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Thermal–Structural Comparison of Aluminium, Steel and Titanium Pistons for a Small Single-Cylinder SI Engine

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Abstract—The piston is one of the most heavily loaded parts of an internal combustion engine, and its material influences strength, temperature, deformation and reciprocating mass. This paper compares aluminium alloy A2218, AISI 4340 steel and Ti-6Al-4V titanium alloy for the piston of a 50 mm bore, 49.5 mm stroke, 9:1 compression ratio single-cylinder spark-ignition engine running at 7500 rpm. A steady-state axisymmetric thermal model, solved by the finite-volume method, provides the temperature field under combustion heat flux and oil-side cooling. The crown is treated as a clamped circular plate under gas pressure with an added thermal-stress term, and the pin boss as a thick-walled ring. For identical geometry (4 mm crown) the peak crown temperature is 219 °C for A2218, 281 °C for AISI 4340 and 473 °C for Ti-6Al-4V, because thermal conductivity falls from 150 to 44.5 and 6.7 W/m·K. The thermal stress contribution is 19 % of the total stress for aluminium but 53 % for titanium. When the crown is sized for a safety factor of 2, A2218 needs 5.5 mm and gives the lightest piston (87.9 g), against 125.6 g for Ti-6Al-4V (3.5 mm) and 228.3 g for AISI 4340 (4.0 mm). Aluminium is therefore the best choice for this engine, titanium requires a crown thermal strategy, and steel is justified only if its walls can be made much thinner. The results are from a simplified model and are intended as a pre-design comparison to be confirmed by three-dimensional FEA and testing.

Index Terms—Piston, material selection, aluminium A2218, AISI 4340, Ti-6Al-4V, thermal stress, finite-volume analysis, internal combustion engine.

I. INTRODUCTION

1. Background and Context

The piston transmits the gas force produced by combustion to the crankshaft through the connecting rod, seals the combustion chamber with its rings, and rejects part of the combustion heat to the cylinder wall and the lubricating oil. It therefore experiences simultaneously a cyclic mechanical load, a steep temperature gradient and sliding contact [1], [2]. Failure modes include crown cracking, ring-groove wear, seizure, and thermal fatigue; all of them are strongly influenced by the piston material.

Aluminium alloys dominate small and medium engines because of their low density and high thermal conductivity, which lower reciprocating mass and keep the crown cool. Steel pistons, which are stronger and more resistant to heat, are increasingly used in highly loaded diesel engines, and titanium alloys have been considered for racing engines for their high specific strength [2], [3]. The choice among these is not obvious for a small, high-speed gasoline engine, where the stiffness and strength gain of steel

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and titanium must be weighed against their higher density and, for titanium, very low thermal conductivity.

Finite-element studies of piston temperature and stress are widely reported, for example for motorcycle-size pistons in aluminium alloys [4], [5], but most of them compare design variants of one material. The purpose of this paper is a direct comparison of three candidate materials under identical geometry and boundary conditions, and then under identical safety factor, using a compact model whose every step can be reproduced by hand or in a spreadsheet. The paper contributes: (i) a thermal–structural pre-design model for the crown and pin boss; (ii) a material comparison at fixed geometry; (iii) a comparison at fixed safety factor in terms of crown thickness and piston mass; and (iv) a discussion of design consequences for each material.

II. ENGINE, GEOMETRY AND MATERIALS

1. Engine Specification

The study uses the engine data of Table I. A maximum combustion pressure of 4 MPa, the mid-range of the 3–5 MPa band for such engines, is used for the structural load [2].

