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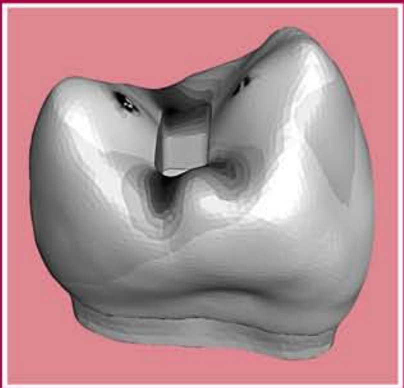

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Dental biomaterials

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Imaging, testing and modelling

Edited by
Richard V. Curtis and Timothy F. Watson

Woodhead Publishing and Maney Publishing
on behalf of
The Institute of Materials, Minerals & Mining

CRC Press
Boca Raton Boston New York Washington, DC

WOODHEAD PUBLISHING LIMITED
Cambridge England

Woodhead Publishing Limited and Maney Publishing Limited on behalf of
The Institute of Materials, Minerals & Mining

Woodhead Publishing Limited, Abington Hall, Abington
Cambridge CB21 6AH, England
www.woodheadpublishing.com

Published in North America by CRC Press LLC, 6000 Broken Sound Parkway, NW,
Suite 300, Boca Raton, FL 33487, USA

First published 2008, Woodhead Publishing Limited and CRC Press LLC

British Library Cataloguing in Publication Data
A catalogue record for this book is available from the British Library.

Library of Congress Cataloging in Publication Data
A catalog record for this book is available from the Library of Congress.

Woodhead Publishing ISBN 978-1-84569-296-4 (book)
Wookhead Publishing ISBN 978-1-84569-424-1 (e-book)
CRC Press ISBN 978-1-4200-7209-9
CRC Press order number: WP7209

The publishers' policy is to use permanent paper from mills that operate a sustainable forestry policy, and which has been manufactured from pulp which is processed using acid-free and elementary chlorine-free practices. Furthermore, the publishers ensure that the text paper and cover board used have met acceptable environmental accreditation standards.

Typeset by SNP Best-set Typesetter Ltd., Hong Kong
Printed by TJ International Limited, Padstow, Cornwall, England

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Preface

The dental and biomaterials science literature is full of excellent research papers that give mention to many exciting techniques for the determination of physical properties, chemical constituents and structural organisation of biomaterials and their biological substrates. Equally, there are review papers that look at materials from the clinical perspective without probing the materials and biological background in depth. However, there are very few sources of information available to the researcher who is setting out in a new field of study, who wants to be updated in a particular area of newly developing technology or who wants to give their postgraduate students a source of primary information for their studies of the literature. *Dental biomaterials: imaging, testing and modelling* may help to redress this balance.

When assembling the list of authors for this book we aimed to incorporate individuals who are the leaders in their respective fields in dental biomaterials. The authors are from many different backgrounds and elegantly illustrate the profitable results of collaboration between the laboratory and the clinic. There is no doubt that there are many areas of endeavour that have not been covered in this present edition, but a book can only be a finite size. We hope that the collection of chapters covers a suitably wide-ranging suite of topics and that it may encourage individual researchers to look to other, parallel, technologies to their own and so advance the exciting world of dental biomaterials science.

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1
Characterizing the performance of dental air-turbine handpieces

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1.1 Outline

After briefly outlining the general importance of air-turbine handpieces in dentistry (Section 1.2), a historical account of their development puts their present status into context (Section 1.3). However, in order to understand performance in general, it is necessary to recognize the large number of factors involved, and their complex interactions (Section 1.4). In essence, it is not yet possible to characterize the cutting performance of these devices, dependent as they are on the behaviour of cutter and substrate, amongst other things. Accordingly, it is as yet only feasible to document the physical aspects of the behaviour of the turbine itself (Section 1.5), but this leads to a number of figures of merit that may be used for product comparisons in an objective fashion that are tied to the physics of these machines. Even so, because of their internal complexity, primarily in terms of gas flow, it is necessary to resort to the 'black-box' approach and document input–output relationships, subsuming much unresolvable detail in some fitted parameters. Selection and application by the end-user nevertheless depends on a number of further issues of great importance, and these are discussed under the general heading of hazards (Section 1.6). The chapter closes with some general remarks on selection, usage, and areas where further study is essential.

1.2 General importance: applications, benefit

The dental air-turbine handpiece rapidly gained widespread acceptance by the dental profession after its introduction in the late 1950s, and it continues to be used as the main means of carrying out cutting work in clinical dental practice, whether of tooth tissue or restorative materials. In comparison with alternatives at the time, the reasons given for its usefulness included the following.

- Power: power-to-weight ratio very favourable, negligible transmission loss;

- Size: size and weight allow better control for long periods without tiring as well as good intraoral access;
- Speed: reduction of unpleasant vibration, finer control of cutting process;
- Effort: lower forces could be used yet with higher removal rates.

These considerations still appear to be pertinent.

1.3 Historical outline: development, features

A turbine is a motor in which a shaft is steadily rotated by the action of a current of fluid upon the blades of a wheel. Turbines powered by various fluids have evolved along several paths, and it is not possible to identify a single source for the development of dental systems.

The first air-powered dental engine design was patented in 1868[1] although in fact this was not a turbine but effectively a lobe pump operated in reverse. It was intended to be operated by mouth, foot bellows, or a compressed air vessel. The first true turbine dental handpiece, with a 13-bladed rotor, was patented in 1874,[2] with similar suggestions for operation as the lobe pump. It received little attention from the profession. A water-powered device in 1877[3] also made provision for a fine stream of water to be directed as coolant onto the cutting instrument. A more elaborate device with a transmission clutch, a rotatable handpiece sheath, and revised mechanism for the attachment of cutting instruments followed in 1879,[4] although details of the turbine rotor were not given and the drive fluid was not specified.

These machines were all somewhat bulky with their weight borne by the dentist's hand. However, a water-powered engine, produced by S. S. White in 1881[5] avoided this problem by the motor being mounted on a floor stand. A flexible shaft transmitted the drive in a fashion similar to that of many foot-treadle engines of the time. Evidently, the problems were greater than the advantages. Improvements made to foot-treadle and electric dental engines in the late 1800s led to fluid-driven devices falling by the wayside and, by the 1920s, the cord-arm drive had been adopted as the *de facto* standard means of transmission from an electric motor to the handpiece.[6]

In the 1870s, the maximum speeds were around 700 rpm (12/s) and 1000 rpm (17/s) for foot-driven and electrical devices, respectively.[7] Speed, recognized to be beneficial, progressively increased, but success depended in part on suitable rotary cutting instruments being available, as well as improved means of cooling the cutting site. An electric engine of 1911 reputedly achieved up to 10 000 rpm (167/s);[6,8–10] separating discs and grinding tools worked more smoothly with less patient discomfort. However, the engine was unsuccessful because of overheating and seizure of the handpiece bearings.[6,9] Effective means of achieving such speeds did not become available until the 1940s.

