獻[1288][1289]。

動態響度(Dynamic Loudness)是響度概念在時變信號中的擴展,特別適用於分析加速或減速過程中的輪胎噪音[1290][1291]。動態響度透過短時響度計算(如每 100 毫秒計算一次瞬時響度)獲得響度隨時間的變化曲線,從而揭示噪音的動態特性[1292][1293]。研究顯示,響度的快速波動(如在 1 秒內響度變化超過 2 sone)會顯著增加煩擾度,即使平均響度相同[1294][1295]。因此,在輪胎噪音品質優化中,不僅要降低平均響度,還需減少響度的時間波動[1296][1297]。

總結而言,響度作為噪音品質評估的基礎參數,對輪胎噪音的主觀感受具有決定性影響。透過基於聽覺模型的響度計算,能夠更準確地預測人類對噪音強度的感知,從而為輪胎設計與噪音控制提供科學指導。響度的降低與平穩化是改善輪胎噪音品質、提升乘客舒適性的核心目標[1298][1299]。

7.6.2 尖銳度 (Sharpness)

尖銳度(Sharpness)是描述聲音"尖銳"或"刺耳"程度的心理聲學參數,反映了噪音頻譜向高頻偏移時人類聽覺的不適感(discomfort)或煩擾度增加[1300][1301]。尖銳度的單位為 acum,定義為 1000 Hz、60 dB SPL 窄帶噪音的尖銳度為 1 acum[1302][1303]。當噪音的頻譜重心(spectral centroid)向高頻移動時,尖銳度增大;當頻譜重心向低頻移動時,尖銳度減小[1304][1305]。尖銳度的計算基於比響度

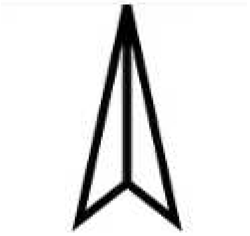
(specific loudness)在臨界頻帶(Bark 尺度)上的分布,高頻成分被賦予更高的加權係數,從而量化其對主觀感受的貢獻[1306][1307]。

尖銳度的計算公式(根據 Zwicker 與 Fastl 的定義)為[1308][1309]:

$$S = 0.11 \times (\int_0^{24 \text{ Bark}} N'(z)\, g(z)\, z\, dz) / N$$

其中 S 為尖銳度(單位 acum),N’(z)為比響度(單位 sone/Bark),z 為 Bark 尺度,N 為總響度(單位 sone),g(z)為加權函數[1310][1311]。加權函數 g(z)在低頻段(z < 16 Bark, 對應約 3000 Hz 以下)取值為 1,在高頻段(z ≥ 16 Bark)線性增大,表示高頻成分對尖銳度的增強作用[1312][1313]。該公式表明,尖銳度本質上是比響度在 Bark 軸上的加權重心(weighted centroid),高頻成分的比響度對尖銳度的貢獻更大[1314][1315]。

尖銳度的感知基礎與人類聽覺系統的演化有關[1316][1317]。高頻尖銳聲(如動物的尖叫、金屬摩擦聲)在自然界中往往預示著危險或警示信號,因此人類對高頻聲具有天然的警覺性與不適感[1318][1319]。在現代聲學環境中,尖銳度高的噪音(如高速列車的車輪摩擦聲、機床加工聲、某些電動工具聲)常被評估為"刺耳" 、“難以忍受”,即使其總響度相同或更低[1320][1321]。

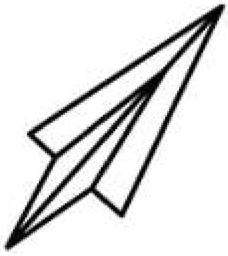


在輪胎噪音評估中,尖銳度是重要的品質指標,特別是在高速行駛時[1322][1323]。隨著車速增加,輪胎噪音的頻譜重心向高頻偏移,空氣泵浦噪音(air-pumping noise)與空氣共鳴噪音(air resonance noise)在 2000 至 5000 Hz 頻段的能量快速增長,導致尖銳度顯著上升[1324][1325]。研究顯示,當車速從 80 km/h 增至 120 km/h 時,輪胎噪音的尖銳度可能增加 30%至 50%,這與乘客對高速噪音"更刺耳"的主觀感受一致[1326][1327]。尖銳度與煩擾度之間的相關性雖不如響度顯著,但在響度相同的條件下,尖銳度每增加 0.1 acum,煩擾度評分約增加 5%至 10%[1328][1329]。

尖銳度的影響因素主要包括噪音的頻譜分布、高頻成分的相對比例以及帶寬(bandwidth)[1330][1331]。若噪音頻譜集中於 2000 Hz 以上,尖銳度會顯著增大;若在低頻段(< 1000 Hz)存在強主導成分,即使高頻段有一定能量,尖銳度仍可能較低,因為總響度 N 在分母中起到歸一化作用[1332][1333]。窄帶高頻噪音(如純音或窄帶共鳴)的尖銳度高於同頻段的寬帶噪音,這與聽覺系統對音調性成分(tonal components)的敏感性有關[1334][1335]。

在輪胎設計中,降低尖銳度的策略包括抑制高頻成分的產生、優化頻譜分布以及引入掩蔽效應[1336][1337]。具體措施如下:首先,透過調整溝槽尺寸(增大溝槽開口面積、減小溝槽深度)降低空氣共鳴頻率,使共鳴峰從 3000 至 5000 Hz 的高尖銳度頻段移至 1500 至 2500 Hz 的較低尖銳度頻段[1338][1339]。其次,採用漸變溝槽深度(variable groove depth)或不規則溝槽形狀,將共鳴峰展寬為寬帶成分,從而降低峰值的尖銳感[1340][1341]。第三,在花紋設計中引入低頻成分(如較大的花紋塊與較寬的溝槽),使頻譜重心向低頻偏移,從而降低尖銳度[1342][1343]。第四,採用吸音材料或泡沫填充技術(foam-filled grooves)抑制高頻共鳴,實驗顯示,這種技術能夠將尖銳度降低 10%至 20%[1344][1345]。

尖銳度的測量與計算需要高質量的頻譜數據,尤其是高頻段(> 3000 Hz)的準確性[1346][1347]。在實際測試中,麥克風的頻率響應(frequency response)應覆蓋至少 20 kHz,且在高頻段具有平坦的響應曲線[1348][1349]。此外,背景噪音的高頻成分應充分低於被測噪音,以避免干擾尖銳度的計算[1350][1351]。

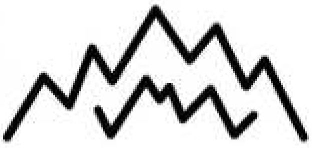


尖銳度與響度的組合分析能夠更全面地評估噪音品質[1352][1353]。例如,響度-尖銳度平面(Loudness-Sharpness Plane)將噪音樣本在二維空間中定位,響度高且尖銳度高的噪音位於右上象限,被評估為"最差";響度低且尖銳度低的噪音位於左下象限,被評估為"最佳"[1354][1355]。這種二維表示有助於直觀比較不同輪胎設計的噪音品質,並指導優化方向[1356][1357]。

