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EVALUATION OF A STIRLING ENGINE HEATER BYPASS WITH THE NASA LEWIS

NODAL-ANALYSIS PERFORMANCE CODE

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E-3113

SUMMARY

In support of the U.S. Department of Energy's Stirling Engine Highway Vehicle Systems program, the NASA Lewis Research Center investigated whether bypassing the P-40 Stirling engine heater during regenerative cooling would improve the engine thermal efficiency. The investigation was accomplished by using the Lewis nodal-analysis Stirling engine computer model.

Bypassing the P-40 Stirling engine heater at full power resulted in a rise in the indicated thermal efficiency from 40.6 to 41.0 percent. For the idealized (some losses not included) heater bypass that was analyzed, this benefit is not considered significant.

The heater-tube and regenerator lengths were optimized for the heater-bypass configuration. The optimization of the regenerator length did not result in a significant improvement in efficiency; the optimization of the heater-tube length resulted in an increase in the indicated thermal efficiency from 41.0 to 42.0 percent. These results suggest that optimizing the entire engine may produce a small benefit for the heater-bypass concept.

The heater-bypass concept was investigated at part-power conditions, controlled by pressure variation and phase-angle variation. At part-power conditions the heater-bypass concept produced a significant benefit in engine performance only when the power was reduced by varying the Stirling engine phase angle. However, for the P-40 Stirling engine, part-load efficiencies were much lower for phase-angle variation than for pressure variation.

INTRODUCTION

This work was done in support of the U.S. Department of Energy (DOE) Stirling Engine Highway Vehicle Systems program. The NASA Lewis Research Center, through interagency agreement DEAI01-77CS51040 with DOE, is responsible for management of the project under the program direction of the DOE Office of Transportation Systems, Heat Engine Propulsion Division.

The Lewis Stirling Engine Project Office evaluates various innovative concepts with the use of the Lewis nodal-analysis Stirling engine computer simulation. One such concept is bypassing the Stirling engine heater during a portion of the cycle to improve engine efficiency. The P-40 Stirling engine was selected as a baseline engine for this study because of the familiarity at Lewis with the engine and its computer simulation. The P-40, described in reference 1, is a double acting, four-cylinder Stirling engine developed by United Stirling of Sweden, capable of producing approximately 40 kW.

The heater-bypass concept involves the rerouting of the working fluid around the Stirling engine heater during a portion of the cycle. It was thought to be inefficient for the working fluid to flow through the heater prior to being cooled. A similar statement can be hypothesized for flow through the Stirling engine cooler prior to the heating of the working fluid.

Rallis, Urieli, and Berchowitz, and Wang have used an ideal adiabatic Stirling cycle model to simulate a Stirling engine with heater/cooler bypass. The ideal adiabatic Stirling cycle model (sometimes referred to as the ideal pseudo-Stirling cycle model) assumes that the engine operates with adiabatic expansion and compression, and constant volume heating and cooling of the working fluid. Their results are described in references 2 and 3. For operating conditions similar to those of the P-40 Stirling engine, the ideal adiabatic Stirling cycle model predicted a small benefit in engine performance when a heater/cooler bypass was used.

The purpose of this study is to evaluate the heater-bypass concept for a P-40 Stirling engine with the use of a more sophisticated nodal-analysis computer model. This evaluation was accomplished by first modifying the P-40 computer simulation to include an idealized heater bypass. The performance with the heater bypass was then compared with the baseline engine performance at various conditions. Finally the computer results were analyzed in order to evaluate the heater-bypass concept.

ANALYSIS

Development of Heater-Bypass Computer Model

The performance predictions for the heater-bypass concept were accomplished by first modifying the P-40 Stirling engine computer model. This nodal-analysis computer code, developed by NASA Lewis, is described in reference 4.

The Stirling engine code was modified to include a separate passage around the Stirling engine heater. A valve assembly was located between the heater and the regenerator to regulate whether the working fluid flowed through the heater or the bypass passage. The valve assembly was placed at that site, instead of between the expansion space and the heater, because the lower working-fluid temperatures at the heater-regenerator interface would benefit valve life in an actual application. The expansion-space ends of the heater and bypass passage were always open.

Several assumptions were made to simplify the modeling process:

(1) The bypass passage had a relatively small volume of 1.6 cm^3 in order to decrease the thermodynamic effect due to volume change, and was assumed to have a simple tube shape.

(2) The bypass passage was idealized to be frictionless and adiabatic.

(3) The valve assembly had no losses associated with it.

(4) Methods used to calculate properties of the working fluid in the bypass passage were identical to those used at other nodes throughout the engine.

The evaluation of the heater-bypass concept depends on the portion of the engine cycle and the duration the heater is bypassed. Computer runs were made by using different portions of the cycle to bypass the heater. The results showed that engine performance was at or near its maximum when bypassing the heater during regenerative cooling. Regenerative cooling refers to the working fluid being cooled in the regenerator when the gas flows in the direction from the expansion space to the compression space. Bypassing the heater during regenerative cooling can be achieved by using simple check valves for the valve assembly. Because of this simplicity of regulating the heater bypass and near-maximum performance, the optimum time to bypass the heater is during regenerative cooling.

The engine for which the heater is bypassed during regenerative cooling, with check valves located at the heater-regenerator interface, will be referred to as the heater-bypass configuration. The heater-bypass configuration will be evaluated in this study, and it is shown in figure 1. The P-40 Stirling engine will be referred to as the baseline configuration.

A measure of the Stirling engine computer code precision is the calculated error in the energy balance. The error in the energy balance is the difference between the predicted energy (engine power and heat) out of the engine and the predicted energy (heat) into the engine, and it relates to either the accuracy of the method used to model the engine or the amount of computer time given for the solution to settle out. For all results obtained in this investigation, the errors in the energy balance were less than 1 percent.

Procedure for Performance Comparisons

The evaluation of the heater-bypass configuration involved three phases: comparison with the baseline configuration at full power, optimization of key parameters, and comparison with the baseline configuration at part power.

Evaluation of heater-bypass configuration at full power. - This evaluation consisted of comparing performance of the heater-bypass configuration with that of the baseline configuration at the design conditions. These design conditions are at the full-power operating point (15 MPa, 4000 rpm) and are shown in appendix A.

Heater-bypass configuration heater-tube and regenerator length optimization. - The purpose of this study was to attempt to improve heater-bypass configuration performance by optimizing two Stirling engine parameters: the heater-tube length and regenerator length. The Stirling engine heater is made up of 18 tubes per working space and is responsible for the majority of the energy input to the engine. The regenerator is a thermal sponge used to store energy that would otherwise be lost in the Stirling engine cooler.