Studies of vibration perception provided evidence in favour of increased speeds.[11–13] The upper frequency threshold of vibration perception was found to be ~650 Hz, with maximum unpleasantness in the range 100–200 Hz. Using burs, stones, and diamond instruments at 3000–4000 rpm (50–67 /s), the vibrations produced were ~110–150 Hz. An air-turbine handpiece was then developed (see below) specifically to produce vibrations above the limit of perception by virtue of its high rotation rate. In the end, it was concluded that, in procedures that resulted in the same range of temperature rise, high-speed devices could remove enamel some three times as fast and at 1/30th of the operating load, as well as with better control and less effort.[14] Indeed, with proper cutting-site cooling, high-speed rotation was not only possible but practical, safe, and effective,[15,16] with advantages for both patient and operator.[17] In fact, it was said that 'few pieces of equipment in dentistry have caused more changes and improved dental service to a greater degree than ultra-high-speed handpieces (i.e. those rotating at 1000–5000 /s)',[18] allowing improved patient response, shortened operating time, reduced vibration perception, and less patient and operator fatigue. For these reasons this development was described as 'one of the most significant contributions to dental health service'.[19]

Nevertheless, high-rotation-rate cutting only became possible when instruments became available that could withstand such speeds. Until at least 1870, steel burs were the only cutting instruments available and these were individually shaped and finished by hand. The mass production of carbon steel burs began by the 1870s.[8] Corundum (Al_2O_3) separating discs and stones were introduced in 1872[6,8,9] and provided the first satisfactory means of cutting enamel, although subsequently supplanted by carborundum (SiC).

Diamond grit cutting instruments were first advertised in 1878[20] but, being on a soft copper core, could not be used at high speed until the development in 1932 of galvanized bonding to harder alloy.[6,8,9] Tungsten carbide burs followed in 1948 and proved to be extremely successful in high-speed applications.[8] There has been no significant development since then.

The problem of heat generation remained, although recognized as much as 2000 years ago[21] for surgical trephines. A cooling system was fitted to one handpiece in commercial production by 1874:[22] water was applied from a rubber bulb though a hose and nozzle, but an integral system soon after was to apply a stream of water onto the cutting instrument.[3] Many patented designs followed,[23–38] some of which allowed compressed air to be applied to the cavity for debris removal. One even heated the air and water to minimize patient discomfort, and they could be released simultaneously. Alongside this, more effective aspiration was required.[39,40] Hollow burs, through which air is passed to supplement the cooling provided by air and water jets, were devised in 1974[19,41] but failed to gain widespread use, despite a similar principle being used in surgical instruments.

The start of the modern turbine era may be 1941, when a patent claimed 25 000 rpm (417/s) for a design using compressed air at 45 psi (310 kPa).[42] The turbine rotor was unusual: a cylinder with a circular arrangement of holes through which air jets from two nozzles were directed. In addition, ball bearings were to be used (as opposed to the sleeve bearings of earlier handpieces), and the inner ball race of the bearing at the chuck was arranged to cause the jaws to open and close by a sliding action, thus facilitating the rapid change of instruments.

The first demonstrations of Norlén's device in London, UK were in May 1958,[43] although it had been patented in 1952.[44] Turbine rotation was transmitted to the instrument via a mechanism in the body of the handpiece, which was interchangeable by a slip-joint connection. Multiple nozzles directed air onto inclined, slightly shovel-shaped turbine blades. Speed control was by means of adjusting the opening of vent holes. Said to reach a rotor speed of 120 000 rpm (2000/s), a ball-race reduction drive gave an instrument speed of 50 000 rpm (833/s). A finger-operated spring released a brake and progressively opened the air inlet.

There has been much debate over the history of the first handpiece with a turbine in the head (see below),[43,45] a distinct advance in design first developed in New Zealand. Arising from the vibration studies mentioned above, as an adaptation of a commercial air-powered drill, it reached 60 000 rpm (1000/s), but was very noisy and exhausted excessive air into the patient's mouth. As no suitable bearings were available, overheating and seizure occurred after a short period of use.[46] The project was abandoned.

A water-driven, head-mounted turbine contra-angle handpiece was reported in 1953.[47] The small rotor (diameter, 7.5 mm; height, 4.8 mm) had six notched blades and could rotate at 61 000 rpm (1017/s). Hollow-shank cutting instruments were fitted directly to the rotor shaft, retained by a spring and keyway. Although referring to a number of earlier turbine designs,[2–4,42,44] the New Zealand design seems to have been unknown to the authors.[48] Further development had a complicated history,[43,45] but by 1956 it was being sold as both straight and contra-angle handpieces and could operate at up to 100 000 rpm (1667/s).[7] The drive water was recirculated by coaxial tubing to the pump, and threaded rotary instruments were attached to the turbine shaft. This was said to be 'extremely quiet during operation' and to have 'the highest torque of any turbine angle handpiece'.[7]

Meanwhile, development of cord-driven equipment continued, although speed was limited by the difficulties of transmission. However, in 1951, a 10 000 rpm (167/s) device using an 'accelerating wrist joint' was produced.[6]

Then, in 1955, with the elimination of gears, the Page–Chayes handpiece reached 100 000 rpm (1667/s), relying on a cord and pulley system inside the handpiece sheath, and went on to reach 180 000 rpm (3000/s) by 1960.[7,49] Although the definite benefits of high rotation rate were obtained, the cord arm mechanism remained cumbersome.

The most significant event in this history was the production in 1957 of the first commercially-viable high-speed air-turbine handpiece, the Borden Airotor. Said to be able to run at up to 300 000 rpm (5000/s) it had oil-mist bearing lubrication and an integral water jet for cooling.[50] It sold quickly: by 1958, it was claimed that 50 000 had been sold. Other manufacturers rapidly followed suit. In 1960, the Borden Airotor 60 was advertised as having the advantages of a smaller head, reduced noise, and an improved cooling system with twin jets. The greater convenience and manoeuvrability of the high-speed turbine handpieces led to their market dominance and the disappearance of cord systems.