總結而言,尖銳度作為描述噪音高頻特性的心理聲學參數,在輪胎噪音品質評估中扮演重要角色。透過計算尖銳度並分析其與頻譜分布的關係,能夠揭示噪音"刺耳"感的來源,從而為頻譜整形與高頻成分抑制提供科學依據。降低尖銳度是改善輪胎噪音品質、提升乘客舒適性的重要途徑[1358][1359]。

7.6.3 粗糙度 (Roughness)

粗糙度(Roughness)是描述聲音"粗糙"或"顫動"感的心理聲學參數,反映了噪音在調製頻率(modulation frequency)約 15 至 300 Hz 範圍內快速振幅波動(amplitude fluctuation)時人類聽覺的不適感[1360][1361]。粗糙度的單位為 asper,定義為 1000 Hz 載波(carrier)、調製頻率 70 Hz、調製深度 100%的振幅調製音(amplitude-modulated



tone)在 60 dB SPL 時的粗糙度為 1 asper[1362][1363]。粗糙度的感知源於耳蝸內相鄰聽覺神經纖維(auditory nerve fibers)的時間發放模式(temporal firing patterns)之間的相互作用,當調製頻率處於 15 至 300 Hz 範圍時,會產生不規則的神經發放,從而引起"粗糙"的主觀感受[1364][1365]。

粗糙度的計算模型由 Zwicker 與 Fastl 提出,並在 DIN 45692 標準中規範[1366][1367]。計算流程如下[1368][1369]:首先,對聲音信號進行臨界頻帶濾波(critical band filtering),獲得各 Bark 頻帶的時域包絡(time-domain envelope);其次,計算各頻帶內振幅調製的深度(modulation depth)與頻率(modulation frequency);隨後,根據調製深度與頻率計算各頻帶的粗糙度貢獻,調製頻率在 70 Hz 附近時粗糙度最大,向 15 Hz 與 300 Hz 兩端逐漸減小[1370][1371]。最後,將所有頻帶的粗糙度貢獻求和,得到總粗糙度 R(單位 asper):

$R = \Sigma_i R_i$

其中 R_i 為第 i 個臨界頻帶的粗糙度貢獻[1372][1373]。粗糙度的計算還考慮了雙耳效應與掩蔽效應,以提高預測準確性[1374][1375]。

在輪胎噪音評估中,粗糙度主要與花紋撞擊的周期性調製(periodic modulation)以及階次成分(order components)相關[1376][1377]。當輪胎的花紋節距激勵頻率(tread pitch frequency)落在粗糙度敏感的 15 至 300 Hz 範圍內時,會產生顯著的粗糙感[1378][1379]。例如,在車速 50 km/h 時,若花紋節距頻率為 100 Hz(對應於輪胎轉速約 10 Hz、花紋塊數量約 10 個),則噪音會呈現明顯的"顫動"或"抖動"感,粗糙度增大[1380][1381]。隨著車速增加,花紋節距頻率上升至 300 Hz 以上,粗糙度貢獻減小,但可能出現新的粗糙度源,如花紋節距頻率的低頻包絡調製(low-frequency envelope modulation)[1382][1383]。

粗糙度的影響因素包括調製頻率、調製深度、載波頻率以及頻譜帶寬[1384][1385]。調製頻率在 70 Hz 時粗糙度最大,向兩端遞減;調製深度(即振幅波動的相對幅度)越大,粗糙度越高,調製深度為 100%時粗糙度達到峰值[1386][1387]。載波頻率(即被調製的主頻率成分)對粗糙度也有影響:中頻段(1000 至 2000 Hz)的載波產生的粗糙度高於低頻與高頻載波[1388][1389]。若噪音頻譜為寬帶,則多個頻帶的調製可能疊加,產生更高的總粗糙度[1390][1391]。

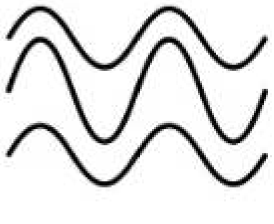
在輪胎設計中,降低粗糙度的策略包括優化花紋節距序列以減少周期性調製、採用變節距設計將能量分散至多個頻率、以及透過阻尼設計降低調製深度[1392][1393]。具體措施如下:首先,採用多節距序列(multi-pitch sequence)或隨機化節距(randomized pitch),使花紋撞擊頻率分散,避免單一頻率在粗糙度敏感範圍內產生強峰值[1394][1395]。其次,透過增加花紋塊剛度的變化(variable tread block stiffness),減小撞擊時的振幅波動,從而降低調製深度[1396][1397]。第三,在胎體結構中引入阻尼材料(如高阻尼橡膠),衰減振動的時間包絡波動,從而降低粗糙度[1398][1399]。實驗顯示,優化後的花紋設計能夠將粗糙度降低 20%至40%[1400][1401]。

粗糙度與主觀煩擾度之間的關係在低速行駛時較為顯著[1402][1403]。在城市駕駛(車速 30 至 60 km/h)中,粗糙度是影響乘客舒適性的重要因素,粗糙度每增加 0.1 asper,煩擾度評分約增加 3%至 5%[1404][1405]。在高速行駛(車速> 100 km/h)時,粗糙度的貢獻相對減小,響度與尖銳度成為主導因素[1406][1407]。

粗糙度的測量需要高時間分辨率的信號採集,通常要求採樣率至少為 10 kHz,以準確捕捉調製頻率高達 300 Hz 的振幅波動[1408][1409]。此外,粗糙度的計算對信號的時間窗長度敏感,窗長應足夠長以包含至少幾個調製周期(通常為 100 至 200 毫秒),但不宜過長以免平滑掉調製特徵[1410][1411]。

總結而言,粗糙度作為描述噪音快速振幅波動的心理聲學參數,在輪胎噪音品質評估中具有重要意義,特別是在低速與中速行駛場景。透過計算粗糙度並分析其與花紋設計的關係,能夠揭示噪音"顫動"感的來源,從而為花紋優化與調製抑制提供科學依據。降低粗糙度是改善輪胎噪音品質、提升低速舒適性的關鍵途徑[1412][1413]。

7.6.4 波動強度 (Fluctuation Strength)



波動強度(Fluctuation Strength)是描述聲音"緩慢波動"或"拍動"感的心理聲學參數,反映了噪音在調製頻率約 0.5 至 20 Hz 範圍內慢速振幅波動(slow amplitude fluctuation)時人類聽覺的感知強度[1414][1415]。波動強度的單位為 vacil,定義為 1000 Hz 載波、調製頻率 4 Hz、調製深度 100%的振幅調製音在 60 dB SPL 時的波動強度為 1 vacil[1416][1417]。波動強度與粗糙度的本質相同,均源於振幅調製,但兩者的調製頻率範圍不同:粗糙度對應於快速調製(15 至 300 Hz),波動強度對應於慢速調製(0.5 至 20 Hz)[1418][1419]。在調製頻率