An optimum length was defined as the length of the component that would result in the maximum indicated thermal efficiency. The optimal heater-tube and regenerator lengths were also predicted by using the computer code for the baseline configuration in order to identify a true shift in the optimal lengths for the heater-bypass concept. The heater-tube and regenerator lengths were optimized independently of one another, and no other changes were made to engine geometry. The optimization computer predictions were made at the full-power design conditions shown in appendix A.

Evaluation of heater-bypass configuration at part power. - The purpose of this study was to evaluate the heater-bypass configuration at part power. Part power was obtained by two different methods: decreasing the engine mean pressure and increasing the phase angle between the expansion-space and compression-space volumes. For full power in the P-40 Stirling engine, the engine mean working-space pressure was 15 MPa, and the volume phase angle was 90°. For these part-power evaluations, the engine speed was held constant at 4000 rpm; other operating conditions were maintained as shown in appendix A.

Although it is not possible to vary the volume phase angle in the actual P-40 Stirling engine, these results will be useful for a Stirling engine with variable-phase-angle power control. Also in this evaluation, the volume phase angle was varied, while the expansion-space and compression-space volume amplitudes remained constant. For a configuration such as a piston-displacer Stirling engine, the volume amplitudes will change when the volume phase angle is increased.

RESULTS AND DISCUSSION

Evaluation of Heater-Bypass Configuration at Full Power

Comparisons of performance for the heater-bypass configuration with that for the baseline configuration are shown in figures 2 to 6 and table I. Both heater-bypass and baseline configurations are defined in the section ANALYSIS. Figure 2 shows the working gas flow rate at the interface of the heater tubes and the expansion-space manifold. Positive flow is defined as flow into the heater tubes. At a crank angle of 0°, the expansion-space volume was a minimum, with the compression-space volume lagging by 90°. For the baseline configuration, from crank angles of 152° to 350°, the working fluid at the interface of the heater tubes and expansion-space manifold flowed toward the heater. For the heater-bypass configuration, the majority of the working fluid bypassed the heater during this period. A relatively small amount of gas flowed into and out of the heater during heater bypass and was caused by differences in pressure between the gas in the heater and gas in the expansion-space manifold.

Figure 3 shows energy transferred from the heater-tube walls to the working fluid for 1° increments of the crank shaft rotation. For the baseline configuration, 68 percent of the total heater heat transfer occurred when the working gas was flowing toward the regenerator. This was primarily due to the drop in working-gas temperature during the expansion process, which caused a higher temperature difference (between the heater wall and working fluid) for flow in this direction. For the heater-bypass configuration, 20 percent of the total heater heat transfer occurred during heater bypass because of working gas flowing into and out of the open end of the heater tubes (see fig. 2).

Figure 4 shows the regenerator hot-end working-gas temperature as a function of the crank angle. For the heater-bypass configuration, the average regenerator hot-end gas temperature was 84 °C lower than that for the baseline configuration. The large, sudden changes in gas temperature occurred during flow reversal. The average gas temperature for the heater-bypass configuration was lower because the working fluid was unable to pick up energy during heater bypass. There was no significant difference in average gas temperature at the cold end of the engine for both configurations. This lower regenerator hot-end gas temperature and ineffectiveness of the heater caused a drop in the expansion-space average gas temperature of 55 °C as shown in figure 5. Figure 6 shows the expansion-space gas pressure as a function of the crank angle for both configurations. The pressure waves are similar for both configurations.

Table I shows the numerical results from the comparison of the heater-bypass configuration with the baseline configuration. Indicated power dropped 3.7 percent (2.3 kW) when the heater bypass was used. This drop in the engine power was primarily caused by the drop in the expansion-space gas temperature, not offset by the 2.2-kW decrease in the engine flow friction loss. The indicated thermal efficiency did increase from 40.6 to 41.0 percent. The primary reason for this slight increase in the indicated thermal efficiency was the 21.3-percent decrease in the flow friction loss. The heat into the engine dropped by 4.5 percent. The heat out of the engine decreased by 4.9 percent primarily because of the lower appendix-gap pumping loss for the heater-bypass configuration. The appendix-gap pumping loss decreased with the decreasing hot-end gas temperature.

The reductions in the engine power and appendix-gap pumping loss due to the decrease in the hot-end gas temperature resulted in a reduction in the heat into the engine. These changes in the engine power and heat into the engine offset each other, causing the indicated thermal efficiency to be rather insensitive to a decrease in the hot-end gas temperature. For this reason, the indicated thermal efficiency improved slightly with the heater bypass primarily because of the lower flow friction loss, shown in table I. Because the heater-bypass passage would not be frictionless in reality, this increase in indicated thermal efficiency would not be as large for actual hardware.

As mentioned in the INTRODUCTION, Rallis, Urieli, and Berchowitz (ref. 2) investigated the heater- and cooler-bypass concept with an ideal adiabatic Stirling cycle model. With operating conditions similar to those for the P-40 Stirling engine, their idealized Stirling cycle model results showed that the thermal efficiency improved from 56.5 to 57.5 percent. This included the effects of both bypassing the heater during constant-volume cooling and bypassing the cooler during constant-volume heating.

Wang (ref. 3) conducted a similar study for Oak Ridge National Laboratory using the Rallis model modified for investigating only heater bypass. The thermal efficiency improved from 50.3 to 50.8 percent. The difference in thermal efficiency from these two studies appears consistent with the results shown in table I, even though the Rallis model predicted that power would remain constant. Both these idealized Stirling cycle models demonstrate that, as the regenerator effectiveness drops, the benefit for the heater-bypass concept increases.

Heater-Bypass Configuration Heater-Tube and Regenerator Length Optimization

<u>Heater-tube length optimization</u>. - From the evaluation of the heater-bypass configuration at full power (table I), note that, when a heater bypass was used, the heat transfer in the regenerator was down by 15 percent, and the expansion-space gas temperature decreased by 55 °C. An attempt was made to improve heater-bypass configuration performance by optimizing the heater-tube and regenerator lengths. Results for the optimization of the heater-tube length are shown in figures 7 and 8 and table II.

Figure 7 shows indicated power as a function of the heater-tube length for the heater-bypass and baseline configurations. Indicated power reached a maximum when the desired effect of the increasing hot-end gas temperature equaled the negative effects of the increasing dead volume and flow friction loss. For the baseline configuration, the indicated power was a maximum of 61.3 kW at the reference length of 280.7 mm. For the heater-bypass configuration, the indicated power increased from 59.0 kW at 280.7 mm to a maximum of 61.0 kW at 449.1 mm.