Patented designs continued to appear, including finger-operated air and water coolant valves, and needle roller bearings for the rotor which acted as the turbine blades.[51] The ball bearings initially used were noisy, required continuous oil-mist lubrication, and wore out rapidly.[50,52,53] Air bearings, introduced in the early 1960s, permitted speeds up to 528 000 rpm (8800/s) to be achieved with air at 60 psi (415 kPa).[50,52,54] Said to be quieter, but requiring a lower load during cutting than ball bearing devices because of the ease of stalling, such handpieces are still in production. Since the 1970s, however, improved, more durable ball races have been available,[50] needing only periodic oiling, and this design is still dominant. Ceramic bearings, said to require no lubrication, were introduced in 1991.

Low-speed work was still considered appropriate in some contexts, but cord systems remained problematic. The Dentatus air motor, introduced in 1960, could operate across a wide range of speeds with relatively high torque. This was in effect, a piston motor: plastic balls acted as pistons in an eccentric rotor. Several manufacturers produced similar devices, but all were ‘rather noisy’.[50]

In the early 1960s, small high-torque 24 V dc electric servo-motors for use in aircraft became the model for a quiet alternative: so-called ‘micromotors’.[50] Air-drive vane motors – the rotor having radial slots containing sliding vanes, rotating in a non-circular casing – were also developed. Capable of 20 000 rpm (333/s) with ‘remarkably high torque’, the advantage was a simple air-driven control system, as have air-turbine handpieces. Both air motors and low-voltage electric motors continue to be used today.

Perhaps prompted by the great success of the Borden Airotor and its successors, there has been lively competition to attribute credit to various individuals.[43,45] As it turns out, the idea of placing the turbine in the head of the handpiece appears to have occurred independently to two separate groups and it is inappropriate, certainly invidious, to seek a single originator. Subsequent developments in high-speed air-turbine handpiece design are of two types: turbine modifications that affect speed and torque, and modifications affecting task suitability or convenience of use. Of these latter can be mentioned multiple coupling connectors with rotatable joints, fibreoptic illumination, push button and lever chuck systems, the ability of the handpiece to withstand routine autoclaving, and ceramic bearings. Larger diameter rotors give increased torque, while smaller rotors (both diameter and length) reduce handpiece head size and thus improve accessibility in the patient's mouth. Improved bearings require simplified lubrication and last longer, but there are no clear distinctions to be made between the many variations of rotor blade design.

1.4 Importance with respect to cutting: work done vs. power in, duty cycle, load, nature of substrate

Cutting performance is understood to relate to the rate of reduction of the workpiece. This is affected by many factors, for example operator characteristics, the handpiece itself, rotary cutting instrument design, coolant applied at the interface, and the workpiece material. Given this, it is not possible to define a representative set of conditions for 'normal' clinical service. Only benchmarking in certain respects, item by item, is presently feasible. A detailed discussion has been given elsewhere,[55] but an outline follows here to illustrate the nature of the problem. It will be understood that this must relate to all aspects of the system: powering device, cutting instrument, cooling, and substrate.

Speed

'Normal' use does not involve constant speed (as in many other systems) but rather an intermittent, continuously-varying rotation rate. As it is expected that the interaction between cutter and substrate depends in part on relative velocity (and thus the strain rate-sensitivity of the substrate), this 'duty cycle' needs to be studied and standardized, and probably with different conditions applying for different cutter–substrate combinations. Furthermore, the volume swept by a cutter blade is proportional to speed, so higher rotation rate means higher rate of removal, if other factors are held fixed.

Angle of attack

This refers primarily to the geometry of the relationship of the blade or cutting point to the surface of the substrate as it affects stresses, flow patterns, and tool wear. However, the attitude and translational motion (i.e. the direction and magnitude of the guiding forces) of the rotating instrument as a whole affects the interface with the substrate, for example: area covered, cutting direction with respect to rotation axis, chip removal, and coolant path, as well as the loads acting on the bearings.

Depth of cut

Obviously, the length of the contact between cutter and substrate affects the volume swept by a blade or point, and thus the work required for cutting.

Cutting instrument design

Factors include: number of blades and their type, e.g. straight, spiral, interrupted; blade rake and clearance angles; sharpness and wear rate, i.e. material properties. Chip clearance affects behaviour: clogging effectively changes cutter design, as does wear, during the course of use.

Coolant

The properties of the substrate depend on temperature (as do those of the cutter), hence on heat transfer rates, via effectiveness of delivery and clearance of coolant. Mechanical properties also depend on the chemical environment provided by the coolant (zeta potential), hence the work of cutting is further affected. Temperature depends on heat delivery, which depends on friction and plastic work done, as well as thermal mass: volume and specific heat, and thermal diffusivity.

Substrate

Again, the mechanical properties of the substrate affect fracture work required, chip behaviour, and cutter wear. No one material can substitute for all possible dental substrates; standardization is therefore not possible.

Power

The rate of material removal depends on the delivery of the energy of fracture and deformation (as well as friction), and thus on the efficiency of the conversion of the drive power to cutting work, via cutter design etc. Even so, this is fundamentally limited by the maximum power that the device can deliver. For a given input, all work done reduces the speed.

Torque and speed

Turbine design, including all aspects that affect air flow to and from the device, determines unalterably the mechanical properties of the machine. Torque and speed are complementary: at maximum, free-running speed no external work can be done as there is no deliverable torque; at maximum torque the machine is stalled. Power is the product of the two, and the maximum occurs at the speed midpoint. The feedback between available torque and actual speed is critical.

Air pressure

As pressure affects flow and the potential for doing useful work, so variation of source pressure affects outcome. Hence, factors such as: connector design; tubing bore smoothness, internal edges and bends; local or remote exhaust; leakage and use as coolant are also involved.

Temperature

Gas behaviour depends on temperature and thus heat flow into the expanding air affects work done.

Bearing friction

Work done in turning the bearings is not available for cutting. Bearing design, conditions – size, design, wear, lubrication, alignment – therefore have a significant influence.

Load

Referring to the normal force of the cutter on the substrate, this affects how much material can be removed but also the forces in the bearings, and thus their friction. However, load is not constant, and the load cycle needs to be studied to ascertain a reasonable standardized pattern.

Substrate relationship

Unlike machine tools, handpieces are hand-held. There is no fixed geometrical relationship (cutting depth) or feed rate. Steady-state conditions cannot be attained.