Parameter	Value
Type	Single-cylinder, 4-stroke SI
Bore × stroke	50 mm × 49.5 mm
Compression ratio	9 : 1
Rated speed	7500 rpm
Peak gas pressure p	4 MPa (design)
Crown heat flux q	0.30 MW/m^2 (assumed)
Underside cooling	h = 1500 W/m^{2}K, 100 °C
Outer wall (ring zone)	h = 1500 W/m^{2}K, 150 °C

B. Piston Geometry

A simplified piston is analysed: a circular crown of radius a = 25 mm and thickness t, a skirt wall 3.5 mm thick and 38 mm in total height, and two pin bosses of 14 mm pin bore (0.28 D) and 12 mm length (Fig. 1). Ring grooves, fillets and the bowl are omitted in order to isolate the effect of material. The reference crown thickness is 4 mm and the reference boss thickness is 3 mm. As a check, the classical head-thickness formula th = √(3pD2/16σt) [6], with a permissible tensile stress of 50 MPa for aluminium, gives 6.1 mm, which is similar to the value obtained below from the safety-factor criterion.

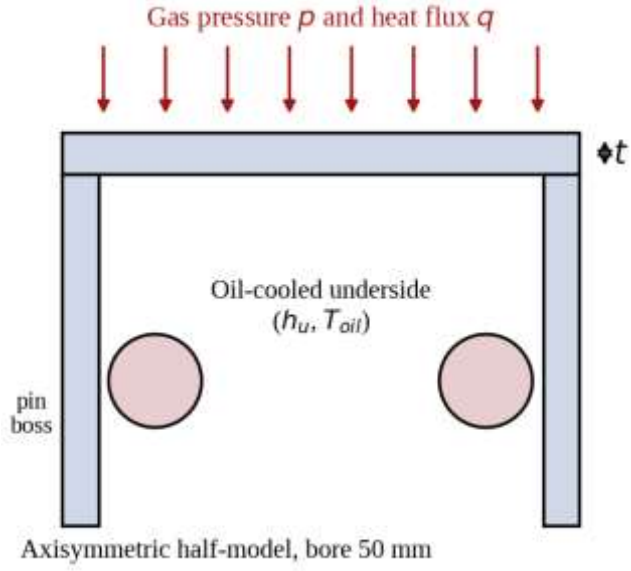


Fig. 1. Axisymmetric piston model with loads and cooling boundaries.

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C. Materials

Table II lists the properties used. Values are typical handbook values for A2218-T61, annealed AISI 4340 and annealed Ti-6Al-4V [7]–[9]; exact numbers depend on temper and supplier. The yield strength is derated linearly with temperature to a fraction of its room-temperature value at a reference temperature (last two rows). These derating assumptions are adopted for this study and should be replaced by supplier curves in a real design

Property	A2218	AISI 4340	Ti-6Al-4V
E (GPa)	74	205	114
ν	0.33	0.29	0.34
k (W/m·K)	150	44.5	6.7
α (10^{-6}/K)	22.0	12.3	8.6
ρ (kg/m^3)	2810	7850	4430
σ_y at 25 °C (MPa)	255	470	880
σ_y/ρ (kN·m/kg)	90.7	59.9	198.6
Derating	50 % at 300 °C	80 % at 400 °C	70 % at 400 °C

III. ANALYSIS METHOD

1. Thermal Model

The steady temperature field in the axisymmetric piston obeys

$$k\,[\,\partial 2T/\partial r2 + (1/r)\,\partial T/\partial r + \partial 2T/\partial z2\,] = 0 \quad (1)$$

with the heat flux q on the crown top and convection on the underside and outer wall (Table I). The equation was discretised by the finite-volume method on a 0.5 mm grid and the sparse system solved directly. The crown heat flux and the heat-transfer coefficients are representative assumed values; they set the absolute temperature level for all three materials equally, so the comparison between materials is meaningful even though the absolute temperatures are not predictions for a specific engine.

B. Crown Stress and Deflection

The crown is idealised as a circular plate clamped at its edge and loaded by the gas pressure p. The maximum bending stress and the centre deflection are [6]

$$\sigma b = 0.75\ p\ (a/t)2, \quad w = 0.171\ p\ a4\ /\ (E\ t3) \qquad (2)$$

The temperature differences obtained from (1) between the crown centre and its underside (ΔTz) and between the crown centre and the crown edge (ΔTr) generate thermal stress. The larger of the two is used in the restrained-gradient expression [10]

$$\sigma th = E\ \alpha\ \max(\Delta Tz, \Delta Tr)\ /\ [2(1 - \nu)] \qquad (3)$$

The equivalent stress is the algebraic sum $\sigma eq = \sigma b + \sigma th$, a conservative combination, and the safety factor is $SF = \sigma y(Tmax)/\sigma eq$. The free thermal growth of the crown diameter is estimated as $\delta = \alpha\ (Tavg - 25) \cdot 2a$ from the mean crown temperature.

