ROBUST SERVO/HYDRAULIC CONTROL SYSTEM AND METHOD

A servo control for a cam phaser (10) includes a pulse width modulated controller (20) that operates a four way solenoid valve (18). The cam phaser piston (12) or vane(s) is positioned with a computer program stored in the controller (20). The program establishes a deadband to exclude noise. It compensates for variations in oil temperature and voltage.
ROBUST SERVO/HYDRAULIC CONTROL SYSTEM AND METHOD

CROSS-REFERENCE TO RELATED APPLICATION

This application claims the benefit of U.S. Provisional Patent Application Serial No. 60/159,393, filed October 14, 1999 (Attorney Docket No. DP-301360).

TECHNICAL FIELD

The present invention relates to hydraulic control systems for use in automotive valve control devices.

BACKGROUND OF THE INVENTION

Automotive hydraulic control systems have been proposed in which the pressure of a control fluid, such as engine oil, is controlled for positioning of a hydraulic actuator. Control fluid viscosity can vary significantly with fluid temperature and age. Control fluid pressure can vary significantly during even one control cycle. Variations in fluid viscosity and pressure significantly affect dynamic hydraulic control performance. Accordingly, some attempt has been made to estimate control fluid viscosity and pressure and vary control gains in response thereto. For example, temperature and pressure have been measured or estimated and the temperature and age estimations used to estimate fluid viscosity, and the estimated viscosity and pressure used to vary control gains. Such complex sensing, estimating and processing yields some improvement in dynamic hydraulic control system performance.

Hydraulic controls are typically difficult and time consuming to calibrate. This is especially true in application that involve a wide range of operating conditions, as in a cam phaser control. Conditions for the system can and do change with temperature changes. The temperature alters the oil viscosity, the resistance of the control valve and clearances for the engine. Furthermore, as temperature increases, the copper wire in the solenoid actuator heats up,
thereby increasing electrical resistance.

A typical analog conventional control system is shown in FIG. 2. There a control command \( P_d \) generated by controller, for example as a predetermined function of such engine parameters as engine speed, load or intake pressure, and in accord with a desired phasing between the camshaft and crankshaft of a system to which the control function is applied, is provided to summing node 60. Sensed piston position signal \( P_a \) from sensor 36 is likewise applied in negated form to summing node 60, so to be subtracted from signal \( P_d \) to form a position error signal \( P_e \), to be minimized in accord with the control function of FIG. 2. Signals \( P_d \) and \( P_a \) are also applied to slope generator 64 which, generates \( m \), a rate of change of \( P_a \) over a predetermined period of time. In alternative embodiments, the slope generator may measure the responsiveness of the actuator 62 to a change in commanded actuator position. Actuator 62 of FIG. 2 simply represents the hydraulic actuator controlled such as the a cam phaser piston in a cam phaser control system.

The analog system compares an amount of change in the value CMD and a resulting amount of change in sensed actuator position \( P_a \) over a predetermined transient response period of time may be used to generate a transient response transfer function providing the responsiveness measure. As another example, the rate of reduction of any significant position error \( P_e \) in the system may indicate system responsiveness in accord with this invention.

The slope generator 64 provides output signal \( m \) representing the time rate of change or other responsiveness measure of actuator 62, for use in adjusting control responsiveness in accord with this invention. For example, when the proportional plus derivative plus integral PID control is used to drive actual actuator position \( P_a \) toward desired or commanded actuator position \( P_d \), the value \( m \) is provided to control blocks 66, 68, and 70 at which control gains \( K_p \), \( K_d \), and \( K_i \) are adjusted as predetermined functions of the value \( m \). Such functions may be simple linear relationships between magnitude of \( m \) and control gain magnitude, wherein gain magnitude increases with increasing \( m \) and decreases with decreasing \( m \). Further detail on such linear relationships would be provided through a conventional calibration process for a given system to provide appropriate and desirable control responsiveness adjustment.

A conventional system such as described above is calibrated as a compromise among the different expected operating conditions. For example, controlling duty cycle controls the percentage of oil flow, not the actual cam phaser mechanism. Moreover, control is not a linear function. In addition, oil flow rate changes with operating conditions. Those skilled in the art
can readily convert such analog controls into digital controls by providing appropriate sensors, processors and programs for replicating the analog control in digital form. However, even the digital form has the same drawbacks of compromised calibration and instability due to unexpected operating conditions.