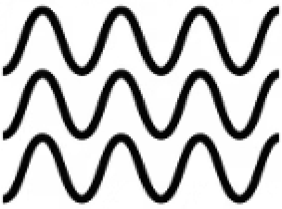
約 4 Hz 時,波動強度達到峰值;在更低頻率(< 0.5 Hz)或更高頻率(> 20 Hz)時,波動強度逐漸減小[1420][1421]。

波動強度的感知基礎與聽覺系統的時間整合特性(temporal integration characteristics)有關[1422][1423]。人類聽覺系統對振幅變化的響應時間約為數百毫秒,當調製頻率在數赫茲時,聽覺系統能夠清晰感知每個調製周期的起伏,從而產生"波動"或"拍動"的主觀感受[1424][1425]。在音樂中,顫音(vibrato,調製頻率約 5 至 7 Hz)正是利用這一頻率範圍的調製來增加音色的豐富性[1426][1427]。然而,在噪音環境中,緩慢的振幅波動往往被評估為"不穩定"或"令人不安",從而增加煩擾度[1428][1429]。

波動強度的計算模型同樣由 Zwicker 與 Fastl 提出,並在 DIN 45692 標準中與粗糙度一同規範[1430][1431]。計算流程與粗糙度類似[1432][1433]:首先,對聲音信號進行臨界頻帶濾波,獲得各 Bark 頻帶的時域包絡;其次,透過低通濾波(截止頻率約 20 Hz)提取包絡的慢變成分,並計算調製深度與調製頻率;隨後,根據調製頻率計算各頻帶的波動強度貢獻,調製頻率在 4 Hz 時波動強度最大[1434][1435]。最後,將所有頻帶的波動強度貢獻求和,得到總波動強度 F(單位 vacil):

$F = \Sigma_i F_i$

其中 F_i 為第 i 個臨界頻帶的波動強度貢獻[1436][1437]。波動強度的計算還需考慮調製深度的非線性效應:當調製深度較小(< 40%)時,波動強度與調製深度近似線性關係;當調製深度較大(> 40%)時,波動強度增長趨於飽和[1438][1439]。



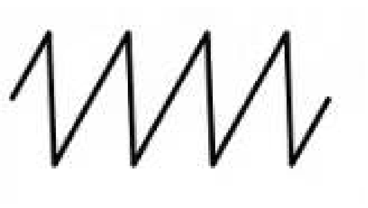
在輪胎噪音評估中,波動強度主要與車速波動、路面不平度以及輪胎不平衡相關[1440][1441]。當車輛在不平整路面上行駛時,輪胎與路面的接觸壓力隨路面起伏而波動,導致噪音振幅出現慢速調製[1442][1443]。若路面波紋(pavement waviness)的波長在 1 至 10 米之間,車速為 50 km/h(約 13.9 m/s)時,對應的激勵頻率為 1.4 至 14 Hz,正好落在波動強度的敏感範圍內[1444][1445]。此外,輪胎不平衡引起的 1 階振動(約 10 至 30 Hz,取決於車速與輪胎半徑)也會對噪音振幅產生調製,從而產生波動強度[1446][1447]。在加速或減速過程中,若車速變化不均勻(如頓挫、抖動),噪音的響度會隨之波動,這種波動若頻率在 0.5 至 20 Hz 範圍內,會被波動強度參數捕捉,並與乘客的"不穩定"感受對應[1448][1449]。

波動強度的影響因素包括調製頻率、調製深度、載波頻率以及調製的規則性(regularity)[1450][1451]。調製頻率在 4 Hz 時波動強度最大,向 0.5 Hz 與 20 Hz 兩端遞減;調製深度越大,波動強度越高,但在高調製深度時增長趨緩[1452][1453]。載波頻率對波動強度的影響與粗糙度類似,中頻段(1000 至 2000 Hz)的載波產生的波動強度高於低頻與高頻載波[1454][1455]。若調製為周期性且規則(如正弦調製),波動強度較高;若調製為非周期性或隨機(如噪音調製),波動強度相對較低[1456][1457]。

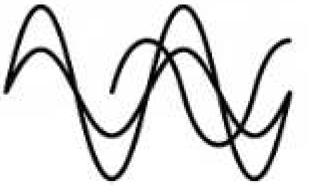
在輪胎設計中,降低波動強度的策略主要針對減少慢速振幅波動的來源[1458][1459]。具體措施包括:首先,改善輪胎的動平衡(dynamic balance),透過精確配重(balancing weights)消除 1 階不平衡,從而減少約 10 至 30 Hz 的振幅調製[1460][1461]。其次,優化胎體結構的均勻性(uniformity),減少徑向力變化(radial force variation, RFV)與側向力變化(lateral force variation, LFV),這些力的波動會導致噪音振幅的慢速調製[1462][1463]。第三,在測試條件選擇上,避免在波紋狀路面(wavy pavement)上進行噪音評估,或在數據處理中濾除由路面不平引起的低頻調製成分[1464][1465]。第四,在車輛動力總成設計中,確保加速與減速的平順性,避免車速的快速波動,從而減少噪音響度的波動[1466][1467]。

波動強度與主觀煩擾度之間的關係在特定場景下較為顯著[1468][1469]。在恆速巡航(cruise driving)中,若輪胎噪音存在明顯的慢速波動,乘客會感知到"不穩定"或"抖動",煩擾度評分會顯著增加[1470][1471]。研究顯示,波動強度每增加 0.1 vacil,煩擾度評分約增加 2%至 4%[1472][1473]。在加速或減速過程中,由於乘客對車速變化有預期,波動強度的影響相對較小[1474][1475]。

波動強度的測量需要足夠長的信號時長(通常至少數秒),以包含多個慢速調製周期[1476][1477]。採樣率要求相對較低(1 kHz 以上即可),但信號的時間穩定性要求較高,應避免外部干擾(如風噪、其他車輛噪音)引入的非系統性波動[1478][1479]。在實際測試中,常採用多次重複測量並取平均值的方法,以提高波動強度估計的可靠性[1480][1481]。



波動強度與粗糙度的結合分析能夠全面描述噪音的時間調製特性[1482][1483]。在粗糙度-波動強度平面(Roughness-Fluctuation Strength Plane)上,可以定位不同噪音樣本的調製特徵:粗糙度高且波動強度低的噪音表現為"快速顫動",粗糙度低且波動強度高的噪音表現為"緩慢拍動",兩者均高則表現為"多時間尺度的不穩定"[1484][1485]。這種二維表示有助於診斷噪音的時間調製來源,並指導時域優化策略[1486][1487]。