Figure 8 shows the indicated thermal efficiency as a function of the heater-tube length. The maximum indicated thermal efficiency for the baseline configuration was 40.6 percent at the reference length of 280.7 mm. For the heater-bypass configuration the indicated thermal efficiency increased from 41.0 percent at the reference length to 42.0 percent at 533.3 mm. The optimum heater-tube length for the heater-bypass configuration was longer than that for the baseline configuration primarily because of the lower hot-end gas temperature and the lower flow friction loss for the heater-bypass configuration, for equivalent heater-tube lengths. This is explained in more detail in appendix B.

Table II lists the numerical data for the variation of the heater-tube length for both configurations. The 90-percent increase in the optimum heater-tube length for the heater-bypass configuration caused a 61 °C rise in the expansion-space gas temperature and a 1.0-kW rise in the flow friction loss.

<u>Regenerator length optimization</u>. - Results for the optimization of the regenerator length are shown in figures 9 and 10 and table III. Figure 9 shows indicated power as a function of the regenerator length for the heater-bypass and baseline configurations. In both cases, as the regenerator length increased, power decreased primarily because of the increase in both the engine dead volume and the flow friction loss. The effects of the increase in the expansion-space gas temperature and decrease in the compression-space gas temperature were small.

Figure 10 shows the indicated thermal efficiency as a function of the regenerator length for the heater-bypass and baseline configurations. The maximum indicated thermal efficiency predicted for the baseline configuration occurred at a regenerator length of 45 mm; the maximum indicated thermal efficiency for the heater-bypass configuration occurred at a regenerator length of 48 mm. The predicted optimal regenerator length for the baseline configuration was different from the reference length (length of actual regenerator in the P-40 engine) of 39 mm. This was possibly due to tradeoffs in the design or differences between the NASA Lewis computer code and the simulation used to design the P-40 Stirling engine.

For the heater-bypass configuration, increasing the regenerator length from the reference length of 39 mm to 48 mm resulted in the increase of the indicated thermal efficiency from 41.0 to 41.2 percent. The change in the indicated thermal efficiency is not considered significant compared to the change in the indicated thermal efficiency when the heater-tube length is optimized.

Table III lists the numerical data for the variation in the regenerator length for both configurations. The pressure ratios are shown to indicate how dead volume affects the amplitude of the gas pressure.

Evaluation of Heater-Bypass Configuration at Part Power

Results for the part-power comparisons between the heater-bypass configuration and the baseline configuration are shown in figure 11 and tables IV and V. Figure 11 shows the indicated thermal efficiency as a function of the percentage of maximum indicated power for both configurations. The indicated power was reduced by two different means. One method reduced the engine mean working-space pressure; the other method increased the phase angle between the expansion-space and compression-space volumes. The engine speed was held constant at 4000 rpm for all points. Tables IV and V list the results for varying the pressure and varying the phase angle, respectively.

When the indicated power was reduced by decreasing the engine mean working-space pressure, the difference in the indicated thermal efficiency between the heater-bypass and baseline configurations was 0.4 percentage points at 15 MPa and 0.9 percentage points at 5 MPa as shown in figure 11. For both configurations, the indicated thermal efficiency remained relatively constant as the engine mean working-space pressure was reduced.

When engine power was reduced by increasing the volume phase angle, the difference in the indicated thermal efficiency for the two configurations was 0.4 percentage points at a volume phase angle of 90° and 3.7 percentage points at a volume phase angle of 150°. This is a significant improvement for the heater-bypass configuration at part power. However, the absolute values of the indicated thermal efficiency for both configurations decreased significantly as the volume phase angle increased. This was primarily caused by increasing flow friction losses as the volume phase angle increased, as seen in table V.

Figure 11 shows that the difference in indicated thermal efficiency between the two configurations increased as the power was reduced; this was due to the flow friction becoming a larger loss relative to the indicated power. The lower flow friction losses for the heater-bypass configuration lessened the impact of the increasing flow friction loss relative to indicated power. Because other methods of reducing power, such as engine stroke reduction or dead volume increase, have relatively smaller heater flow friction losses than does increasing the volume phase angle, they would be expected to show less benefit with the heater-bypass concept at part-power conditions.

CONCLUDING REMARKS

The performance of a P-40 Stirling engine with the working-fluid flow bypassing the heater during regenerative cooling, the heater-bypass configuration, was compared with that for a standard P-40 Stirling engine, the baseline configuration, by using the NASA Lewis nodal-analysis Stirling engine computer simulation. When a heater bypass was used at full power, the indicated thermal efficiency did not significantly improve. Furthermore, this did not include the effects of flow friction in the bypass passage and losses in the check valve assembly.

An attempt was made to improve the performance of the heater-bypass configuration by optimizing the heater-tube length and regenerator length. Judging from these optimization results, a small improvement in the indicated thermal efficiency may be gained by optimizing all key Stirling engine dimensions. This would require the use of a Stirling engine design code.

At part-power conditions, the heater-bypass configuration showed no significant benefit except when the power was varied by changing the volume phase angle. However, for the P-40 Stirling engine, this method of varying the power caused the thermal efficiency to decrease significantly compared with varying the power by reducing the engine mean working-space pressure.

SUMMARY OF RESULTS

The performance of the heater-bypass configuration was compared with that for the baseline configuration by using the NASA Lewis nodal-analysis Stirling engine computer simulation. The major results from the evaluation of the heater-bypass concept for a P-40 Stirling engine are as follows:

1. For the engine design point, bypassing the heater during regenerative cooling resulted in the decrease of indicated power by 3.7 percent. This was primarily due to a 55 °C decrease in the expansion-space gas temperature.

2. For the engine design point with heater bypass, the indicated thermal efficiency increased from 40.6 to 41.0 percent primarily because of a 21.3-percent decrease in the engine flow friction loss.

3. The optimum heater-tube length for the heater-bypass configuration (P-40 Stirling engine with the heater bypassed during regenerative cooling) was 90 percent longer than that for the baseline configuration (standard P-40 Stirling engine). The indicated thermal efficiency improved from 41.0 to 42.0 percent for the 90-percent increase in the heater-tube length.

4. The optimum regenerator length for the heater-bypass configuration was 7 percent longer than that for the baseline configuration; the effect on the indicated thermal efficiency was insignificant.

5. At a low power point obtained by reducing the engine mean working-space pressure, the use of the heater bypass improved the indicated thermal efficiency from 39.5 to 40.4 percent.