Feedback

The user has auditory and tactile feedback during the course of cutting, depending on substrate material and cutter design. The interpretation and effect of this must vary between operators, as well as expectations of handpiece behaviour, and are also affected by intention, i.e. whether gross material removal or fine adjustments are required.

It is clear that there is much interaction between all of these factors. Furthermore, there are several significant impediments to full standardized testing: load and duty cycles are unknown, and suitable standardized substrates for tooth tissue are not available. Thus, while the primary concern is to understand actual cutting, it is clear that it is at present not possible to address this in any useful fashion. Accordingly, we are limited to characterizing the machine driving the cutter. It is these measurements and tests that are set out below.

1.5 Testing: equipment, procedure, calculations

Although it is the core of the device, the design requirements for the rotor are poorly understood. In common with many other developments in dentistry, there has been no theory to guide design. This is in contrast to that of large-scale industrial turbines, where a great deal is known. At the scale relevant here, there is little that can be said about which factors control which aspects of behaviour. Despite an extensive search, the characteristics of a wide variety of handpieces could not be explained in terms of the dimensions (other than diameter) of the rotor, the number of blades or their shape.[56] This is an area worthy of further study, as the variety of existing designs suggests that, like tyre treads, there is little to choose between them: they all work, indistinguishably. Other factors have greater importance. Nevertheless, devices vary. It has been found necessary to resort to the 'black-box' approach and determine the 'pressure effectiveness' of the unit. That said, air flow is clearly controlling and, therefore, details of the plumbing, nozzles and vents are important. Here, too, it is not possible to assign quantitative descriptions to these factors – a 'black-box' approach is again necessary. This results in the 'equivalent orifice' determination described below. Further information and background may be found in the authors' publications.[43,45,55–61]

The aspects of the performance of air-turbine handpieces of principal concern with respect to turbine performance are free-running speed (maximum rotation rate), that is, with no external load applied, and torque (and hence power) as functions of speed, rate of air flow, and supply pressure. However, since the bearings are the primary source of internal friction in most designs, a standardized lubrication protocol (and according to the manufacturer's requirements) is an essential first step in any testing, at least for steel bearings. A self-contained test system has been designed to perform the most important tests.[61]

1.5.1 Pressure

Air pressure is the principal input variable, and sufficiently precise, high-quality pressure transducers are readily obtainable: with high accuracy and linearity, low hysteresis, and high repeatability. However, it should be recognized that the value of the pressure observed in a flowing gas is not directly usable in any calculations: there is a correction to be applied to obtain the so-called 'stagnation pressure', of the stationary gas. This can be done as described elsewhere.[60] Pressure needs to be measured as close as possible to the handpiece connector.

In order to control pressure a regulator is required in the supply line, but while manual types are acceptable, for any work that involves measurements at a series of pressures, a stepping motor-driven regulator allows both ready setting at fixed values and scanning when continuous recording is undertaken. The linearity of such regulators is, however, only approximate at best, and actual observed pressure (for subsequent conversion to stagnation pressure) should be recorded rather than controller setting; some variation will occur.

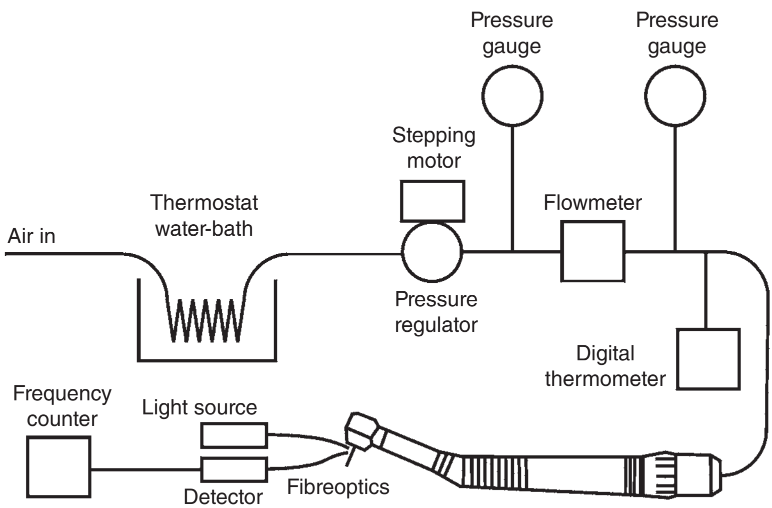
1.5.2 Flow

The accurate measurement of flow rate is a relatively difficult proposition. Temperature variation in the supply pipework, whether from location or the flow of the air (which entails expansion and thus cooling or heat absorption) needs to be carefully controlled. A thermostatted heat exchanger (a long coil of copper tubing in a water bath) close to the point of use has proved to be essential, as well as working in a closely temperature-controlled environment. No matter where the control mechanism is placed, pressure must be measured close to the handpiece (and downstream of the controller) because there is necessarily a pressure drop along any practical size of tubing for this work. This latter also means that, where possible, large-bore connections, without restrictions, are used. Of course, expansion of air through the pressure controller affects temperature, but in practical terms it is only necessary to ensure steady-state conditions and then record the relevant values. Too great a pressure reduction should, even so, be avoided, and the supply line should be arranged to be at a pressure only slightly above the maximum ever required (typically about 4 bar) and of sufficient bore that, when running at full capacity for the handpiece in test, the pressure drop to the regulator does not become a problem, as some do not have a completely independent output.

'Rotameters' are the most convenient device for measuring flow, but rarely can they be read visually to better than a few percent of full scale. A rotameter with an optical position-sensing mechanism is better and both avoids reading errors and reduces the effect of the unavoidable noise in the float position. Devices good to ± 1% full scale are available, but even so the absolute accuracy is likely to be inadequate for precise work and a calibration curve needs to be constructed. It is beyond the present scope to give full details of this, but it essentially requires: a direct measurement of the amount of air delivered, corrected to 1 bar, in a measured time (or time for a particular volume) at a large number of points in the range; a smoothing function to be applied which preserves the non-linearity and local anomalies of the device; and a look-up table being constructed to convert each discrete display value to an actual flow rate in equivalent terms at the standardized conditions of temperature and pressure. This must allow for the effect of varying density in the flowmeter (which affects the reading directly) due to the actual pressure in the flowmeter. Note must also be taken of the working altitude (barometric pressure), and corrections made appropriately – the handpiece behaviour is also affected by these variables.