C. Pin Boss and Mass

The gas force $F = p\pi a2$ (7.9 kN) loads the two bosses, giving a bearing pressure $pb = F/(2dL)$ on the projected pin area. The boss is treated as a thick-walled ring of inner radius ri = 7 mm and outer radius ro, whose hoop stress at the bore is

$$\sigma\theta = pb\ (ro2 + ri2)\ /\ (ro2 - ri2) \quad (4)$$

The boss temperature is taken from the thermal solution at mid-wall height. Piston mass is computed from the volumes of the crown, skirt wall and bosses and the density in Table II.

D. Design Procedure

The calculation is arranged as a repeatable procedure so that a candidate material or thickness can be evaluated in minutes: (1) fix the engine data, peak gas pressure and thermal boundary conditions; (2) choose the material and a trial crown and boss thickness; (3) solve (1) for the temperature field and read

Tmax, ΔTz, ΔTr, Tavg and the boss temperature; (4) derate the yield strength at Tmax and evaluate (2)–(4); (5) compute SF and the mass; (6) if SF is below the target, increase the crown thickness in 0.5 mm steps and repeat from step 3, because the temperature field depends on thickness. The procedure converges in a few iterations because the temperature changes only slowly with t, while the stress changes with 1/t2. This is the same logic that a three-dimensional model would follow, with the closed-form stress formulas replaced by a finite-element solution.

Two choices in the procedure deserve comment. First, the thermal stress is combined with the bending stress by addition, rather than by a root-sum-square, which is conservative since the two stresses do not necessarily peak at the same point or have the same direction. Second, the safety factor is evaluated against the yield strength at the peak crown temperature, not at the mean temperature; this is again conservative because only a small region of the crown reaches the peak, but it is appropriate for a screening study whose aim is to avoid underestimating the risk to a hot, low-conductivity material.

IV. RESULTS

1. Temperature Distribution

Fig. 2 shows the temperature field for the three materials with the same 4 mm crown, plotted on a common scale. The peak crown temperature is 219 °C for A2218, 281 °C for AISI 4340 and 473 °C for Ti-6Al-4V (Table III). The ranking follows thermal conductivity: aluminium conducts the crown heat to the cooled wall with a small temperature drop, whereas titanium, with a conductivity of only 6.7 W/m·K, builds up a large through-thickness gradient (177 K) as well as a large radial gradient (79 K). The titanium crown is also hotter than the range in which Ti-6Al-4V retains its strength, which is why its derating is the most severe.

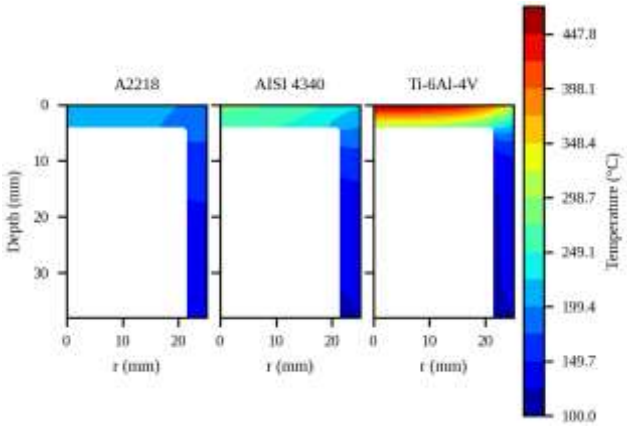


Fig. 2. Steady temperature fields for A2218, AISI 4340 and Ti-6Al-4V with a 4 mm crown (common colour scale).