總結而言,波動強度作為描述噪音慢速振幅波動的心理聲學參數,在輪胎噪音品質評估中反映了噪音的時間穩定性與乘客對"穩定"感的感知[1488][1489]。透過計算波動強度並分析其與輪胎不平衡、路面不平度以及車速波動的關係,能夠揭示噪音"拍動"或"不穩定"感的來源,從而為動平衡優化、結構均勻性改善以及測試條件控制提供科學依據。降低波動強度是改善輪胎噪音品質、提升恆速行駛舒適性的重要途徑[1490][1491]。

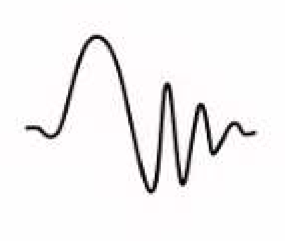
總結與展望

第七章全面而深入地探討了輪胎噪音頻譜分析的理論基礎、技術方法以及實務應用。從聲學基礎理論出發,系統闡述了聲壓與噪音值、頻率與波長、A 加權與其他加權等核心概念,為後續分析奠定了物理與數學基礎。時域分析揭示了輪胎噪音的時間包絡、峰值特徵以及瞬態事件,為動態特性研究提供了直接視角。頻域分析透過 FFT、倍頻程與窄頻分析,將複雜的時域信號轉換為頻譜表示,揭示了能量在頻率軸上的分布與峰值結構,為噪音機制識別與頻譜整形提供了關鍵工具。時頻分析結合短時傅立葉變換、小波變換與階次追蹤,突破了時域與頻域單一視角的局限,能夠同時揭示信號在時間與頻率上的演變,特別適合於分析非穩態的輪胎噪音,為車速變化過程中的噪音特性研究、瞬態事件檢測以及旋轉機械診斷提供了強大技術支撐。

輪胎噪音特徵頻率的識別與理解,將頻譜分析與物理機制緊密聯繫,揭示了花紋激勵、空氣泵浦與共鳴、胎體結構共振、路面紋理激勵以及不平衡與軸承故障等多種噪音源的頻率特徵與能量貢獻,為噪音源診斷與控制設計提供了科學依據。噪音品質評估則從人類聽覺感知的角度出發,透過響度、尖銳度、粗糙度與波動強度等心理聲學參數,建立了噪音物理測量與主觀感受之間的定量關係,為輪胎噪音的"品質"提升與"舒適性"優化提供了理論框架與實務工具。

隨著電動車時代的到來、自動駕駛技術的發展以及消費者對靜音舒適性要求的不斷提升,輪胎噪音頻譜分析的重要性將日益凸顯。未來的研究方向包括:高精度時頻分析算法的開發,如基於稀疏表示、壓縮感知以及深度學習的自適應時頻分解方法;多物理場耦合建模,結合輪胎結構動力學、空氣聲學以及路面交互作用的多尺度仿真,實現頻譜特性的精確預測;智能診斷與預測,利用大數據與人工智能技術,建立輪胎噪音頻譜與磨耗狀態、故障模式之間的映射關係,實現狀態監測與壽命預測;主動噪音控制,透過實時頻譜分析與自適應濾波技術,在車內實現輪胎噪音的主動抑制;以及心理聲學參數的精細化研究,深入探討不同文化背景、年齡群體以及駕駛場景下的主觀感知差異,為個性化噪音品質設計提供依據[1492][1493][1494][1495]。

輪胎噪音頻譜分析作為聯繫物理測量、數學工具與人類感知的橋樑,將持續推動輪胎工業向更低噪音、更高品質、更佳體驗的方向發展,為構建安靜、舒適、永續的交通環境做出重要貢獻[1496][1497]。



結論（Conclusions）

以下是輪胎噪音頻譜分析的詳細說明:

7.1 聲學基礎理論 (Acoustic Basic Theory)

7.1.1 聲壓與聲壓級 (Sound Pressure and Sound Pressure Level)

聲壓 (Sound Pressure) 定義:

- **符號**: p(t)
- **單位**: 帕斯卡 (Pascal, Pa)
- **定義**: 聲波傳播過程中,介質中某點的瞬時壓力相對於靜壓的擾動量

典型聲壓值範圍:

- **正常交談**: 約 0.02 Pa
- **繁忙街道**: 約 0.2 Pa
- **噴氣發動機近場**: 可達 200 Pa
- **人耳聽閾** (最小可聽): 20 μPa (0.00002 Pa)
- **痛閾** (引起疼痛): 200 Pa

動態範圍: 從聽閾到痛閾相差 **10^7 倍** (一千萬倍)

聲壓級 (Sound Pressure Level, SPL) 公式:

$Lp = 20 \log_{10}(prms / p_0)$ dB

其中:

- **prms**: 聲壓均方根值 = $\sqrt{(\langle p^2(t)\rangle T)}$
- **p_0**: 參考聲壓 = **20 μPa** (對應人耳在 1000 Hz 的聽閾)
- **單位**: 分貝 (dB)

聲壓級特性:

- 將巨大動態範圍 (0.00002~200 Pa) 壓縮為 **0~140 dB** 常用範圍
- 聲壓每增加 **10 倍**,SPL 增加 **20 dB**
- 聲壓每增加 **2 倍**,SPL 增加約 **6 dB**

噪音疊加原則 (能量加總):

- 兩個 70 dB 聲源疊加 = **73 dB** (非 140 dB)
- 計算公式: $L_{total} = 10 \log_{10}(10^{(L_1/10)} + 10^{(L_2/10)})$

時間積分等效聲壓級 (Leq):

$Leq = 10 \log_{10}[(1/T)\int_0^T (p^2(t)/p_0^2)\, dt]$ dB

- 反映一段時間內的平均噪音水平
- ISO 13325:2019 規定輪胎滑行噪音測量記錄 **LAmax** 及 **LAeq**

7.1.2 頻率與波長 (Frequency and Wavelength)

基本關係式:

$c = f \times \lambda$

其中:

- **c**: 聲速 (20°C、1 大氣壓乾燥空氣中約 **343 m/s**)
- **f**: 頻率 (Hz)
- **λ**: 波長 (m)

典型波長計算示例:

頻率 (Hz)	波長 (m)	特性
100	3.43	低頻,易穿透與繞射
1,000	0.343 (34.3 cm)	中頻,人耳敏感
10,000	0.0343 (3.43 cm)	高頻,易衰減與吸收

人耳可聽頻率範圍:

- **完整範圍**: 20 Hz ~ 20,000 Hz (20 kHz)
- **年齡衰退**: 中老年人高頻聽力通常衰減至 10-12 kHz
- **最敏感頻段**: 500-4,000 Hz (語言頻率範圍)

輪胎噪音頻率範圍:

頻率範圍	特徵	噪音來源
<100 Hz	低頻,結構共振	輪胎空腔共鳴、車身共振
100-500 Hz	低中頻	低階振動模態、衝擊噪音
500-2,000 Hz	峰值能量區	空氣泵浦、花紋激勵
2,000-5,000 Hz	中高頻	溝槽共鳴、高階振動
>5,000 Hz	高頻,能量較低	微紋理摩擦、空氣湍流