The pressure drop across the rotameter requires that an additional pressure gauge, of identical type and specifications to that already fitted close to the handpiece connector, is installed in the line near the flowmeter's gas input (Fig. 1.1). The rotameter sees its supply pressure, the handpiece sees the pressure after that drop, hence the need to reduce the flow data to common, standard conditions.

In general, flow is unaffected by handpiece load (i.e. turbine speed) and, in most cases, it is essentially unaffected by stalling the rotor. In a few cases,



1.1 Outline diagram of the instrumentation for characterizing an air-turbine handpiece.

a small reduction (0.1–0.3 L/min) occurs in specific rotor positions (i.e. with a rotor blade adjacent to the supply nozzle, effectively blocking it), although slow rotation (>1/s) causes the effect to become undetectable and thus of no practical importance.

Connection of the air supply to the handpiece needs to be made via the appropriate three-hole or four-hole connector. In normal clinical use, four-hole connectors have flexible tubing attached to the exhaust port to carry the exhaust gas back to the dental unit. Variation in the resistance to exhaust flow is to be expected from this source, depending on the length, bore, and degree of bending of the tubing. It is a matter for consideration whether this effect is to included or avoided. No attached tubing is preferred for handpiece characterization. Typically, three-hole connectors have apertures for the exhaust air, and these must be left unobstructed. It is worthwhile checking that the entire system up to the handpiece itself is leak tested. Pressurizing to 4 or 5 bar, say, and shutting off the air supply at source, allows observation of any pressure drop with time. Less than, say, 0.05 bar in 2 min may be considered adequate.

Since the purpose is generally to measure flow through the turbine alone, it is necessary to block any coolant nozzles that use air, or by closing the supply tube at the handpiece connector. Otherwise, no attempt should be made to close any leaks around the handpiece head as these are properly part of the actual working design. Any internal flow restricting mechanism in the handpiece (to allow a switch to be made between two alternative supply pressures at the connector) should be noted, but most commonly these are set in the open position in practice.

The complexity of the gas path in the handpiece means that no simple theoretical equation exists to relate actual flow rate, $\dot{V}$, after the necessary corrections, to the supply (absolute) stagnation pressure, p_0, and at that pressure. The pragmatic 'black-box' approach entails a simple non-linear curve fit. The following function provides an adequate description of the behaviour:

$$\dot{V} = \dot{V}_{\mathrm{L}}\left[1-\left(\frac{A}{p_0}\right)^b\right]^{1/2}$$

A and b are the parameters to be fitted and which control the curve shape (A represents ambient pressure, but b cannot be interpreted in terms of a specific feature of handpiece design), and $\dot{V}_{\mathrm{L}}$ is the limiting flow rate (i.e. at 'choke') which corresponds to the flow at some point in the system reaching the sonic velocity. Since such a condition can occur even in a simple circular orifice (such as the nozzle in the turbine head), the handpiece can be characterized as being equivalent to such an orifice via the equation for choke. The relationship is:

$$\dot{V}_L = \pi r_e^2 c$$

where r_e is the effective radius of that orifice and c is the speed of sound, itself given by:

$$c = \sqrt{\frac{\gamma p_0}{\rho}}$$

where ρ is the density under the prevailing conditions, p_0 is the absolute stagnation pressure, and γ is the ratio of specific heats c_p/c_v (which may be treated as a constant as the absolute temperature is sufficiently high and changes are moderate). The area of the equivalent orifice, πr_e^2, is therefore a measure of the ability of the handpiece to deliver air to the turbine in a strictly controlling sense.

If the flow rate needs to be expressed in terms of 'free air', i.e., at atmospheric pressure, to relate this to compressor pumping capacity, for example, the isothermal gas law (Boyle's) is used:

$$V_2 = p_1 V_1 / p_2$$

where p_1, p_2 are the pressures and V_1, V_2 the volumes before and after expansion to atmospheric pressure, respectively.

1.5.3 Temperature

Supply air temperature can be measured well enough with a type-K thermocouple placed in the gas flow beyond the flowmeter, providing this does not interfere with the flow, i.e. it is small enough. A T-piece such that the thermocouple is level with the supply line wall is adequate. However, it must be remembered that all gas calculations are in terms of absolute temperature.

1.5.4 Speed

Many methods have been used to monitor rotation, with varying degrees of success. These techniques have been reviewed in detail elsewhere,[58] and the deficiencies identified. Clearly, any technique that requires work from the rotating system is unacceptable, and non-contact methods are essential. Likewise, anything that appreciably changes the angular inertia or balance of the system is inappropriate. Severe errors are introduced if these points are not adhered to. The most effective means of speed monitoring is optical, and a fibreoptic tachometer can be used to measure the rotational speed of any handpiece, cutting instrument, or test mandrel; it provides no load on the turbine and has minimal bulk. Ultimately, such an approach would be intrinsically safe for patient contact.

A sheathed 1-mm-diameter acrylic optical fibre with polished ends can be used to transmit light from a (heat-filtered) quartz halogen lamp onto the shaft of the tool in the handpiece chuck. The shaft needs to have approximately one-half of its circumference close to the nose of the chuck coated with a matt black ink (i.e. of very small mass), the remainder being left bright. A second optical fibre, similar to the first and at right angles to it, is then held in a position to receive light reflected from the unpainted half of the shaft such that at each rotation the receiving fibre returns a pulse of light. This is then to be applied to a photo-detector whose output is connected to a frequency counter. This latter needs to be of high stability and with no zero-offset,[62] but with a recordable output signal proportional to frequency, and hence speed. Alignment of the optical fibres is best done using an oscilloscope to view the signal, using auto-triggered time-sweep mode. Adjustment should be aimed at obtaining an approximately 50% on–off (duty) cycle.

1.5.5 Free-running speed

The mandrel with the half-black sector is fixed in the handpiece chuck and the optical fibres adjusted and checked for correct operation (oscilloscope). If a pressure scan is undertaken, the rate of pressure change needs to be slow in relation to the response time constant of the system as a whole (equilibration of the gas stream, inertia of the turbine assembly); this can be determined by trial and error, but probably no more than about 1 or 1.5 bar/min is appropriate. The maximum pressure used depends on the equipment used, but clearly it needs to be checked for safety. Even so, 4 bar probably represents a general upper limit, depending on the relevant manufacturer's recommendations. A double cycle of raising and then lowering the pressure to zero allows a check of reproducibility. Supply pressure, temperature, and rotation rate are logged.