2. Stress, Safety Factor and Deformation

At the same crown thickness the bending stress is identical for all materials (117.2 MPa) because it depends only on pressure and geometry; the difference is the thermal stress, which rises from 27 MPa for A2218 to 78 MPa for steel and 131 MPa for titanium, so that the thermal contribution is 19 %, 40 % and 53 % of the total stress, respectively (Fig. 4a). Despite this, the safety factor of the two stronger materials is higher (2.08 for steel and 2.27 for titanium) than for aluminium (1.14), because their derated yield strength is much higher. A2218 at 4 mm thus does not meet a typical safety factor of 2

In OB/GYN ultrasound, CNNs have been used to detect standard fetal planes in real time [2], to recognize common maternal–fetal planes with accuracy close to experts [4], to measure fetal head circumference [22], to detect fibroids [23] and PCOS [7], and to separate benign from malignant ovarian tumours at a level comparable to expert examiners in a large multicentre study [6]. Most of these models are trained by transfer learning: a network pre-trained on millions of natural photographs is fine-tuned on a smaller set of medical images.

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Quantity	A2218	AISI 4340	Ti-6Al-4V
Peak temperature (°C)	219.2	281.1	472.6
ΔT_z / ΔT_r (K)	6.3 / 22.4	24.0 / 43.7	177.0 / 79.4
σ_b (MPa)	117.2	117.2	117.2
σ_{th} (MPa)	27.3	77.6	131.5
σ_{eq} (MPa)	144.4	194.8	248.7
Crown SF	1.14	2.08	2.27
Plate deflection w (mm)	0.056	0.020	0.037
Free growth δ (mm)	0.20	0.14	0.14
Boss hoop stress (MPa)	68.3	68.3	68.3
Boss SF	2.92	6.47	11.84
Piston mass (g)	81.7	228.3	128.8

Crown deflection under gas pressure is small for all three materials (0.02–0.06 mm) and is smaller for stiffer materials. Free thermal growth of the crown diameter is 0.20 mm for aluminium and 0.14 mm for steel and titanium; because aluminium expands more, the piston-to-bore running clearance must be larger for the aluminium piston, which is a known design requirement for aluminium pistons in iron or steel liners [1], [2]. The boss hoop stress is the same for all materials (68.3 MPa), and its safety factor ranges from 2.9 for aluminium to 11.8 for titanium, so the pin boss is not governing in any of the cases.

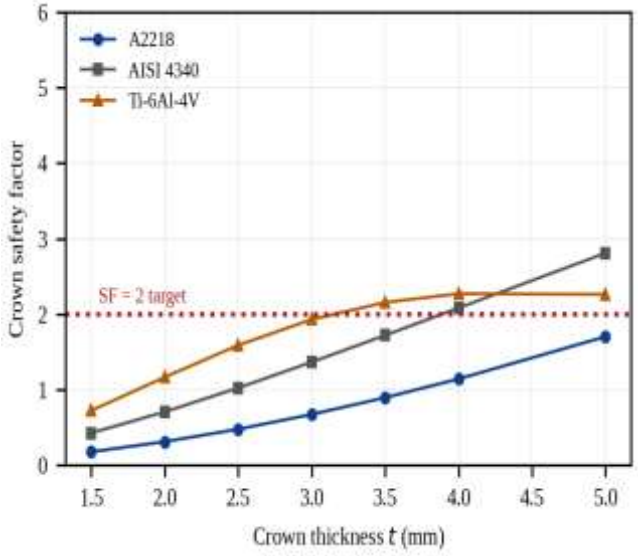


Fig. 3. Crown safety factor versus crown thickness for the three materials.

3. Equal Safety Factor and Mass

Fig. 3 shows how the safety factor grows with crown thickness. If the crown is sized to the smallest thickness (in 0.5 mm steps) that gives SF ≥ 2 at the computed temperature, the required thickness is 5.5 mm for A2218, 4.0 mm for AISI 4340 and 3.5 mm for Ti-6Al-4V (Table IV). The resulting piston masses are 87.9 g, 228.3 g and 125.6 g. Although aluminium needs the thickest crown, its low density makes the piston 61 % lighter than the steel piston and 30 % lighter than the titanium piston for the same wall geometry (Fig. 4b). The aluminium result of 5.5 mm is consistent with the 6.1 mm of the classical head-thickness formula.