法規管制重點: **500-2,000 Hz** (人耳最敏感,噪音能量最集中)

波長與聲學特性:

- **長波長** (低頻): 易穿透與繞射,難以屏蔽
- **短波長** (高頻): 易衰減與吸收,方向性強

7.1.3 A 加權與其他加權 (A-weighting and Other Weightings)

頻率加權將物理聲壓級轉換為反映主觀感知的加權聲壓級。

A 加權 (A-weighting) - 最廣泛使用

原理:

- 基於 **40 phon 等響曲線**的倒數
- 模擬人耳對不同頻率的敏感度
- 對低頻 (<1 kHz) 和極高頻進行明顯衰減

數學定義 (IEC 61672-1:2013):

$$R_A(f) = (12194^2 \times f^4) / [(f^2 + 20.6^2)(f^2 + 12194^2)\sqrt{((f^2 + 107.7^2)(f^2 + 737.9^2))}]$$

$$W_A(f) = 20 \log_{10}(R_A(f)) + 2.0 \text{ dB}$$

典型衰減值:

頻率 (Hz)	A 加權修正 (dB)
31.5	-39.4
63	-26.2
125	-16.1
250	-8.6
500	-3.2
1,000	**0.0** (參考)
2,000	+1.2
4,000	+1.0
8,000	-1.1
16,000	-6.6

應用:

- **全球環境噪音標準**: dB(A) 或 LA
- **交通噪音法規**: 包括輪胎噪音 (ECE R117)
- **職業噪音暴露**: 工作場所噪音限值

C 加權 (C-weighting)

原理:

- 基於 **100 phon 等響曲線**
- 低頻衰減輕,頻率響應相對平坦
- 適用於高聲級環境

數學定義:

$RC(f) = (12194^2 \times f^2) / [(f^2 + 20.6^2)(f^2 + 12194^2)]$

$WC(f) = 20 \log_{10}(RC(f)) + 0.06$ dB

應用:

- 工業高噪音環境
- 機場噪音監測
- 爆炸與衝擊噪音
- 低頻噪音評估

Z 加權 (Z-weighting) - 線性/無加權

特性:

- 頻率響應在整個測量範圍內保持 **0 dB** (±1.5 dB 容差)
- 不進行任何頻率修正

應用:

- 物理機理研究
- 頻譜詳細分析
- 保留原始信號特性

B 加權 (B-weighting) - 已很少使用

原理:

- 基於 **70 phon 等響曲線**
- 現代標準中極少使用

D 加權 (D-weighting) - 航空噪音專用

特性:

- 專為航空噪音設計
- 對 **2,000-10,000 Hz** 高頻有增強
- 反映航空發動機噪音特性

加權選擇指南:

加權類型	適用場景	典型應用
A 加權	環境噪音 (40-85 dB)	交通、輪胎、環境法規
C 加權	高聲級 (85-130 dB)	工業、機場、低頻評估
Z 加權	物理分析	研究、頻譜分析
D 加權	航空噪音	飛機噪音認證

7.2 時域分析 (Time Domain Analysis)

時域分析是聲學信號處理的最直接方法,直接觀察聲壓隨時間的變化。

時域波形 (Time Waveform)

定義: 聲壓 $p(t)$ 隨時間 t 的完整變化曲線

主要觀察特徵:

- **穩態 vs 瞬態**: 判斷噪音是連續還是間歇性
- **週期性 vs 隨機性**: 識別規律性激勵
- **幅值變化**: 檢測異常峰值與波動

時域統計參數

1. 均方根聲壓 (RMS)

公式:

$$prms = \sqrt{[(1/T)\int_0^T p^2(t)\,dt]}$$

離散形式:

$$prms = \sqrt{[(1/N)\Sigma_{i=1}^{N} p_i^2]}$$

意義: 反映信號的能量大小,是計算 SPL 的基礎

2. 峰值聲壓 (Peak)

公式:

$$ppeak = \max|p(t)|$$

意義: 反映最大瞬時幅值,用於評估衝擊性噪音

3. 峰值因數 (Crest Factor)

公式:

$$CF = ppeak / prms$$

典型值:

- **正弦波**: $CF = \sqrt{2} \approx 1.414$

- **高斯白噪音**: CF ≈ 3-4
- **衝擊性噪音**: CF > 5

應用: 描述動態範圍與衝擊性,高 CF 表示信號含強烈瞬態

4. 峭度 (Kurtosis)

公式:

$\text{Kurtosis} = E[(p - \mu)^4] / (E[(p - \mu)^2])^2$

典型值:

- **高斯分布**: Kurtosis = 3
- **均勻分布**: Kurtosis < 3
- **衝擊性信號**: Kurtosis > 3

應用: 檢測衝擊性異常,超峭度 (Kurtosis > 5) 表示強烈瞬態事件

5. 偏度 (Skewness)

公式:

$\text{Skewness} = E[(p - \mu)^3] / (E[(p - \mu)^2])^{(3/2)}$

意義:

- **Skewness = 0**: 對稱分布
- **Skewness > 0**: 右偏 (正值多)
- **Skewness < 0**: 左偏 (負值多)

應用: 判斷分布對稱性,識別單向激勵

其他時域分析方法

6. 自相關/互相關分析

- 揭示信號週期性結構
- 多通道信號延遲關係
- 識別隱藏的週期成分

7. 包絡分析 (Envelope)

- 提取幅度調製特徵
- 分析花紋週期調製
- 檢測軸承故障特徵頻率

工程應用

1. 設計評估:

- 降低衝擊性 (降低峰值因數)

- 降低峭度 (減少瞬態事件)

2. 產線質控:

- RMS 偏差檢測
- 峰值異常報警

3. 異常檢測:

- 峭度監控軸承/輪胎缺陷
- 包絡分析識別調製異常

4. 聲源分辨:

- 互相關確定聲源位置
- 時間延遲估計

7.3 頻域分析 (Frequency Domain Analysis)

頻域分析將時域信號轉換為頻率成分,是噪音分析的核心方法。

7.3.1 FFT 分析 (FFT Analysis)

快速傅立葉變換 (Fast Fourier Transform) 是離散傅立葉變換 (DFT) 的高效算法實現。

數學原理

離散傅立葉變換定義:

$X[k] = \Sigma_{n=0}$^(N-1) x[n] exp(-j2πkn/N)

其中:

- **x[n]**: 時域信號 (n = 0, 1, …, N-1)
- **X[k]**: 頻域信號 (k = 0, 1, …, N-1)
- **N**: 採樣點數

頻率索引對應物理頻率:

$f_k = k \times fs / N$

頻率解析度:

$\Delta f = fs / N$

其中:

- **fs**: 採樣率 (Hz)
- **N**: 採樣點數

示例:

- 採樣率 fs = 25,600 Hz

- 採樣點數 N = 2,560
- 頻率解析度 Δf = 10 Hz
- 最高分析頻率 = fs/2 = 12,800 Hz (Nyquist 頻率)