The upper limit to the rotational speed attainable for a rotor is set by the speed of sound, so the actual free-running speed N_f is expressed through the peripheral Mach number Ma^* at a given supply pressure:

$$Ma^* = 2\pi r_r N_f / c$$

where r_r is the radius of the rotor. There remain turbine behavioural variables that have yet to be resolved, as indicated above, so the 'black-box' approach is used to define a rotor performance coefficient, α_r:

$$\alpha_r = \frac{\left\{1-(\gamma-1)(Ma^*)^2\right\}^{-\frac{\gamma}{\gamma-1}}-1}{\pi r_e^2 p_{0g}}$$

which takes into account all such remaining aspects, determined from the free-running speed and absolute stagnation pressure of the air supply, through the equivalent orifice. Note the subscript 'g'; this is a gauge pressure, i.e. the difference between the actual absolute stagnation pressure and the surrounding atmospheric pressure. However, as far as the user is concerned, it is how well the device utilizes or translates the supply pressure. Hence, we may combine the effective orifice and the rotor performance coefficient into a single descriptor, the 'pressure effectiveness', α:

$$\alpha = \alpha_{\mathrm{r}} \pi r_{\mathrm{e}}^2 = \left[\left\{ 1 - (\gamma - 1)(Ma^*)^2 \right\}^{-\frac{\gamma}{\gamma-1}} - 1 \right] / p_{0\mathrm{g}}$$

This can be determined as the slope of the least upper bound of a plot of the square-bracketed term on the right vs. $p_{0\mathrm{g}}$. Numerically, free-running speed can then be estimated from

$$\hat{N}_{\mathrm{f}} = \frac{5.0358}{r} \sqrt{T \left\{ 1 - (1 + \alpha p_{0\mathrm{g}} / p_{\mathrm{at}})^{-0.2867} \right\}}$$

where p_{at} is the (absolute) atmospheric pressure in bar, and T is the absolute temperature of the supply air.

1.5.6 Torque

Torque (τ) is defined as the moment of the force that tends to produce rotation. From Newton's third law of motion, this force is equal and opposite to the shaft braking force:

$$\tau = Fr$$

where F is the braking force and r is the radial distance of the point of application of that force from the axis of rotation. If the braking force is gradually increased from zero, the turbine speed will progressively reduce until it stalls. Thus, for a given turbine, τ vs. rotation rate (N/Hz) can be plotted, the maximum value for τ being at the point of stall.

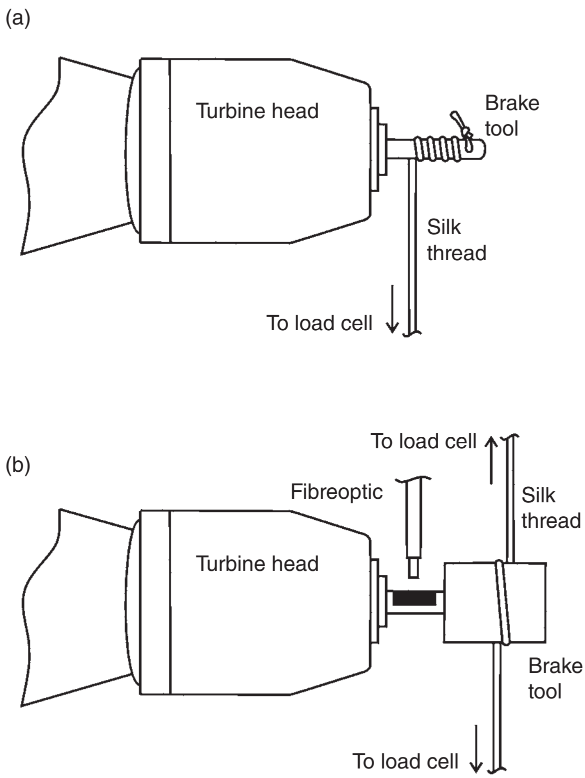
Braking force is most easily determined by the technique known as a 'rope brake'.[63] This requires a thread wrapped around the rotating shaft, friction being applied by tension in that thread, the braking force. The measurement is best performed with a load cell on a universal mechanical testing machine, or equivalent frame, for stability and accuracy. The shaft or drum in contact with the thread needs to be rather well-polished if excessive wear (and frequent replacement) of the thread is to be avoided in dynamic torque tests. Even so, replacement can be expected to be required when oxidation becomes excessive and polishing no longer practical. The shaft can become quite hot and it is important to allow it to cool between runs if the thread is not to burn. While in principle almost any kind of thread will suffice, in practice there are some factors to be borne in mind. The heat of friction may melt thermoplastic polymers, and stable close approximation of thread to shaft is essential for steady values to be obtained. The latter implies flexibility, as obtained from a fine-fibre yarn. Similarly, a yarn that tends to flatten somewhat and obtain more points of contact will be preferable. Overall, the most effective thread has been found to be braided suture silk. Lubricants are of no benefit in this context. Displacement over time causes erratic effects, and their stickiness can cause the fibres of the thread to catch and bind.

The effective end of the moment arm does not lie at the surface of the drum or shaft in contact with the thread, and a small error will be present if this is assumed. For the braided suture silk that is recommended for the thread, the correction amounts to one-third of the thickness when flattened under load. This is measured well enough by a caliper gauge on the overall diameter with the thread in place.

A universal mechanical testing machine fitted with two load cell amplifiers (LCAs) and two load cells (10 N full scale range is appropriate) can be used to measure forces and thus deduce torque. One load cell is mounted (say) on the upper, fixed, crosshead and the other to the upper surface of the moveable crosshead, ensuring the two load axes are accurately aligned. A self-centering thread attachment is required on each. Provision for clamping the handpiece accurately in place must be made so that its position with respect to the load axis is preserved, most conveniently this is fixed to a crosshead (a hard rubber 'V-block' mounting is satistfactory, but see reference 61 for a more precise approach).

1.5.7 Bearing resistance

This is the torque required to rotate the turbine against the friction of the bearings. Note that this cannot be done for air bearings (which one hopes would have negligible intrinsic friction) as they require air to be supplied in order to operate. A brake tool can be prepared from a highly polished mandrel, of accurately measured diameter, by drilling a small (say, 0.70 mm) diameter hole perpendicular to its axis and 1.5 mm from one end (Fig. 1.2a). The tool being fitted in the handpiece chuck, the handpiece is then placed in the mounting. Alignment of the brake tool is critical. A datum can be provided by attaching a (silk) thread to the upper and lower load cells, and placing this under slight tension. The handpiece position can then be adjusted so that the surface of the brake tool just contacts the thread, some 2–3 mm from the drilled hole. An engineer's parallel can be used as a horizontal datum for the axis of the brake tool with respect to the frame of the testing machine and thus normal to the thread. The handpiece must then be firmly clamped and the thread between the load cells can be removed.