6. At a low power point obtained by increasing the Stirling engine volume phase angle, the use of the heater bypass improved the indicated thermal efficiency from 29.4 to 33.1 percent.

APPENDIX A - OPERATING CONDITIONS

The following is a list of the Stirling engine computer model operating conditions:

(1) The real gas state equation was used.

(2) Twenty-five engine cycles were used to reach a converged solution.

(3) The working gas was pure hydrogen.

(4) A second pass was not made through the calculations.

(5) No calibration factors were input into the computer model.

(6) Engine mean working-space pressure was 15 MPa (except for the pressure variation investigation).

(7) Engine angular speed was 66.67 Hz.

(8) The expansion-space wall temperature was 720 °C.

(9) The outside-wall temperature of the heater tubes was 720 °C.

(10) The cooling-water flow rate per cylinder was 1.0 liter/sec.

(11) The cooling-water inlet temperature was 50 °C.

(12) Coolant properties were used to calculate the cooler-tube inside-wall temperature.

(13) Heater-tube wall thermal resistance was taken into account.

(14) Engine geometry was set as shown in reference 1.

APPENDIX B - OPTIMIZATION OF HEATER-TUBE LENGTH

The purpose of this appendix is to explain why the optimum heater-tube length is longer for the heater-bypass configuration than for the baseline configuration. As the heater-tube length is increased, the flow friction loss increases linearly (see fig. 12), the engine dead volume increases, and the expansion-space gas temperature increases exponentially (see fig. 13). The expansion-space gas temperature follows the heater-tube gas temperature, which increases exponentially because of heat transfer from the heater-tube walls at constant temperature.

For both configurations, as the heater-tube length is increased, the compression-space gas temperature and the compression-space work decrease slightly because of the increase in the engine dead volume.

The flow friction loss increases at a lower rate for the heater-bypass configuration than for the baseline configuration. This is due to the frictionless bypassing of the heater during approximately half the cycle. In reality there would be flow friction in the bypass passage but not as much as the flow friction in the Stirling engine heater. The engine dead volume increases at the same rate for both configurations. Because of the higher temperature difference between the working fluid and the heater-tube wall, the hot-end gas temperature increases at a higher rate for the heater-bypass configuration.

As the heater-tube length is increased from the reference value (heater-tube length for the actual P-40 Stirling engine), the effects of the heater-bypass configuration flow friction loss and hot-end gas temperature cause the indicated power to increase while the indicated power for the baseline configuration decreases. The appendix-gap pumping loss increases with the increasing hot-end gas temperature for both configurations. The effect of the engine dead volume increase offsets the increasing appendix-gap pumping loss to cause the heat out of the engine to remain approximately constant for both configurations.

When increasing the heater-tube length, the indicated thermal efficiency reaches a maximum approximately when the indicated power reaches a maximum. The conclusion of this analysis is that the optimum heater-tube length is longer for the heater-bypass configuration than for the baseline configuration because the hot-end gas temperature and the flow friction losses are lower for the heater-bypass configuration, for equivalent heater-tube lengths.

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TABLE I. - COMPARISON OF BASELINE AND HEATER-BYPASS CONFIGURATIONS AT FULL POWER

[Engine speed, 4000 rpm; mean working-space pressure, 15 MPa.]

Predicted engine performance	Baseline configuration	Heater-bypass configuration
Indicated thermal efficiency, percent	40.6	41.0
Indicated power, kW	61.3	59.0
Heat into engine,[a] kW	150.9	144.1
Heat out of engine,[a] kW	90.4	85.9
Regenerator heat rate,[b] kW	692.9	591.1
Appendix-gap pumping loss, kW	8.3	5.3
Total flow friction loss, kW	10.2	8.0
Pressure ratio	1.69	1.68
Average expansion-space gas temperature, °C	636	581
Average compression-space gas temperature, °C	90	90
Average regenerator hot-end gas temperature, °C	601	516
Average regenerator cold-end gas temperature, °C	117	116

[a]Does not include mechanical and auxiliary losses.
[b]Regenerator heat rate is heat stored in regenerator during regenerative cooling.

TABLE II. - OPTIMIZATION OF HEATER-TUBE LENGTH FOR BASELINE AND HEATER-BYPASS CONFIGURATIONS

[Engine speed, 4000 rpm; mean working-space pressure, 15 MPa.]

Predicted engine performance	Heater-tube length,[a] mm							
	196.5	280.7	364.9	280.7	364.9	449.1	533.3	617.5
	Baseline configuration			Heater-bypass configuration				
Indicated thermal efficiency, percent	40.4	40.6	40.4	41.0	41.7	41.9	(42.01) 42.0	(41.97) 42.0
Indicated power, kW	60.8	61.3	60.3	59.0	60.6	61.0	60.7	60.0
Heat into engine,[b] kW	150.3	150.9	149.4	144.1	145.5	145.3	144.4	143.0
Heat out of engine,[b] kW	90.4	90.4	90.0	85.9	85.8	85.4	85.0	84.4
Regenerator heat rate,[c] kW	665.5	692.9	709.3	591.1	622.6	645.9	665.2	682.0
Appendix-gap pumping loss, kW	6.7	8.3	9.1	5.3	6.9	7.9	8.5	8.8
Total flow friction loss, kW	9.6	10.2	10.9	8.0	8.4	8.7	9.1	9.4
Pressure ratio	1.71	1.69	1.67	1.68	1.67	1.66	1.64	1.63
Average expansion-space gas temperature, °C	605	636	653	581	611	629	642	651
Average compression-space gas temperature, °C	91	90	90	90	89	88	87	87

[a]The P-40 engine design heater-tube length is 280.7 mm.
[b]Does not include mechanical and auxiliary losses.
[c]Regenerator heat rate is heat stored in regenerator during regenerative cooling.

TABLE III. - OPTIMIZATION OF REGENERATOR LENGTH FOR BASELINE AND HEATER-BYPASS CONFIGURATIONS

[Engine speed, 4000 rpm; mean working-space pressure, 15 MPa.]