窗函數 (Window Function)

窗函數用於減少頻譜洩漏,不同窗函數有不同特性。

常用窗函數:

1. 矩形窗 (Rectangular)

w[n] = 1

- **特點**: 不加權,頻率解析度最好
- **缺點**: 頻譜洩漏最嚴重
- **應用**: 瞬態信號、已知週期信號

2. 漢寧窗 (Hanning)

$w[n] = 0.5(1 - \cos(2\pi n/(N-1)))$

- **特點**: 平衡解析度與洩漏抑制
- **應用**: 通用,最常用

3. 漢明窗 (Hamming)

$w[n] = 0.54 - 0.46\cos(2\pi n/(N-1))$

- **特點**: 比 Hanning 更強的旁瓣抑制
- **應用**: 語音分析

4. 平頂窗 (Flat-top)

- **特點**: 幅值測量精度最高
- **缺點**: 頻率解析度較低
- **應用**: 校準、幅值精確測量

頻譜洩漏效應 (Spectral Leakage)

原因:

- 信號在窗邊界處不連續
- 本應集中在某頻率的能量"洩漏"到鄰近頻率
- 形成旁瓣 (side lobes),掩蔽真實頻譜結構

解決方法:

1. **選擇適當窗函數**: Hanning/Hamming 可大幅降低洩漏
2. **零填充 (Zero Padding)**: 提升顯示解析度 (但不增加真實分辨力)

3. **頻譜平均 (Welch 方法)**: 多分段平均降低隨機誤差

7.3.2 倍頻程分析 (Octave Band Analysis)

倍頻程分析是聲學領域標準化的頻帶分析方法。

1/1 倍頻程 (Octave Band)

定義: 上限頻率是下限頻率的 **2 倍**

$fu = 2 \times fl$

中心頻率:

$fc = \sqrt{(fl \times fu)} = fl \times \sqrt{2}$

頻帶寬度:

$\Delta f = fu - fl = fl \approx 0.707 \times fc$

標準中心頻率序列 (基於 1000 Hz):

31.5, 63, **125**, **250**, **500**, **1000**, **2000**, **4000**, 8000, 16000 Hz

1/3 倍頻程 (One-Third Octave Band)

定義: 每個倍頻程劃分為 **3 個** 頻帶

頻率比: 相鄰中心頻率相差 **$2^{(1/3)} \approx 1.26$ 倍**

頻帶寬度:

$\Delta f \approx 0.23 \times fc$

標準中心頻率 (部分):

100, 125, 160, 200, 250, 315, 400, 500, 630, 800, 1000, 1250, 1600, 2000, 2500, 3150, 4000, 5000 Hz …

共 31 個常用頻帶 (20 Hz - 20 kHz 範圍)

應用場景

1. 法規報告:

- ECE R117 等法規通常要求 **1/3 倍頻程**數據
- 提供標準化、可比較的頻譜信息

2. 材料特性評估:

- 吸音材料吸收係數 vs 頻率
- 隔音材料傳輸損失 vs 頻率

3. 交通噪音預測:

- 道路交通模型輸入
- 噪音地圖計算

4. 噪音降低量 (NR) 分析:

- 評估降噪措施的頻率選擇性效果

7.3.3 窄頻分析 (Narrowband Analysis)

定義: 採用精細頻率解析度 (通常為數 Hz 或更小) 的頻譜分析

特點

高頻率解析度:

- Δf = 1-10 Hz (甚至更小)
- 能夠分辨頻率接近的離散成分

長時間窗:

- 需要更長的數據段
- 適合穩態、連續信號

應用場景

1. 花紋節距頻率識別:

- 精確定位基頻及其諧波
- 例如: 80 km/h, R=0.3m, N=60 → fp ≈ 707 Hz

2. 音調性 (Tonal) 噪音檢測:

- 計算音調可聽度比 (Tone-to-Noise Ratio, TNR)
- 符合 ISO 1996-2:2017 與 DIN 45681 標準

3. 共振峰識別:

- 空腔共鳴頻率 (200-250 Hz)
- 溝槽共鳴頻率 (1000-4000 Hz)
- 結構模態頻率 (50-1000 Hz)

4. 階次分析基礎:

- 提取與轉速同步的頻率成分
- 結合 Campbell 圖追蹤峰值隨速度變化

局限性

- **大數據量**: 需要長時間記錄
- **穩態要求**: 適合連續、恆速測試
- **信噪比要求**: 對測試環境要求高

7.4 時頻分析 (Time-Frequency Analysis)

時頻分析同時提供時間和頻率信息,適合非穩態信號。

7.4.1 短時傅立葉變換 (Short-Time Fourier Transform, STFT)

核心思想: 將非穩態信號分割為一系列短時段,對每個時段進行傅立葉變換。

數學定義

STFT_x(t, f) = ∫_{-∞}^{+∞} x(τ) w(τ - t) e^{-j2πfτ} dτ

其中:

- **x(τ)**: 原始信號
- **w(τ - t)**: 窗函數,中心在時刻 t
- **t**: 時間變量
- **f**: 頻率變量

時頻解析度權衡

海森堡不確定性原理:

$\Delta t \cdot \Delta f \geq 1/(4\pi)$

權衡:

- **窄窗** (Δt 小):
    - 時間解析度高
    - 頻率解析度低
    - 適合瞬態事件定位
- **寬窗** (Δt 大):
    - 頻率解析度高
    - 時間解析度低
    - 適合穩態頻譜分析

實務參數設置

採樣率: 48 kHz (典型)

窗長選擇:

- **1024 點**: 約 21 ms,高時間解析度
- **2048 點**: 約 43 ms,高頻率解析度
- **4096 點**: 約 85 ms,最高頻率解析度

重疊比例:

- **50% 重疊**: 平衡計算量與平滑度
- **75% 重疊**: 更高時間解析度,計算量增加

窗函數: Hanning 或 Hamming (最常用)

應用

- 噪音瞬態事件偵測
- 非平穩過程分析
- 車速變化下噪音演變
- 時頻圖 (Spectrogram) / 瀑布圖 (Waterfall Plot) 可視化

7.4.2 小波分析 (Wavelet Analysis)

小波分析通過多尺度分析提供比 STFT 更靈活的時頻表示。

連續小波變換 (CWT)

數學定義:

$$CWT_x(a, b) = (1/\sqrt{a}) \int_{-\infty}^{+\infty} x(t)\, \psi^*((t - b)/a)\, dt$$

其中:

- **a**: 尺度參數 (scale, a > 0),對應頻率
- **b**: 平移參數 (translation),對應時間
- **ψ(t)**: 母小波函數
- **ψ***: 母小波的共軛

母小波選擇

1. Morlet 小波:

- **定義**: 高斯包絡調製的複正弦波
- **特點**: 適合分析調製信號,良好的時頻局部化
- **應用**: 花紋節距調製、共鳴追蹤

2. Daubechies 小波 (dbN):

- **特點**: 正交性強,支持長度可調
- **應用**: 信號分解與壓縮

3. Symlet 小波:

- **特點**: 近似對稱,低延遲
- **應用**: 特徵提取

4. 墨西哥帽小波 (Mexican Hat):