Quantity	A2218	AISI 4340	Ti-6Al-4V
Crown thickness (mm)	5.5	4.0	3.5
Peak temperature (°C)	215.6	281.1	452.5
σ_{eq} (MPa)	82.7	194.8	268.5
Crown SF	2.01	2.08	2.16
Piston mass (g)	87.9	228.3	125.6
Mass relative to A2218	1.00	2.60	1.43

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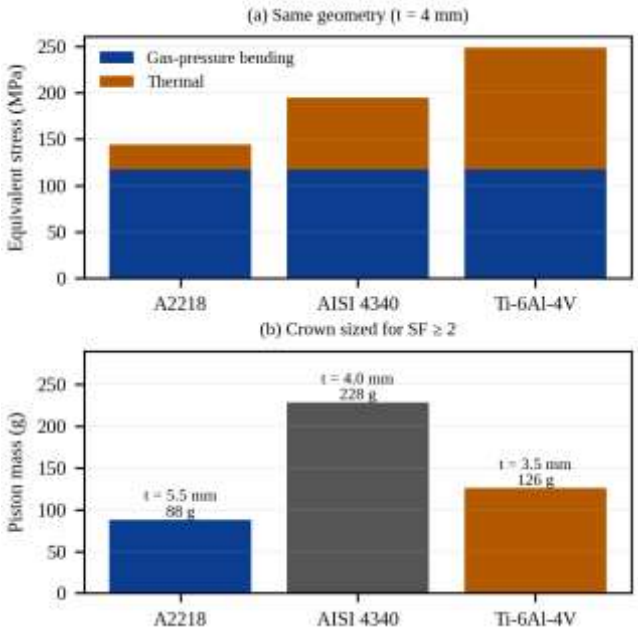


Fig. 4. (a) Stress components at equal geometry; (b) crown thickness and mass at equal safety factor.

4. Sensitivity to Heat Flux and Gas Pressure

The absolute results depend on two assumed loads, the crown heat flux and the peak gas pressure, so their effect on the 4 mm geometry was examined (Table V). Raising the heat flux from 0.2 to 0.4 MW/m2 raises the peak crown temperature by about 67 K for A2218 (185.5 to 252.8 °C), 113 K for AISI 4340 (224.5 to 337.7 °C) and 247 K for Ti-6Al-4V (349.0 to 596.2 °C). The safety factor falls with heat flux for all three materials; the titanium crown, which has the highest SF at 0.2 MW/m2 (3.18), has a lower SF than steel at 0.4 MW/m2 (1.63 versus 1.72), because its thermal stress and its strength derating both increase with the crown temperature. The aluminium crown falls below unity at 0.4 MW/m2. The gas pressure acts only on the bending stress and so leaves the temperatures unchanged; the safety factor of the aluminium crown is 1.43, 1.14 and 0.95 at 3, 4 and 5 MPa, so the 4 mm crown does not satisfy even SF = 1.5 anywhere in the 3–5 MPa band, while steel and titanium remain above 1.8 throughout.

Case	A2218	AISI 4340	Ti-6Al-4V
q = 0.2 MW/m^2	1.35 (186 °C)	2.59 (225 °C)	3.18 (349 °C)
q = 0.3 MW/m^2 (base)	1.14 (219 °C)	2.08 (281 °C)	2.27 (473 °C)
q = 0.4 MW/m^2	0.96 (253 °C)	1.72 (338 °C)	1.63 (596 °C)
p = 3 MPa	1.43	2.45	2.58
p = 4 MPa (base)	1.14	2.08	2.27
p = 5 MPa	0.95	1.81	2.03

Two practical points follow from the sensitivity study. First, the crown temperature of titanium is so sensitive to the heat flux (596 °C at 0.4 MW/m2) that a reliable estimate of the flux, from measurements or from a combustion simulation, is more critical for a titanium piston than for aluminium. Second, the ranking of the materials in terms of safety factor is not fixed: steel overtakes titanium at the highest heat flux. A material choice based on a single operating point could therefore be misleading, and the whole operating map of the engine (idle, part load and full load) should be considered in a complete study.