As a result of these limitations of conventional controls, there is need for a controls system and method that accounts for variations in actuator voltage and oil temperature.

**SUMMARY OF THE INVENTION**

The present invention overcomes the shortcomings of the prior art through a hydraulic control system providing compensation for variations in fluid temperature and voltage. The invention provides a closed loop control for a hydraulic device using a pulse width modulated four way control valve. Multiple closed loop terms are adaptively tuned and selectively activated to achieve both stable operation and improved performance when compared to conventional closed loop controls.

More specifically, the present invention continuously calibrates the control system to account for changes in oil temperature and variations in applied voltage. Using a position, integral and deceleration control scheme, the integral and deceleration terms are selectively applied to improve the response of a cam phasing system. The control valve is tested by reading the voltage and increasing duty cycle to find the current at which the control valve first begins to move. The output is slowly increased until a response is observed. More specifically, with the control valve in a hold position the duty cycle of the PWM is gradually increased until movement in the hydraulic device is observed. This provides a threshold that corresponds to the initial temperature.

In the hold state, the integral term dominates control. “Term” is the product of gain and error. During movement the integral term is ignored and the proportional term controls. The deceleration gain of velocity squared transitions from the proposal to the integral as the spool comes to a stop. Subroutines in the program compensate the system for changes in temperature and voltage.

**BRIEF DESCRIPTION OF THE DRAWINGS**

The invention may be best understood in reference to the preferred embodiment and to
the drawings in which:

FIG. 1 is a general illustration of the hydraulic control system hardware of the preferred embodiment in an automotive application;

FIG. 2 is a block diagram of an analog control system.

FIG. 3 is a graph showing variations in fluid flow vs. duty cycle for the control valve for at different voltages and at the same temperature;

FIG. 4 is a graph showing variations in fluid flow vs. duty cycle for the same voltage and different oil temperatures;

FIGS. 5A and 5B are a computer flow diagram illustrating a step by step procedure for carrying out the control function described in FIG. 2 in accord with the preferred embodiment; and

FIG. 6 is a set of curves showing the duty cycles for hold, maximum phase direction and default direction.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to FIG. 1, a hydraulic control system is provided to control the position of a hydraulic actuator 12 such as a cam phaser piston, to provide for linear positioning thereof along a range of motion. The piston 12 may move in this embodiment bi-directionally, wherein hydraulic fluid pressure is applied to a first side of the piston 12 from hydraulic fluid admitted through passage 14 to a first side of the piston, and may move in a reverse direction of motion from pressure applied by hydraulic fluid passing through a second passage 16. The piston may move, as influenced by hydraulic pressure applied thereto, along a sleeve attached to a phasing device 10, wherein the phasing device may be of conventional design for varying the angular relationship between a crankshaft and camshaft as is generally understood in the art.

For example, the piston 12 may be attached, such as via a conventional paired block configuration or a conventional helical spline configuration, to a toothed wheel (not shown), on which is disposed a chain linked to an engine crankshaft. The phaser 10 may then be fixedly mechanically linked to a camshaft. A four way solenoid operated control valve 18 admits a varying quantity of hydraulic fluid through respective first and second passages 14 and 16 to respective first and second sides of piston 12 to apply pressure to such sides. The relative pressure applied to the first and second sides of piston 12 determines the steady state (hold)
position of the piston. Precise piston positioning along a continuum of positions within the sleeve of phaser 10 is provided through precise control of the relative position of control valve 18. The valve 18 is a conventional solenoid control valve with a supply 32 and vent ports 30, 31 are connected, respectively, to a source of pressurized oil and to a reservoir. Outlet ports 33, 34 are selectively coupled to the opposite sides of the cam phaser cylinder to drive its piston 12 in one or the opposite direction. The spool valve element 19 is biased in one direction by a bias spring. The solenoid actuator 22 operates on the spool valve element 19 to axially displace it to a desired position and hold it in that position until a change in position is called for by the control system.