1.2 *a* Design and use of brake tool for determining bearing resistance, *b* design and use of brake tool for determining dynamic torque.

A 150 mm length of silk thread is then passed through the hole in the mandrel and secured with a knot, and the other end is attached to the thread holder of the (movable) load cell by means of a loop tied at its free end. The moving crosshead is then moved to apply a slight tension to the thread (~1 N), and air at a pressure of ~1 bar is supplied to the handpiece while the crosshead is slowly moved (~10 mm/min) so that the thread can be guided manually to wrap around the braking tool with four to five slightly-spaced turns (they must not overlap), driven by the turbine.

The air supply is then removed and the crosshead set to move at (say) 5 mm/min so that the the force can be recorded for at least one full revolution of the turbine. Rotational position needs to be noted carefully for this. The bearing resistance (as torque) is then calculated as the mean for one full rotation.

1.5.8 Stall torque

This is the torque generated by the air flowing through the non-rotating turbine. The same set-up is used as for bearing resistance (Section 1.5.7), but now the air pressure is adjusted to that required for the test, allowing a few seconds for stabilization. The moving crosshead can then be moved away at, say, 5 mm/min for recordings to be made of pressure vs. load and flow rate until the turbine has rotated through at least 360°e. For repeat tests, or tests at other pressures, the crosshead is returned to its original position while the thread is guided back into evenly spaced turns on the tool as above. The rate of crosshead movement chosen must be slow enough to effectively stall the turbine. No distinction can then be made between the values obtained with this rate of movement and those achieved by stopping the crosshead at frequent small increments. Stall torque is then taken as the average over a full rotation, as determined by reference to the variation (if any) with position and a knowledge of the number of blades and nozzles in the device. Such recordings also allow the effect of rotor position to be investigated. It is to be noted that averaged stall torque is not, in general, the same as the mean of the upper and lower limits observed: the variation with rotor position is more complicated than permits this simple calculation. In effect, it is determined by the area under the curve for one full rotation, a properly weighted mean over all positions.

Stall torque is directly proportional to the stagnation air pressure, p_{0g}:

$$\tau_p = \Phi p_{0g}$$

where Φ is the stall torque coefficient, atmospheric pressure of 1 bar. Thus, in principle, only one determination needs to be made in order to predict behaviour at other pressures, providing the air flow remains subsonic (which is the case in the dental context).

1.5.9 Dynamic torque

Although the theoretical torque behaviour of a driven rotating system is well-understood, the effects of imbalance in rotor or tools can only be detected dynamically. Thus, a direct measurement of dynamic torque may be of value as a means of detecting imbalance or resonance which can only reduce the available torque.

Brake tools, with highly polished drums, may be made by precise machining from stainless steel. Because the balance of the brake tool is crucial to this test, those to be used must first be screened using a handpiece that has been shown by trial to be well-balanced and to show no resonance effects itself. A free-running speed vs. air pressure scan then allows those tools with defects to be discarded. This is a very severe test, but a discrepancy of more than 300/s at any pressure can be taken as a convenient criterion for discard. If the brake tool drum diameter is too small, the thread tends to break before stall can be achieved; too large and obtaining satisfactory balance becomes too difficult. Some 3.0 ± 0.5 mm may be suitable.

The diameter of the brake drum needs to be determined accurately with a micrometer screw gauge or equivalent device (and corrected for thread thickness as described for stall torque) before being fitted in the handpiece chuck. The handpiece is then mounted and aligned as for stall torque. A check should be made that the speed detector is working properly and the load cells zeroed and calibrated, confirming by applying a slight tension to the thread (–2 N) that the outputs are equal. However, it is the difference in load that is required for the actual test and this may be obtained directly by a differential voltage recording, or by post-processing of the two load signals.

The movable crosshead is then moved (no air pressure applied to the handpiece) to allow a single turn of the thread to be passed over the circumference of the brake drum (Fig. 1.2b). The crosshead is then repositioned to tighten the thread sufficiently to prevent rotation of the turbine at the maximum supply pressure to be tested. (Differential) force vs. speed can then be recorded while the crosshead is raised and lowered, at 1 mm/min between the points giving (close to) the free-running speed (with zero difference signal from the load cells) and stall. It should be noted that the slightest touch of the thread is enough to slow the turbine, and the true free-running speed will not be reached in this test. Resonance effects can lead to marked (negative) deviations of the torque–speed plot from the expected straight line.

The supply pressure required to start a turbine may vary according to the position of the rotor at stall. It is therefore easiest to determine the stall pressure or torque only on lowering the pressure, although the direct measurement of stall torque as described above is a simpler procedure.

The expected torque at any supply pressure, $\hat{\tau}_{\mathrm{p}}$, is expressed by the following equation:

$$\hat{\tau}_{\mathrm{p}} = \Phi p_{0\mathrm{g}} \left(1 - \frac{N}{\hat{N}_{\mathrm{fp}}}\right)$$

That is, it decreases linearly with speed N to zero at the free-running speed, $\hat{N}_{\mathrm{fp}}$, simply scaling the stall torque by the proportion of the free-running speed then exhibited. (The subscript ‘p’ is added to the free-running speed symbol to emphasize that it is pressure-dependent and specified for the stated supply pressure, $p_{0\mathrm{g}}$).

1.5.10 Power

The rate at which a handpiece supplies energy to the cutting site, its power, is determined by the handpiece's torque and speed. These describe the ability to carry out cutting work but such data will be essential for use in the analysis of cutting behaviour, when this becomes feasible.