V. DISCUSSION

Aluminium. A2218 has the lowest mass at equal safety factor and the lowest crown temperature, but its strength is the lowest and falls fastest with temperature. Its low density more than compensates for the

thicker crown required. The 4 mm crown does not reach SF = 2, so for a 4 MPa peak pressure either a thicker crown of about 5.5 mm or a thermal and structural enhancement (for example crown ribs, a fillet radius to the wall, or anodised crown) is needed. Because aluminium has the greatest expansion coefficient, running clearance and ring-land design must account for the larger growth.

Steel. AISI 4340 offers the highest stiffness and a moderate crown temperature, and the thermal stress is moderate. But its density makes the piston 2.6 times as heavy as the aluminium one when the walls are of the same thickness, and the greater reciprocating mass at 7500 rpm would increase inertia loads on the pin, rod and crank. A steel piston is thus sensible only if the design can be made much thinner. In this model the skirt wall is fixed at 3.5 mm, which unfairly burdens steel; actual steel pistons use walls of 1.5–2 mm and are practical mainly in high-pressure diesel engines. A thin-wall steel piston would also further raise the crown temperature by shortening the conduction path to the cooled sides.

Titanium. Ti-6Al-4V has the greatest specific strength and its thin crown (3.5 mm) gives a mass lower than steel by 45 %. However, its very low thermal conductivity produces a crown temperature above 450 °C and a thermal stress which accounts for more than half of the total stress. Practical use would require measures to lower the crown temperature, such as oil-jet cooling of the underside, a thinner crown with ribs, or a thermal barrier coating that lowers the surface heat flux. The material is also expensive and difficult to machine, as noted in [3].

Comparison with published work. The trends obtained here for aluminium agree with the general finding of finite-element studies that the crown and the ring-groove and pin-boss regions are the most critical, and that crown temperature rises as the crown becomes thinner or as conductivity falls [4], [5]. Some studies of thin crowns report lower stress together with lower mass for thinner sections; in the present plate-type model the opposite holds for the crown, since bending stress increases as the inverse square of thickness, while the pin-boss ring behaves in the same sense. Differences of this kind arise from the load path and support conditions assumed and are a reason to check any piston simplification against a detailed model before it is used for a design decision.

Practical recommendations. The following guidelines are suggested for a designer of a small SI engine piston: (i) use an aluminium alloy unless the peak pressure or the thermal load is demonstrated to exceed its capacity; (ii) size the crown for a safety factor of at least 2 at the peak crown temperature, not at room temperature; (iii) if titanium is considered, set a design limit on crown temperature and provide underside oil cooling or a thermal barrier coating; (iv) if steel is considered, reduce the wall thickness and verify the crown temperature again, since a thinner wall lengthens the conduction path; (v) enlarge the running clearance of the aluminium piston in proportion to its higher expansion; and (vi) validate the final design with three-dimensional FEA and, where possible, with temperature measurements by thermocouples or temperature-indicating plugs on a running engine.

Selection. For this engine and these assumptions, A2218 is the best compromise: lowest mass at equal safety factor and the lowest temperatures. Where higher pressure or turbocharging push the aluminium crown beyond its strength at temperature, a forged high-strength aluminium alloy or a steel crown with thin-wall design would be the next options, and titanium would be reserved for racing applications where mass is the overriding criterion and cost is secondary.

VI. LIMITATIONS

This study is a pre-design analysis with simplifications that readers should keep in mind. The geometry omits ring grooves, fillets, the combustion bowl and the oil gallery; the plate and ring formulas cannot capture local stress concentrations, which are typically found at ring grooves and pin-boss edges and

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may double the local stress. The crown is clamped at its edge, whereas in reality it is supported by the ring zone and ribs. The heat flux and cooling coefficients are representative rather than measured; they were identical for all materials, but real heat flux depends on surface temperature, and a hotter crown would receive a lower flux. The derating curves are assumptions. Loads are static, so fatigue life, which is critical for pistons, is not evaluated. Finally, the same wall thickness was used for all materials, which favours aluminium. For these reasons, the quantitative results should be confirmed by three-dimensional coupled thermal–structural finite-element analysis (for example in ANSYS Workbench) with measured boundary conditions, and by engine tests.