(a) Baseline configuration

Predicted engine performance	Regenerator length,[a] mm							
	20	35	39	43	45	48	50	55
Indicated thermal efficiency, percent	37.9	40.4	40.6	40.7	40.8	40.7	40.7	40.4
Indicated power, kW	69.0	63.2	61.3	59.3	58.3	56.9	55.9	53.4
Heat into engine,[b] kW	185.3	156.5	150.9	145.6	143.2	139.6	137.4	132.0
Heat out of engine,[b] kW	115.2	94.3	90.4	86.9	85.4	83.2	81.8	78.8
Regenerator heat rate,[c] kW	612.7	679.9	692.9	704.7	710.1	717.3	722.7	734.0
Appendix-gap pumping loss, kW	10.5	8.8	8.3	7.8	7.6	7.3	7.0	6.5
Total flow friction loss, kW	7.8	9.7	10.2	10.7	11.0	11.4	11.6	12.2
Pressure ratio	1.89	1.72	1.69	1.66	1.64	1.62	1.61	1.58
Average expansion-space gas temperature, °C	618	633	636	638	640	641	642	645
Average compression-space gas temperature, °C	105	92	90	89	88	87	86	85

(b) Heater-bypass configuration

Predicted engine performance	Regenerator length,[a] mm										
	20	30	35	38	39	45	47	48	52	55	60
Indicated thermal efficiency, percent	37.3	40.1	40.7	40.9	41.0	41.2	(41.21) 41.2	(41.22) 41.2	(41.18) 41.2	41.1	40.9
Indicated power, kW	65.8	62.8	60.8	59.5	59.0	56.3	55.4	55.0	53.2	51.8	49.6
Heat into engine,[b] kW	176.4	156.7	149.5	145.5	144.1	136.8	134.5	133.4	129.1	126.0	121.1
Heat out of engine,[b] kW	108.6	94.7	89.5	86.7	85.9	81.3	79.9	79.3	76.9	75.2	72.8
Regenerator heat rate,[c] kW	502.5	555.8	576.4	587.7	591.1	610.6	616.5	619.4	630.5	638.0	649.8
Appendix-gap pumping loss, kW	5.5	5.5	5.4	5.4	5.3	5.3	5.2	5.2	5.1	5.0	4.9
Total flow friction loss, kW	5.9	7.0	7.6	7.9	8.0	8.7	9.0	9.1	9.6	9.9	10.5
Pressure ratio	1.89	1.77	1.72	1.69	1.68	1.64	1.62	1.62	1.59	1.57	1.54
Average expansion-space gas temperature, °C	559	573	578	580	581	587	589	590	594	596	599
Average compression-space gas temperature, °C	104	94	92	90	90	87	87	86	85	83	83

[a]The P-40 engine design regenerator length is 39.0 mm.
[b]Does not include mechanical and auxiliary losses.
[c]Regenerator heat rate is heat stored in regenerator during regenerative cooling.

TABLE IV. - COMPARISON OF BASELINE AND HEATER-BYPASS CONFIGURATIONS AT PART POWER FOR VARIOUS PRESSURES

[Engine speed, 4000 rpm; volume phase angle, 90°.]

Predicted engine performance	Baseline configuration			Heater-bypass configuration		
	Engine mean pressure, MPa					
	5.0	10.0	15.0	5.0	10.0	15.0
Indicated thermal efficiency, percent	39.5	41.1	40.6	40.4	41.8	41.0
Indicated power, kW	22.9	43.5	61.3	22.7	42.4	59.0
$\frac{\text{Indicated power}}{\text{Maximum indicated power}}$, percent	37.4	71.0	100.0	38.4	71.8	100.0
Heat into engine,[a] kW	58.1	106.0	150.9	56.0	101.5	144.1
Heat out of engine,[a] kW	35.5	63.1	90.4	33.7	59.8	85.9
Regenerator heat rate,[b] kW	274.1	499.7	692.9	235.8	427.6	591.1
Appendix-gap pumping loss, kW	3.1	6.1	8.3	2.6	4.2	5.3
Total flow friction loss, kW	4.8	7.7	10.2	3.8	6.1	8.0
Pressure ratio	1.676	1.682	1.688	1.671	1.677	1.683
Average expansion-space gas temperature, °C	652	643	636	611	594	581
Average compression-space gas temperature, °C	66	79	90	66	78	90

[a]Does not include mechanical and auxiliary losses.
[b]Regenerator heat rate is heat stored in regenerator during regenerative cooling.

TABLE V. - COMPARISON OF BASELINE AND HEATER-BYPASS CONFIGURATIONS AT PART POWER FOR VARIOUS VOLUME PHASE ANGLES

[Engine speed, 4000 rpm; mean working-space pressure, 15 MPa.]

Predicted engine performance	Baseline configuration			Heater-bypass configuration		
	Stirling volume phase angle, deg					
	90.0	130.0	150.0	90.0	130.0	150.0
Indicated thermal efficiency, percent	40.6	37.7	29.4	41.0	39.8	33.1
Indicated power, kW	61.3	46.0	25.7	59.0	46.8	28.3
$\frac{\text{Indicated power}}{\text{Maximum indicated power}}$, percent	100.0	75.0	42.0	100.0	79.2	47.9
Heat into engine,[a] kW	150.9	121.9	87.6	144.1	117.6	85.5
Heat out of engine,[a] kW	90.4	76.6	62.5	85.9	71.9	58.4
Regenerator heat rate,[b] kW	692.9	893.4	990.1	591.1	800.7	926.9
Appendix-gap pumping loss, kW	8.3	7.3	6.8	5.3	5.1	5.3
Total flow friction loss, kW	10.2	14.9	16.6	8.0	11.8	13.2
Pressure ratio	1.69	1.45	1.38	1.68	1.45	1.37
Average expansion-space gas temperature, °C	636	665	682	581	614	644
Average compression-space gas temperature, °C	90	83	74	90	81	73

[a]Does not include mechanical and auxiliary losses.
[b]Regenerator heat rate is heat stored in regenerator during regenerative cooling.