- **特點**: 二階導數形式
- **應用**: 邊緣檢測、突變點識別

離散小波變換 (DWT)

- **Dyadic 採樣**: 尺度以 2 的冪次增長

- **多層分解**: 將信號分解為不同尺度的細節 (Detail) 與逼近 (Approximation)
- **計算效率**: 遠高於 CWT

小波包變換 **(WPT):**

- 實現頻譜均勻或自適應細分
- 增強窄頻特徵分辨

優勢

1. 多分辨率:

- 高頻: 高時間解析度
- 低頻: 高頻率解析度
- 自動適應信號特性

2. 適應性強:

- 捕捉多尺度特徵
- 突變檢測
- 周期與非穩態並存

3. 計算效率:

- DWT 計算速度快
- 適合實時處理

應用

- 噪音信號分離 (去噪)
- 瞬態檢測與定位
- 自動特徵提取
- 多尺度噪音源識別

7.4.3 階次追蹤 (Order Tracking)

階次追蹤是專門針對旋轉機械在轉速時變條件下的信號處理技術。

核心概念

階次 (Order) 定義:

階次 = 振動頻率 / 轉速頻率

示例:

- **1 階**: 與轉速同步 (輪胎不平衡)
- **N 階**: 轉速的 N 倍 (花紋節距數 = N)

方法

1. 計算階次追蹤 (Computed Order Tracking, COT):

步驟:

1. 用轉速信號對振動/噪音信號進行 **角域重採樣**
2. 將非穩態時域信號轉為 **角域信號** (每角度等間隔)
3. 對角域信號進行傅立葉變換
4. 得到階次譜 (Order Spectrum)

優勢: 消除轉速波動影響,聚焦週期性成分

2. Vold-Kalman 濾波階次追蹤 (VKF-OT):

- 用狀態空間模型與卡爾曼濾波
- 直接在時域提取各階次瞬時幅值與相位
- 適合多階次同時追蹤

與 FFT 差異

特性	**FFT 分析**	**階次追蹤**
轉速要求	假設穩態	適用時變轉速
頻率漂移	頻譜線拖尾模糊	階次線清晰聚焦
週期成分	難以分離	精確提取
應用	恆速測試	加速/減速測試

應用

1. 輪胎不平衡診斷:

- 追蹤 **1 階** (轉速同步)
- 識別靜/動不平衡

2. 花紋週期性分析:

- 追蹤 **N 階** (節距數階)
- 評估花紋設計效果

3. 軸承故障診斷:

- 提取故障特徵頻率階次

4. 車速變化下噪音分析:

- Campbell 圖: 階次 vs 轉速
- 識別共振交叉點

7.5 輪胎噪音特徵頻率 (Characteristic Frequencies of Tire Noise)

特徵頻率是輪胎噪音頻譜中由特定物理機制產生的峰值或能量集中頻率。

1. 花紋節距頻率 (Tread Pattern Excitation Frequencies)

基本公式

單一節距長度:

$p = 2\pi R / N$

其中:

- **R**: 輪胎半徑 (m)
- **N**: 圓周上節距總數

花紋節距頻率:

$f_p = v / p = (v \cdot N) / (2\pi R)$

其中:

- **v**: 車速 (m/s)
- **fp**: 節距頻率 (Hz)

階次分析中: 花紋節距頻率對應 **N 階** (N-th order)

計算示例

條件:

- 車速 v = 80 km/h = 22.22 m/s
- 輪胎半徑 R = 0.3 m
- 節距數 N = 60

計算:

$f_p = (22.22 \times 60) / (2\pi \times 0.3)$

$= 1333.2 / 1.885$

≈ 707 Hz

諧波頻率: 2fp ≈ 1414 Hz, 3fp ≈ 2121 Hz, …

2. 空氣共鳴頻率 (Air Resonance Frequencies)

溝槽共鳴 (Groove Resonance)

公式:

$f_{res} = (n \cdot c) / (2L)$

其中:

- **n**: 模態階數 (1, 2, 3, …)
- **c**: 空氣中聲速 (343 m/s)
- **L**: 溝槽長度 (m)

典型範圍:

- 橫向溝槽 (L ≈ 0.1-0.15 m): **1000-4000 Hz**
- 縱向溝槽 (L ≈ 0.3-0.5 m): **300-800 Hz**

輪胎空腔共鳴 (Tire Cavity Resonance)

頻率範圍: 200-250 Hz

特徵:

- 車內轟鳴聲 (boom)
- 低頻共振
- 與輪胎內容積相關

楔形空腔共鳴 (Wedge Cavity Resonance)

頻率範圍: 800-1500 Hz

特徵:

- 輪胎與路面間楔形空腔
- 中高頻峰值
- 與接觸區幾何相關

3. 結構共振頻率 (Structural Resonance Frequencies)

低階徑向模態: 50-100 Hz

- 輪胎整體徑向振動

高階徑向模態: 100-400 Hz

- 輪胎周向多波節振動

切向/側向模態: 100-1000 Hz

- 輪胎側壁彎曲、扭轉

識別方法

1. 頻域分析 (FFT、倍頻程):

- 識別明顯峰值頻率

2. 時頻分析 (STFT、小波):

- 追蹤峰值隨速度變化

3. 階次追蹤:

- 確定峰值是否隨轉速等比例移動
- 區分花紋激勵 (同步) vs 共鳴 (固定頻率)

4. 對比實驗:

- 不同花紋設計對比

- 不同路面對比
- 不同速度對比

7.6 噪音品質評估 (Sound Quality Evaluation)

噪音品質評估超越簡單的聲壓級,評估聲音的主觀感知特性。

7.6.1 響度 (Loudness)

響度反映聲音的"大小"主觀感受。

單位

- **sone**: 響度單位
- **定義**: 1000 Hz、40 dB SPL 純音在雙耳聆聽條件下的響度 = **1 sone**
- **phon**: 響度級單位,等同於等效的 1000 Hz 純音聲壓級 (dB SPL)

Zwicker 響度模型 (ISO 532-1:2017)

步驟:

1. **1/3 倍頻程濾波**: 將信號分解為頻帶
2. **等響曲線轉換**: 根據等響曲線得到響度級 (phon)
3. **Bark 尺度轉換**: 轉為臨界頻帶尺度 (0-24 Bark)
4. **比響度計算**: 計算 $N'(z)$ (sone/Bark),考慮頻率掩蔽效應
5. **總響度積分**:

$$N = \int_0^{24\ \text{Bark}} N'(z)\, dz$$

其中:

- **N**: 總響度 (sone)
- **$N'(z)$**: 比響度 (sone/Bark)
- **z**: Bark 尺度

Stevens 冪律 (Stevens’ Power Law)

