Discussion: "Design Improvements to a Biomass Stirling Engine Using Mathematical Analysis and 3D CFD Modeling" (Mahkamov, K., 2006, ASME J. Energy Resour. Technol., 128, pp. 203–215)

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As the designer of the engine[1] featured in Dr. Mahkamov's paper,[2] I welcome his pioneering application of CFD modeling to the Stirling cycle. The technique would have been helpful in 1996 when this engine was on the drawing board and then during testing in 2003, when absence of dynamic pressure measuring equipment hindered diagnosis of the problem he subsequently identified by modeling—flow restriction caused by a dimensional error in the crown of the power piston.

Unfortunately, however, his models do not appear to have achieved the close agreement between predicted and actual power outputs required to justify fundamental design decisions or, as in this case, a change of operating mode from the engine's original gamma layout to the less efficient alpha regime. Some explanations for the poor correlation between prediction and reality, set out below, may be found in differences between the actual engine and the data and assumptions fed into the model, a few of which I have listed below. I will leave others to seek explanations in the modeling itself.

(1) The power predictions achieved by the two methods, as shown in Figs. 7 and 15 (second-order model) or Figs. 14 and 17 3D CFD model), bear little relation to those obtained testing the real engine, undermining the claim that "results are reasonably close to those predicted." Table 1 shows that the power output predicted by the second-order model is out by a factor of 12:1, for the original gamma version of the engine, and 2.5:1 for the engine in alpha mode. The CFD model is slightly better, with a 3.5:1 error in gamma mode and 1.45:1 in alpha mode.

(2) It is gratifying that Dr. Mahkamov's models confirm my prediction that a 10% gain would result from incorporating what I described as "biaxially asymmetric piston motion" (Mahkamov's Ref. 2). However, as I did not actually employ any lateral offset in this prototype—just a very short rod-to-stroke ratio of 1.263:1—I was puzzled to read that "the numerical results… are presented for the engine with biaxial asymmetric piston motion."

(3) Describing the rocker beam seal assembly, the author writes,

Table 1 Power output predictions

Engine mode	Test bed (kW)	Model predictions: Second-order mathematical model (kW)	Model predictions: 3D CFD model (kW)
Gamma	3.09[a]	0.25	0.737
	(16 bar charge)	(2 MPa max)	(2.2 Mpa max)
	4 Hz–240 rpm	4 Hz	3.33 Hz–200 rpm
Alpha	7.5[b]	2.97	3.87
	(15 bar charge)	(2.1 MPa max)	(2 MPa max)
	6.7 Hz–400 rpm	7 Hz	5 Hz–300 rpm

[a]Test 17 October 2003 in presence of R. F. Kinnersly and D.W. Hislop.
[b]Personal communication from D. W. Hislop, 19 May 2004.

"*it is thought* that the frictional speed and distance are low compared to those for conventional rod seals." This is not a matter of conjecture: seal face area and specific pressure are, respectively, 50% and 40% of those for conventional systems, while sliding distance (and so speed) is less than 20%. During testing the unloaded engine would run on for ten minutes after the burner was extinguished.

(4) It is stated that the 9720 cooler ducts are 0.5 mm diameter. In fact they are "U" section slots of 0.45 mm width and 0.65 mm radial depth, so the cooler surface area and flow cross-sectional area are, respectively, 27% and 39% larger in the real engine than the author presumes.

(5) I am concerned that this error may have been eclipsed by an even larger misunderstanding of the cooler assembly I designed for the engine. On p. 204 it appears that Dr. Mahkamov has taken "a set of six annular cylindrical coolers (13)" and then assumed that "The casing of every cylindrical cooler houses 9720 tubes…." In fact the total number of slots in the entire cooler assembly of six elements is 9720.

What we now need to know is if this sixfold error was actually fed into the model. If this was the case it would have had fundamental and damaging effects on the predicted performance of an engine in which it was a design priority to squeeze out dead space wherever possible, within the constraints of prototype production. Instead of having a total dead space of 181 cc in the six cooler elements, Dr. Mahkamov's model of my engine would have a dead space of more than one liter in the coolers. There would be a severe reduction in gas speed and consequently in heat transfer; surface area of the individual cooler elements would have gone up from what I believe was an impressive 1.268 m^2 to an unfeasibly large area of more than 7.6 m^2—surely beyond any reasonable design target!

Clarification is needed here. Was this merely another slip in the drafting of the text—of which there are unfortunately too many to list in a short discussion paper—or was it a genuine misunderstanding on the part of the author, whose knowledge of the physical details of the engine does seem at times less exact than one might hope?

A sixfold error in this particular input to the models might help to explain why his model outputs fall so far short of actual test data—the 12fold error in the second-order model, and 3.5-fold error in the CFD model, as shown in the Table 1 and referred to in paragraph 1, above.

[1]US patents 6205782; 6296417; 5345765; 5309715; and GB 3032855T.
[2]Mahkamov, K., 2006, "Design Improvements to a Biomass Stirling Engine Using Mathematical Analysis and 3D CFD Modeling." ASME J. Energy Resour. Technol., **128**, pp. 203–215.