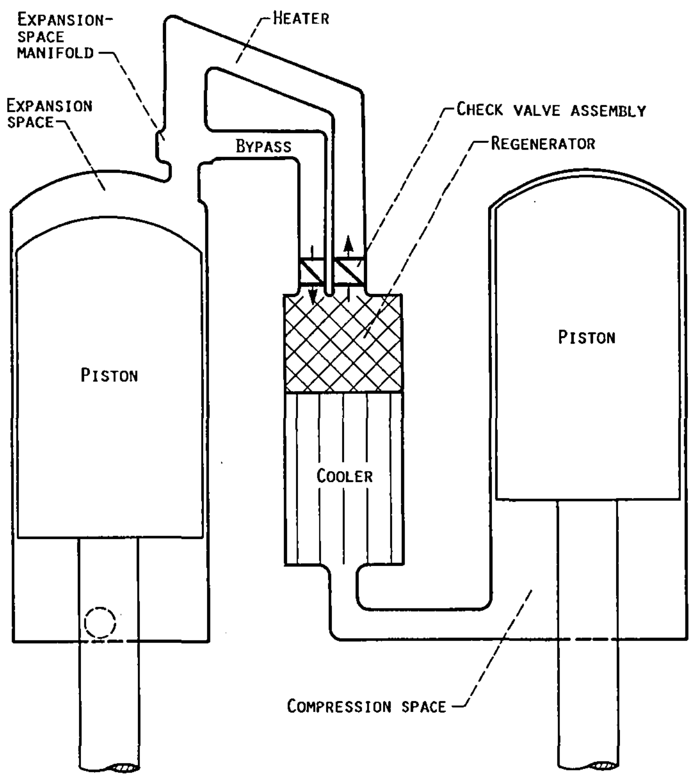

FIGURE 1. - DIAGRAM OF WORKING SPACE FOR ONE CYCLE OF HEATER-BYPASS CONFIGURATION. THE ENGINE CONSISTS OF FOUR CYCLES IN A DOUBLE-ACTING ARRANGEMENT.

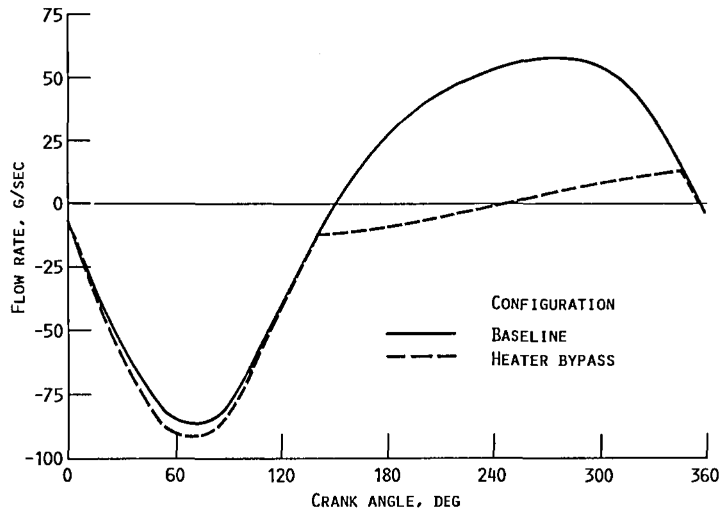

FIGURE 2. - FLOW RATE AT INTERFACE OF HEATER TUBES AND EXPANSION-SPACE MANIFOLD AS FUNCTION OF CRANK ANGLE. ENGINE SPEED, 4000 RPM; MEAN WORKING-SPACE PRESSURE, 15 MPa.

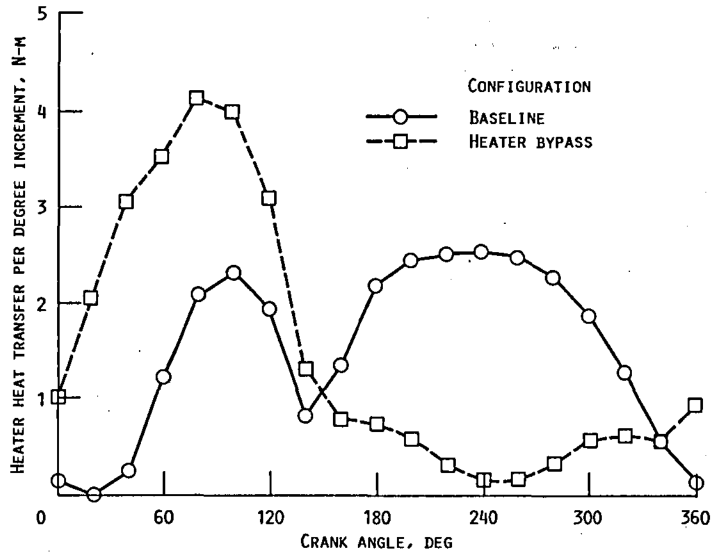

Figure 3. - Heat transfer in heater per degree increment as function of crank angle. Engine speed, 4000 rpm; mean working-space pressure, 15 MPa.

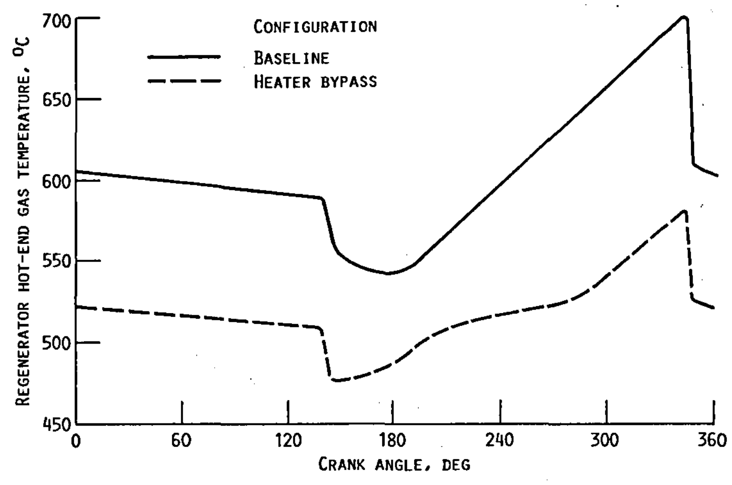

Figure 4. - Regenerator hot-end gas temperature as function of crank angle. Engine speed, 4000 rpm; mean working-space pressure, 15 MPa.

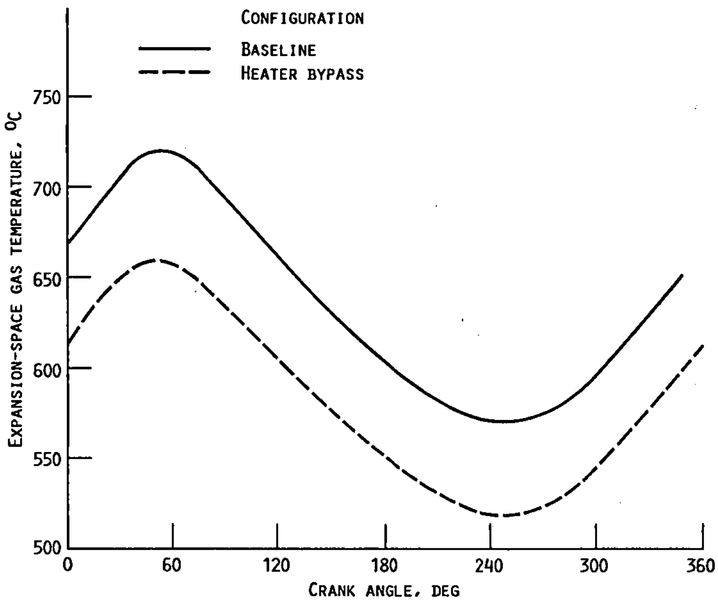

FIGURE 5. - EXPANSION-SPACE GAS TEMPERATURE AS FUNCTION OF CRANK ANGLE. ENGINE SPEED, 4000 RPM; MEAN WORKING-SPACE PRESSURE, 15 MPA.