The valve has vent ports 30, 31, a supply port 32, and cam phaser ports 33, 34. By adjusting the axial position of the spool 19, its lands selectively connect the cam phaser ports 33, 34 to one of the vent ports 30, 31 or the supply port 32. In this manner fluid is respectively supplied and drained from opposite ends of the cam phaser piston 12. As shown in FIG. 1, the spool is in its hold position and the ports 33, 34 are isolated from the vents 30, 31 and the supply 32. The pulse width modulator (PWM) controller 20 applies a PWM voltage signal to the solenoid 22 to hold or move the solenoid and the spool 19 connected thereto. As the spool 19 moves from its hold position through passage of current through the solenoid 22, a portion of the vented fluid is directed to the corresponding side of piston 12 to apply a hydraulic force thereto, to displace the piston away from its rest position in accord with the relative fluid pressure force applied across the piston. A somewhat linear relationship exists between valve current and hydraulic pressure applied to the piston 12. The force applied to the piston may generally be expressed as hydraulic pressure multiplied by piston area. In the embodiment of this invention in which piston 12 is linearly actuated in accord with the relative pressure there across. The piston is positioned along a substantial continuum of positions, so as to vary the angular relationship between crankshaft and camshaft in accord with generally understood automotive phasing techniques, variable valve timing is provided by varying the linear displacement of piston 12 within phaser 10. Examples of such phasing hardware may be generally found in U.S. Pat. Nos. 5,119,691, 5,033,327, and 5,163,872. The invention may also control a vane type cam phaser as shown and described in U. S. Patent Application Serial No. 60/147,329, filed August 5, 1999. There the piston is replaced by a rotor with multiple vanes and the chambers on opposite sides of the vanes hold the fluid that acts on the vanes.

PWM controller 20 receives position signals from the cam position sensor 24 and the
crankshaft sensor 26. By comparing the time differences in the received signals, the controller 20 determines the relative phase between the cam and the crankshaft. The spool 19 is held, for a given duty cycle, substantially at a fixed position corresponding to the average current in the coil, as is generally understood in the solenoid control art. The frequency of the PWM signal is set high enough that position of piston 12 is stable for a fixed PWM value. Hereinafter is described a calibration method to provide critical dampening of the motion of the phaser and continuous calibration of the system for varying operating temperatures.

FIG. 3 shows the effects of voltage on a typical system. The three plots show results of the same solenoid valve operated at the same temperature but at different voltages. Level 1 is the highest voltage, Level 3 is the lowest and Level 2 is intermediate Levels 1 and 2. The lowest point on each curve represents the duty cycle where the spool 19 holds the cam phaser piston 12 and the oil flow is zero or only leakage. The maximum oil flow occurs at the tops of the curves. FIG. 3 shows that, for a known temperature, one can predict the duty cycle that corresponds to the hold position for the spool and to the positions of maximum oil flow. For example, consider the Level 1 voltage. At a 35% duty cycle it holds its position. At about 14% duty cycle the valve carries maximum flow in one direction (in the default/spring biased direction) and at about 50% duty cycle it carries maximum flow in the opposite direction.

In FIG. 4 the voltage is held constant and the results of oil flow at two different temperatures are shown. Temperature 2 is higher than temperature 1 and the curves show that increased temperatures require higher duty cycles to achieve the same results as operation at lower temperatures. FIG. 4 also shows that the duty cycle can be used to calculate temperature. The control system program described hereinafter uses this feature to calculate temperature. After temperature is calculated, the various system gains (position and integral) are computed to provide dynamic adjustment for the system. At the end of the program the supply voltage of the solenoid is read. Then the calculated gains are compensated to adjust for changes in voltage.

FIG. 6 provides the duty cycle bands to determine the lower limits for 100% oil flow. The solid line curve shows the duty cycle for the hold position. The bands for the maximum phase direction (against spring bias) and default direction (with spring bias) are shown.

The operating characteristics of the valve as shown in the graphs is stored as data in the memory of the controller. The data is provided from initial design criteria and is updated in the memory by measured values.
The control operations accomplished by controller 20 are carried out as a series of programmed steps by a processor (not shown) in controller 20. That controller includes a processor with conventional components and accessories including a read only memory for storing operating system and application programs, a random access memory for holding data and an arithmetic logic unit for performing computation and logical operations. One of the application programs is a series of operations as further shown in the flow chart of Figs. 5A and 5B.

The series of steps performed by the controller of the control system includes a first step 101, where the engine for the automobile is turned on. The control system learns the offsets between the cam tooth wheel and a reference signal such as a non-hyphened phase cam or crank reference. These offsets are learned at or above a given number of revolutions per minute so that the system is stabilized. In step 102, the software determines the amount of cam or crank drive noise in the system. In any given system, there are inherent fluctuations. These fluctuations are usually minor and are not the result of changes in the operating conditions. As such, the system learns these variations and uses these variations to establish a noise level. Once the noise level is established, the software will ignore making changes if any changes are sensed within the noise level. In effect, this establishes a deadband for system changes. In order to be recognized and adjusted, any changes must be greater than this deadband. This step improves the ability of the control system to hold the position and avoids the unwanted effect of a system trying to correct itself for what are, in effect, normal variations.

