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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Some very impressive modeling work lies behind this paper, but leaves the reader with many questions which Dr. Mahkamov may care to answer.

We are told that the Biomass Stirling Engine "has been experimentally tested" and that "experimental data, which is available (where?) of the mechanical brake power output and speed, indicates that"—these "are *reasonably close* to those predicted by using the 3D CFD model." Could the author provide the figures so that we can judge for ourselves how close is "reasonably close?"

One observes that, with fluid flow losses included, the γ engine achieves an inferior predicted performance to the modified α, whether using the second-order model, or the 3D CFD model.

Mode	2nd order	3D CFD
γ	250 W	737 W
	4 Hz	3.33 Hz
α	2.970 W	3.870 W
	7 Hz	5 Hz

So the α mode power output would seem to be 6.8 times better than the γ using the results of the second-order model, or 3.5 times better using the 3D CFD model. However, this is misleading; we are not comparing like with like.

The γ engine had apparently a simple geometrical error which caused much loss of power owing to the crown of the power piston restricting flow as it approached TDC. Furthermore the γ engine model is run with a 31% porosity regenerator, while the α engine is run with an improved 40% regenerator. Did the author run his two programs for the γ engine with the crown restriction removed and with 40% porosity? Might he be able to provide the figures so that α and γ modes can be compared on a level playing field?

On the basis that the pressures, *P7*, *P5*, and *P1* in Fig. 13 correspond, respectively, to pressures *P2* (of *P2V2*), *P3* (of *P3V3*) and *P1* of (*P1V1*) of Fig. 14, then one can make an estimate for how much the gas entrapment is costing the γ engine in power output. From Fig. 13 it can be seen that for some 25° either side of the 180° crank angle (where the power piston crown is near TDC) there is a significant difference in pressure between *P7* and *P5* and the gas is restricted in its passage from compression space 5 to compression space 6 (see Fig. 1).

Transferring this pressure difference so as to modify the anticlockwise compression loop *P2V2* one is able to deduce that the γ engine output will be raised at least from 0.737 to 1.77 kW. Does Dr. Mahkamov feel this is a reasonable estimate?

As a measure of goodness (proportional to the Beale number) one can compare the outputs in the modified Fig. 14 (γ mode) and Fig. 17 (α mode) on the basis of kW/(Charge pressure × rpm × power swept volume). In changing from γ mode to α mode the power swept volume changes. In γ mode this volume is provided by the power piston (4 in Fig. 1) together with a small contribution from the unbalanced stanchion of the displacer piston. In α mode it is the larger expansion piston that becomes effectively the power piston. Using the outputs from the 3D CFD model and a nominal charge pressure of 15 bar then one obtains

	kW	rpm	Power swept volume (liter)	kW / Bar × liters × rpm
γ	1.77	200	2.46	0.24×10^{-3}
α	3.87	300	3.42	0.25×10^{-3}

It would seem that using the "measure of goodness factor" in the right-hand column, not very much has been gained in moving from γ to α mode. Indeed, much has been lost in terms of the mechanical soundness of the engine crank shaft gas loads, etc.

The value of 0.25×10^{-3} kW bar^{-1} liter^{-1} rpm^{-1} seems to be low. Might the 3D CFD model be significantly underestimating the performance?