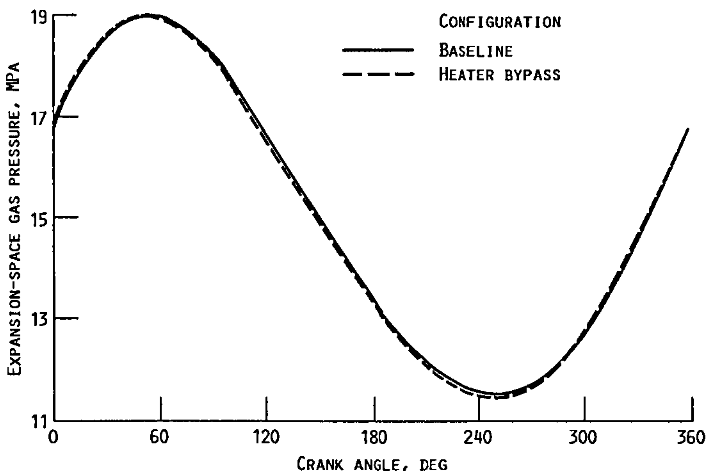

FIGURE 6. - EXPANSION-SPACE GAS PRESSURE AS FUNCTION OF CRANK ANGLE. ENGINE SPEED, 4000 RPM; MEAN WORKING-SPACE PRESSURE, 15 MPA.

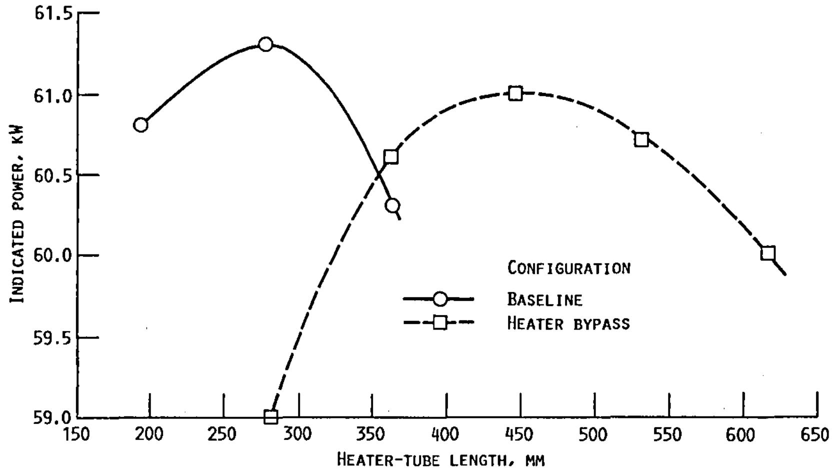

Figure 7. - Indicated power as function of heater-tube length. Engine speed, 4000 rpm; mean working-space pressure, 15 MPa; reference heater-tube length, 280.7 mm.

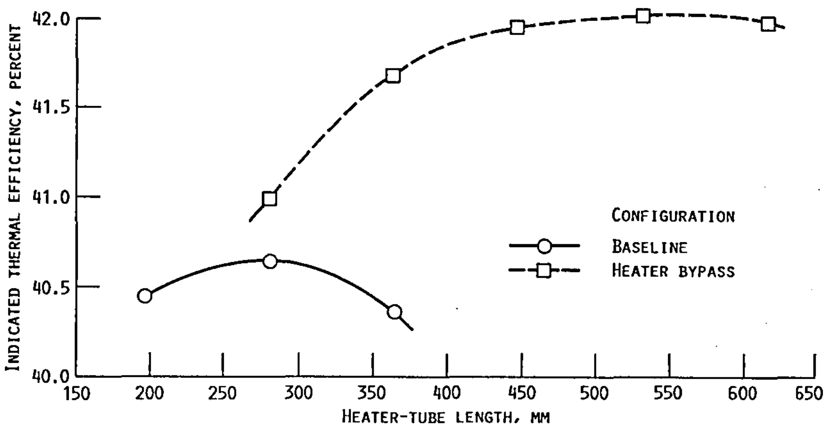

Figure 8. - Indicated thermal efficiency as function of heater-tube length. Engine speed, 4000 rpm; mean working-space pressure, 15 MPa; reference heater-tube length, 280.7 mm.

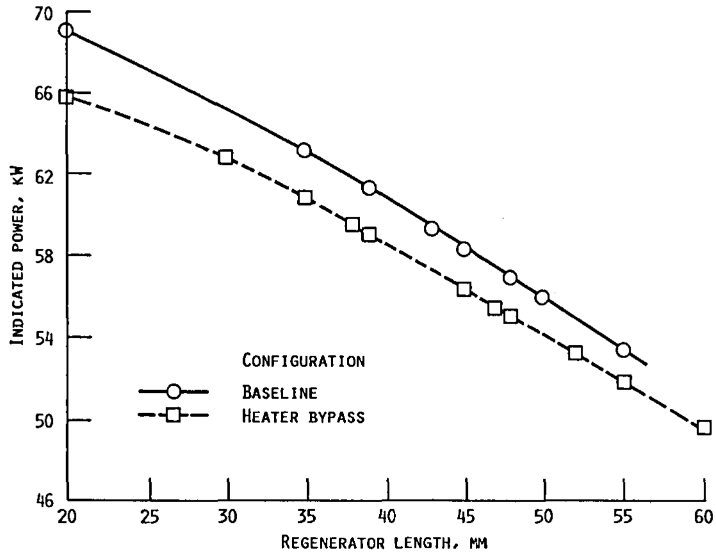

Figure 9. - Indicated power as function of regenerator length. Engine speed, 4000 rpm; mean working-space pressure, 15 MPa; reference generator length, 39 mm.