公式:

$$N \propto (I / I_0)^{0.3}$$

意義:

- 聲強每增加 **10 dB**,響度約 **加倍**
- 響度與聲強的 **0.3 次方** 成正比

Moore-Glasberg 模型 (ISO 532-2:2017)

- 更精細的聽覺濾波模擬
- 適合複雜聲場

- 提供更準確的雙耳響度預測

工程應用

- **設計目標**: 降低總響度 (sone)
- **頻譜優化**: 降低高比響度頻帶的能量
- **主觀評價對比**: 響度與 dB(A) 的差異分析

7.6.2 尖銳度 (Sharpness)

尖銳度描述聲音"尖銳"或"刺耳"程度。

單位與定義

- **單位**: acum
- **定義**: 1000 Hz、60 dB SPL 窄帶噪音 = **1 acum**

計算公式 (Zwicker & Fastl)

$S = 0.11 \times (\int_0^{24\ \mathrm{Bark}} N'(z)\, g(z)\, z\, dz) / N$

其中:

- **S**: 尖銳度 (acum)
- **N’(z)**: 比響度 (sone/Bark)
- **g(z)**: 加權函數 (高頻增強)
- **z**: Bark 尺度
- **N**: 總響度 (sone)

影響因素

1. 頻譜重心:

- 重心越高,尖銳度越大
- 高頻能量比例直接影響

2. 帶寬:

- 窄帶高頻 > 寬帶高頻
- 純音比噪音更尖銳

3. 總響度:

- 響度增加,尖銳度感受也增強

工程控制

降低尖銳度方法:

1. **抑制高頻能量** (>2000 Hz)
2. **展寬共鳴帶寬** (避免窄帶峰值)

3. **頻譜整形** (平滑高頻滾降)
4. **增加低頻成分** (降低頻譜重心)

7.6.3 粗糙度 (Roughness)

粗糙度描述聲音"粗糙"或"顫動"感。

單位與定義

- **單位**: asper
- **定義**: 1000 Hz 載波、70 Hz 調製、調製深度 100%、60 dB SPL = **1 asper**

物理基礎

- 反映 **15-300 Hz** 調製頻率範圍內的快速振幅波動
- **70 Hz** 調製最敏感 (最大粗糙度)

計算方法 (Zwicker, Fastl, DIN 45692)

$R = \Sigma_i R_i$

其中:

- **R**: 總粗糙度 (asper)
- **R_i**: 第 i 個臨界頻帶的粗糙度貢獻

計算步驟:

1. 分析各 Bark 臨界頻帶的包絡
2. 提取包絡的調製頻率與深度
3. 計算各頻帶粗糙度貢獻
4. 加總得到總粗糙度

影響因素

1. 調製頻率:

- **15-300 Hz** 範圍有效
- **70 Hz** 附近最大

2. 調製深度:

- 深度越大,粗糙度越大
- 100% 調製 > 50% 調製

3. 載波頻率:

- **1000-2000 Hz** 最易產生粗糙感

工程控制

降低粗糙度方法:

1. **變節距設計**: 抑制單一週期激勵
2. **增加阻尼**: 減小包絡調製深度
3. **頻譜擴散**: 避免窄帶調製

7.6.4 波動強度 (Fluctuation Strength)

波動強度描述聲音"緩慢波動"或"拍動"感。

單位與定義

- **單位**: vacil
- **定義**: 1000 Hz 載波、4 Hz 調製、調製深度 100%、60 dB SPL = **1 vacil**

物理基礎

- 反映 **0.5-20 Hz** 調製頻率範圍內的慢速振幅波動
- **4 Hz** 調製最敏感 (最大波動強度)

計算方法

$F = \Sigma_i F_i$

其中:

- **F**: 總波動強度 (vacil)
- **F_i**: 第 i 個臨界頻帶的波動強度貢獻

與粗糙度的差異

參數	粗糙度 (Roughness)	波動強度 (Fluctuation)
調製頻率	15-300 Hz (快速)	0.5-20 Hz (緩慢)
最敏感點	70 Hz	4 Hz
主觀感受	顫動、粗糙	拍動、波動
典型源	花紋週期、高頻調製	不平衡、車速波動

工程應用

評估場景:

- 輪胎不平衡診斷
- 車速波動影響
- 路面起伏引起的噪音變化

控制方法:

- 優化動平衡
- 改善懸掛系統阻尼
- 路面平整度提升

總結

輪胎噪音頻譜分析是一個多層次、多維度的系統工程:

基礎層面:

- 聲學基本理論提供物理與感知基礎
- A 加權連接物理量與主觀感受

分析方法:

- **時域**: 直觀、快速,適合瞬態與統計
- **頻域**: 經典、標準,FFT/倍頻程為核心
- **時頻**: 動態、全面,STFT/小波/階次追蹤為高級工具

特徵識別:

- 花紋節距頻率: 設計的直接體現
- 共鳴頻率: 結構與幾何的固有特性
- 多方法聯合識別: 確保準確性

品質評估:

- 超越 dB(A): 響度、尖銳度、粗糙度、波動強度
- 多維度評價: 全面反映主觀感受
- 工程優化: 針對性改進設計

通過這些方法的綜合應用,可以實現從測量、分析到優化的完整輪胎噪音控制流程。

研究顯示，輪胎噪音的頻譜特性與其物理來源之間存在高度對應關係。低頻成分多與輪胎結構振動與空腔共鳴相關，中頻能量主要來自胎面花紋激勵與輪胎—路面接觸行為，而高頻成分則與路面微觀粗糙度、花紋細節及空氣動力效應密切相關。透過頻譜峰值、音調性與調製特徵的識別，工程師得以精準定位降噪設計的優先順序，而非僅依賴整體噪音值的微幅變化。

本章亦凸顯心理聲學觀點在輪胎噪音評估中的關鍵角色。大量實證顯示，人類對噪音的煩擾感並非單純取決於能量大小，而深受頻譜形狀與時間結構影響。即便總噪音值相同，含有明顯音調成分或高頻尖銳特徵的輪胎噪音，往往引起更強烈的不適感。因此，將響度、尖銳度與粗糙度等心理聲學指標納入頻譜分析流程，有助於從「降低分貝」進一步邁向「改善噪音品質」的設計思維。

從工程與產業應用角度觀之，輪胎噪音頻譜分析已成為產品研發、型式認證輔助判讀、品質監控與售後問題診斷的重要工具。其價值不僅在於提升單一產品的聲

學表現，更在於建立可重複、可比較的聲學資料基礎，支撐長期技術演進。展望未來，隨著電動車普及、道路鋪面多樣化及智慧聲學量測技術的成熟，頻譜分析將進一步與聲源定位、數值模擬與資料驅動方法整合，成為低噪音輪胎設計與交通噪音治理不可或缺的核心技術。本章所建構的頻譜分析架構，為後續輪胎噪音機制研究、心理聲學評估與法規技術發展，奠定了堅實而前瞻的理論與實務基礎。

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