Power (P), the rate of doing work, is given by:

$$P = \omega\tau$$

where ω is the angular velocity in radians/s

$$\omega = 2\pi N$$

i.e.

$$P = 2\pi N\tau$$

Determination of power thus, in principle, requires (dynamic) torque and rotation rate to be determined simultaneously. We can, however, write for the expected power, $\hat{P}$:

$$\hat{P} = 2\pi\Phi p_{0g}\left(1 - \frac{N}{\hat{N}_{fp}}\right)N$$

which is therefore a parabolic curve with zeros at stall and free-running. The maximum expected power, $\hat{P}_{max}$, therefore occurs at half the free-running speed, $N = \hat{N}_{fp}/2$:

$$\hat{P}_{max} = \frac{\pi}{2}\Phi p_{0g}\hat{N}_{fp}$$

1.5.11 Efficiency

There are several possible approaches to the definition of efficiency in the present context, but the primary interest may be in the demands placed by the handpiece on the compressor. Thus, we take the efficiency of a handpiece as the ratio of the useful work done at the rotary cutting instrument to the potential work in the supplied compressed air. Actual power is obtained from torque and speed as above, while the potential energy of the air supply is the maximum work available when the air expands to atmospheric pressure. Here, we may reasonably assume that the air behaves sufficiently like an ideal gas. As the expansion occurs quite rapidly it may further be assumed (conservatively) that this process is adiabatic and reversible (i.e. non-dissipative). The maximum work available on expansion (W) for a given flow rate $\dot{V}$ m^3/s (at the supply pressure) is given by:

$$W = \frac{\dot{V}}{1-\gamma}\left(p_1^{1/\gamma} p_2^{(\gamma-1)/\gamma} - p_1\right)$$

where p_1 and p_2 (in Pa) are the inlet and outlet absolute stagnation pressures, respectively. Efficiency (η) is then given by:

$$\eta = \frac{P}{W}$$

The difference between P and W depends on the rate at which 'unused' energy is discharged in the exhaust plus the total rate of energy loss due to friction in rotor bearings, air flow within the handpiece, and noise emission (which is in fact small). It may therefore be seen that handpiece efficiency (η) at any particular speed (N) may be determined from measurements of supply pressure (p_1), air flow (V_1/s), and torque (T). The maximum efficiency is seen to occur at the maximum expected power:

$$\eta_{\max} = \frac{\hat{P}_{\max}}{W}$$

1.5.12 Longevity

The working life of a dental air turbine is limited by the length of time its bearings can adequately function under the conditions imposed during clinical work. Standardized laboratory longevity tests of the rotor-bearing assembly require that the handpiece is subjected to the same patterns of use that might be expected in clinical practice, that is, in respect of duty cycle, axial and lateral loading, lubrication state, and cleaning and sterilization procedures. This is very time consuming and laborious. Only one report of a study involving this kind of work has been published.[64] Bearing longevity tests in which only gross failure (such as collapse of bearing casings or seizure) can be detected require the handpieces to be run for lengthy periods, but, in view of the ordinary expectation of lubrication for metal bearings, such results may not be very meaningful. However, bearing resistance may be monitored as a sensitive means of detecting wear or corrosion. Acceptability criteria have not been established in this sense.

1.5.13 Comparisons

Given the sensitivity of several of the above measures to the supply pressure, it is clearly not easy to make comparisons between devices whose recommended pressures differ, except in a restricted sense. Indeed, a manufacturer could raise the torque and power of a handpiece simply by specifying a higher working pressure. It is therefore informative to use standardized measures at an arbitrary but representative supply pressure, say, 2 bar. We can then define the following:

- standardized free-running speed

$$N_{f2}$$

- standardized power index

$$PI_2 = \pi \Phi \hat{N}_{f2}$$

- standardized efficiency index

$$EI_2 = \frac{PI_2}{W}$$

- standardized air consumption

$$p\dot{V}_2 = 3\dot{V}_L \left[1 - \left(\frac{1}{3} \right)^b \right]^{1/2}$$

(where $p_{0g} = 2$, so $p_0 = 3$, and A is set to 1 bar in the equation on page 12)

It might also be convenient to determine the corresponding values for the recommended supply pressure, when this differs.

1.5.14 Figures of merit

Table 1.1 summarizes the descriptors and performance measures detailed in the above sections. The table is divided into two parts, the first containing

Table 1.1 Summary of the descriptive parameters and performance figures of merit for handpiece air-turbine behaviour

Character of factor	Symbol
Values required in the course of testing	
Turbine rotor radius	r
Limiting air flow	$\dot{V}_L$
Flow exponent	b
Equivalent orifice radius	r_e
Standardized free-running speed (2 bar)	N_{f2}
Rotor performance coefficient	α_r
Figures of merit of interest to the user	
Handpiece pressure effectiveness	α
Weighted mean stall torque coefficient	Φ
Standardized free air consumption (2 bar)	$p\dot{V}_2$
Standardized power index (2 bar)	PI_2
Standardized efficiency index (2 bar)	EI_2

those items that need to be determined for the testing itself, and the second those figures of merit that convey quantities that are directly comparable between devices and which are therefore of interest to the user. Some examples of values can be found elsewhere.[56]

1.6 Hazards

There is a vast literature dealing with the hazards posed to the patient and dental personnel by air-turbine handpiece use. It is only possible to give a brief account here of the problems of heat, airborne materials, sterility, noise, projectiles and surgical emphysema.

1.6.1 Heat

Heat generation has possible harmful effects on the dental pulp, dental hard tissues and restorative materials. Temperature changes occur because of the thermalization of the work of friction and plastic deformation occurring at the cutter–substrate interface. Actual temperature depends on rate of power deposition and the thermal diffusivity of the materials involved, including the effects of coolants. However, care must be taken to distinguish purely thermal effects on vital teeth[65–67] from damage that occurs due to mechanical vibration,[68,69] dehydration,[67,70–72] and the chemical effects of solutions or restorative materials that are subsequently applied.[73]

Effects range from internal resorption[74] to dentinal reddening or 'blushing' due to dilatation and rupture of blood vessels, and extravasation of erythrocytes.[75] Histological changes include an inflammatory response (e.g. vasodilatation, haemorrhage, oedema, and infiltration of polymorphonuclear leucocytes),[76] changes in the layer of odontoblasts (e.g. the appearance of vacuoles within the layer,[76] and displacement of odontoblastic nuclei with separation from their cytoplasmic processes).[77] Some practitioners prefer to perform cavity preparation dry on the grounds that visibility is improved[78,79] but there is much evidence that this is hazardous to the pulp.[72,78–90] Air coolant alone is inadequate.[91]

Surface effects – such as smearing, cratering, and crack formation – that may occur during preparation of cavities and finishing of restorations have been observed,[78,79,92–102] while dry cutting of enamel can induce sufficiently high thermal stress to fracture it,[101] possibly aided by cracks induced by the hammering of blades. Raising the temperature of amalgam may melt it peritectically, while polymer-based materials may be raised above their glass-transition temperatures and suffer permanent plastic deformation or decomposition. Glass-ionomer and similar cements may be dehydrated.