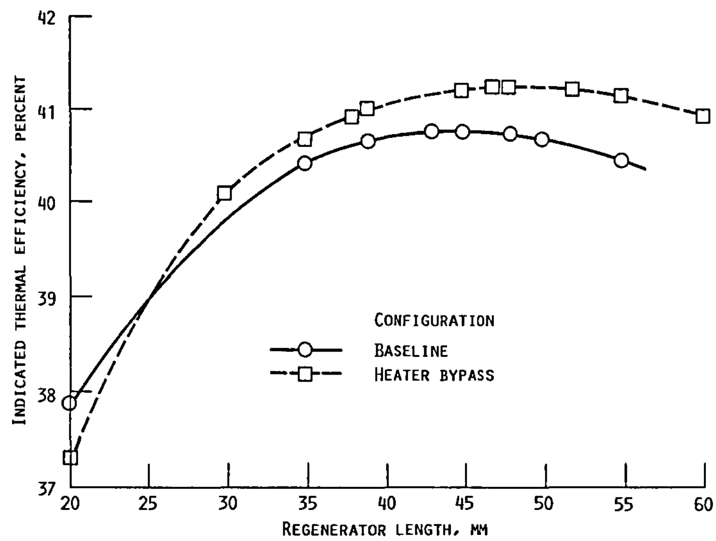

Figure 10. - Indicated thermal efficiency as function of regenerator length. Engine speed, 4000 rpm; mean working-space pressure, 15 MPa; reference regenerator length, 39 mm.

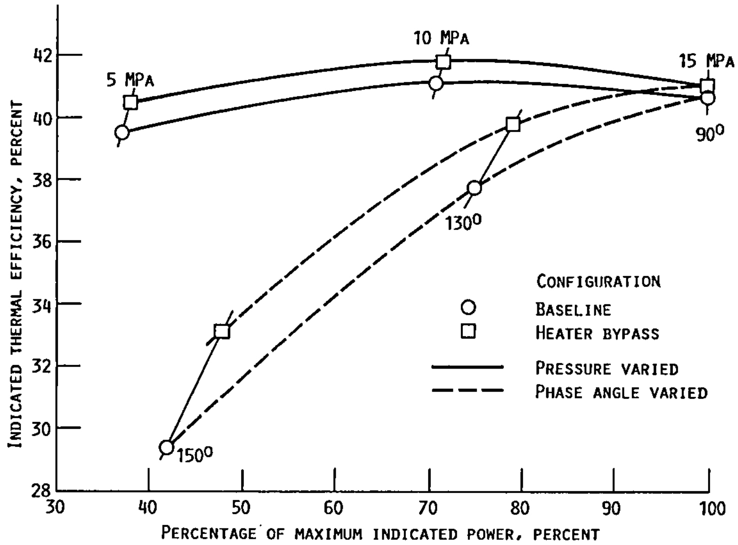

Figure 11. - Indicated thermal efficiency as function of percentage of maximum indicated power controlled by pressure and phase angle. Engine speed, 4000 rpm; P-40 maximum indicated power, 61.3 kW; heater-bypass maximum indicated power, 59.0 kW.

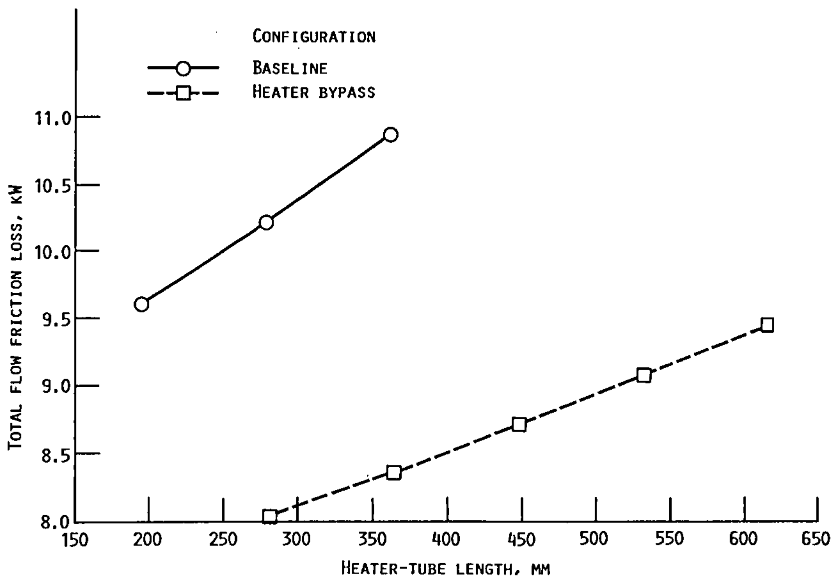

Figure 12. - Total flow friction loss as function of heater-tube length. Engine speed, 4000 rpm; mean working-space pressure, 15 MPa; reference heater-tube length, 280.7 mm.

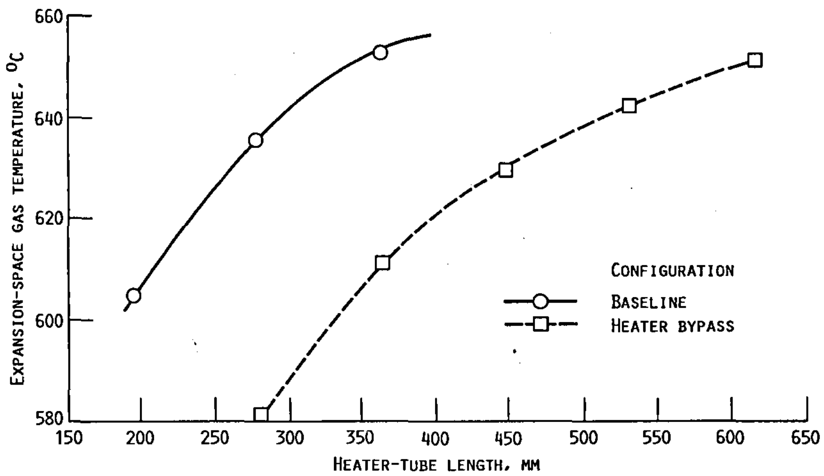

Figure 13. - Expansion-space gas temperture as function of heater-tube length. Engine speed, 4000 rpm; mean working-space pressure, 15 MPa; reference heater-tube length, 280.7 mm.

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15. Supplementary Notes

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16. Abstract

In support of the U.S. Department of Energy's Stirling Engine Highway Vehicle Systems program, the NASA Lewis Research Center investigated whether bypassing the P-40 Stirling engine heater during regenerative cooling would improve engine performance. The Lewis nodal-analysis Stirling engine computer simulation was used for this investigation. Results for the heater-bypass concept showed no significant improvement in the indicated thermal efficiency for the P-40 Stirling engine operating at full-power and part-power conditions. Optimizing the heater-tube length produced a small increase in the indicated thermal efficiency with the heater-bypass concept.

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