In step 103, the algorithm learns the voltage compensated integral term for a controller. The integral term is initialized to a low value. The output duty cycle to the control valve is the integral term modified by the voltage correction. The algorithm increases the integral term until the sensor 24 of the control system detects movement as seen in the cam phaser 10. The integral term just prior to the initiation of movement becomes a starting term that is used in the control loop. By separating out the integral term for other voltage effects, the system reacts faster to voltage changes. In conventional systems the integral term is used to compensate for voltage changes. However, since the integral is a reaction to the magnitude of error over time, it cannot respond instantaneously to correct for a change in voltage. In effect, the integral term slows down the reaction time of the system. This is desirable when the spool 9 is in a hold position. The invention overcomes this problem by establishing certain windows in the operation for using
the integral term. Step 104 helps establish those windows. The algorithm compares the previous setpoint to a desired setpoint. If the new setpoint is not greater than the old setpoint by previously defined and stored hysteresis or background noise, then the algorithm uses the old setpoint. This increases the stabilization, especially if the desired position is generated externally. Step 105 prevents the system from exceeding either user-designed or physical limits. The setpoint is compared to these limits and if the new setpoint exceeds the limits, it is appropriately adjusted to a lower value.

Having established the baseline for the operating system, the next step is to determine the oil temperature using the voltage compensated integral term. This is done in step 106 with the data shown in Fig 6. The algorithm is designed so that the integral term is primarily used to keep the valve in a steady state hold position. The hold position is represented in the valve profiles as the area on the oil flow curves that correspond to zero oil flow. In other words, it corresponds to the minimum point of the curve shown in Figs. 3 and 4. As those figures show, the duty cycle changes with voltage and temperature. Since the integral term is already compensated for voltage, any change in the integral term, i.e., the hold position, represents a change in temperature.

Once the temperature is established and the changes in temperature are known, the next step provides the proportional gain based upon temperature change. This is step 107. Following that calculation, step 108 calculates the duty cycle for which maximum flow from the control valve for the given hold position. This takes into account changes in temperature and voltage. Using the oil flow profiles shown in Fig. 4, the algorithm predicts the required duty cycle at which the valve delivers its maximum oil flow. The algorithm uses the integral term to determine the steady state hold position, i.e., the position where zero oil flow occurs, which in turn is adjusted by a voltage correction term. That voltage adjusted integral term is then used to calculate the duty cycles that represent the maximum oil flow.

Having compensated for temperature, the system then calculates the velocity of the cam phasing movement in step 109. As shown in step 110, if the average velocity exceeds a referenced velocity while at maximum oil flow (found in step 108), then the program provides a velocity compensation factor. That factor is the average velocity divided by the reference velocity. In step 111, the servo gains and bands are adjusted by the velocity-compensation factor. As velocity increases, proportional and integral gains decrease while derivative gain and the
bands increase. In step 112, the error from the setpoint and feedback readings are calculated and the direction of the error is determined (step 113.) In step 114, positive servo factors are used to correct the direction and in step 115, further adjustments are made to the proportional gain to compensate for velocity.

In step 116, the program checks to see if the system's velocity and error are less than user-defined limits. If so, it means that the system is not moving fast, the error is small and it is likely holding a position. The system only updates the integral term when holding a position. This prevents the system from wind-up and unnecessary hunting.

In step 117, the integral term is calculated and in step 118, the algorithm determines if the error is less than the defined limit to calculate the deceleration term. The deceleration term is not used until the system approaches the setpoint. It is primarily used as a brake. The deceleration term is proportional to the kinetic energy of the system. The kinetic energy is proportional to the square of velocity ($V^2$). In effect, the deceleration term is only used during a certain window as the system approaches its desired setpoint. In step 119, the system ensures that no single term will dominate the system.