VII. VALIDATION PLAN

Because the present model is deliberately simple, a validation plan is needed before any numerical value is used for a design decision. The plan has three steps. In the first, the same three materials and the same load cases are analysed in a three-dimensional model built in a CAD package (for example CATIA V5) and solved with a finite-element code (for example the steady-state thermal and static structural modules of ANSYS Workbench), using the thermal results as a body load for the structural run. The geometry should include ring grooves, fillets, the combustion bowl and the oil gallery, and the mesh should be refined at the ring grooves and pin-boss edges until the peak stress changes by less than 5 % between successive refinements. The maximum temperature and the crown stress from this model should be compared with those of Tables III and IV; agreement within about 15–20 % would indicate that the screening model is adequate for ranking materials, whereas a larger difference would show which assumption (support condition, heat flux, or stress concentration) needs revision.

In the second step, the boundary conditions are replaced with measured or simulated values. The crown heat flux can be obtained from a cycle-resolved combustion simulation or from engine heat-balance measurements, and the cooling coefficients can be calibrated so that the computed temperature at several points matches thermocouple or temperature-indicating-plug measurements on a running piston. In the third step, the pressure load is applied as a cycle rather than as a static peak so that a fatigue safety factor can be computed with the material's endurance limit at temperature, which is the governing criterion for long life. Only after these steps should a final material and crown thickness be chosen. The approach used in this paper is thus a quick first filter that narrows the candidates, not a replacement for detailed analysis.

VIII. CONCLUSION

A thermal–structural pre-design comparison of A2218, AISI 4340 and Ti-6Al-4V pistons for a 50 mm bore single-cylinder SI engine was presented. With identical geometry the peak crown temperatures were 219, 281 and 473 °C, and the thermal stress fraction rose from 19 % to 53 % from aluminium to titanium, in line with the fall in thermal conductivity. Sizing the crown for a safety factor of 2 required 5.5, 4.0 and 3.5 mm and gave masses of 87.9, 228.3 and 125.6 g, so that aluminium gives the lightest piston for this engine. Titanium needs crown cooling or coating, and steel requires thin walls to be competitive. Future work will use detailed three-dimensional models with ring grooves and a bowl crown, a fully coupled transient analysis, fatigue life prediction, thermal barrier coatings, and experimental validation by strain and temperature measurement.

APPENDIX: WORKED EXAMPLE (A2218, T = 4 MM)

The following hand calculation, using Table II and Table I, reproduces the structural quantities of the A2218 column of Table III from the thermal results (ΔTz = 6.3 K, ΔTr = 22.4 K, Tmax = 219.2 °C). The bending stress from (2) is σb = 0.75 × 4 MPa × (25/4)2 = 117.2 MPa. The thermal stress from (3) with the larger gradient (22.4 K) is σth = 74 GPa × 22 × 10-6 K-1 × 22.4 K / (2 × 0.67) = 27.3 MPa, so that

σeq = 144.4 MPa. The yield strength at 219.2 °C, derated linearly to 50 % at 300 °C, is 255 MPa × [1 − 0.5 × (219.2 − 25)/(300 − 25)] = 165 MPa, and SF = 165/144.4 = 1.14. The plate deflection is w = 0.171 × 4 MPa × (25 mm)4/(74 GPa × (4 mm)3) = 0.056 mm. For the boss, F = 4 MPa × π × (25 mm)2 = 7.85 kN, pb = 7.85 kN/(2 × 14 mm × 12 mm) = 23.4 MPa, and with ro = 10 mm, σθ = 23.4 × (100 + 49)/(100 − 49) = 68.3 MPa. The mass is 81.7 g from the three volumes (crown 7.85 cm3, wall 17.4 cm3, bosses 3.8 cm3, total 29.1 cm3) at 2810 kg/m3.

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