In step 120, the individual gain terms are then combined, including the proportional, integral and deceleration terms. This total term is independent of voltage. The impact of voltage variations are included in step 121. As a result, the control system separates out the voltage influences from the proportional, integral and deceleration terms. This improves the robustness of the system. It allows the system's voltage to change at any time without causing the proportional, integral and deceleration terms to chase and correct for voltage changes. Because the system does not correct for voltage changes until this point in the program, it is a more stable system. As a result, a steep change will produce the same proportional, integral and deceleration term values independent of voltage. Accordingly, regardless of the voltage, the duty cycle of the system will be adjusted as needed at the valve in order to maintain the same oil flow. In addition, voltage fluctuations can occur quickly and often, especially in an automotive environment. If the voltage is not monitored and compensated, it could cause havoc with the ability to control the hydraulic system.

In step 122, the system output is applied to the control valve and in step 123, the program is repeated beginning from step 105.

Having described the preferred embodiment, those skilled in the art understand that modifications, changes, additions and deletions may be made to the steps of the program without departing from the spirit and scope of the claims.
Claims:

1. A hydraulic servo control system for controlling the delivery of hydraulic fluid from a reservoir of fluid a load to drive a load piston to a desired position wherein the fluid flow and applied voltage vary with operating temperature, comprising:

   a control valve having supply and vent ports in fluid communication with the reservoir, forward and reverse ports in fluid communication with opposite end of the load piston and a valve element moveable from a hold position sealing all the ports from each other to a forward position for connecting the supply port to the forward port and the reverse port to the vent port, respectively or to a reverse position for connecting the supply port to the reverse port and the forward port to the vent port, respectively;

   an electromagnetic actuator coupled to the moveable valve element for displacing the valve element to one of its hold, forward or reverse positions;

   a pulse width modulated supply having a variable voltage and a controlled duty cycle and generating a pulse width modulated voltage duty cycle for driving the electromagnetic actuator to the one of its positions;

   one or more sensors couple to the load for generating signals representative of the position of the load;

   a feedback control system coupled to the pulse width modulated supply and to the sensors for adjusting the duty cycle of the pulse width modulated supply to compensate for variations in temperature of the hydraulic fluid.

2. The hydraulic servo control of claim 1 wherein the feedback control further comprises means for measuring the voltage applied to the electromagnetic actuator;

   means for increasing the current applied to the electromagnetic actuator until the load piston moves;

   means for storing a holding voltage and duty cycle values less than but proximate to the voltage and duty cycle that generated movement in the load piston to determine proportional gain for current operating temperature;

   means for calculating a predicted duty cycle corresponding to a maximum fluid flow rate through said control valve.
3. The hydraulic servo control of claim 2 wherein the feedback control means adjusts proportional gain based on the duty cycle for the stored holding values of voltage and duty cycle; and periodically updates the holding voltage and duty cycle values for changes in temperature.

4. The hydraulic servo control of claim 1 wherein the feedback control system monitors velocity of the load piston at maximum fluid flow rate and adjusting proportional, integral and deceleration gains in accordance with the sensed velocity.

5. The hydraulic servo control of claim 3 wherein the feedback control further compensates the combined gain of the system in accordance with changes in the voltage level.

6. A method for operating a hydraulic servo control system for controlling the delivery of hydraulic fluid from a reservoir of fluid a load to drive a load piston to a desired position with a pulse width modulated signal applied to a control valve wherein the fluid flow and applied voltage to the control valve vary with operating temperature, said method comprising:
   generating signals representative of the position of the load;
   providing multiple servo loop gains including gains for proportional, integral and deceleration changes of the position of the load;
   adjusting the duty cycle of the pulse width modulated supply to compensate for variations in temperature of the hydraulic fluid;
   after adjusting for the temperature, adjusting the duty cycle of the pulse width modulated supply to compensate for changes in voltage level.

7. The method of claim 6 further comprising the steps of measuring the voltage applied to an electromagnetic actuator on the control valve; increasing the current applied to the electromagnetic actuator until the load piston moves; storing a holding voltage and duty cycle values less than but proximate to the voltage and duty cycle that generated movement in the load piston to determine proportional gain for current operating temperature;
   calculating a predicted duty cycle corresponding to a maximum fluid flow rate through said control valve.
8. The method of claim 7 further comprising the steps of:
   adjusting proportional gain based on the duty cycle for the stored holding values of
   voltage and duty cycle; and
   periodically updating the holding voltage and duty cycle values for changes in
   temperature.

9. The method of claim 7 further comprising the steps of
   monitoring velocity of the load piston at maximum fluid flow rate and adjusting
   proportional, integral and deceleration gains in accordance with the sensed velocity.

10. The method of claim 9 further comprising the step of decelerating the motion of the load
    in accordance with the kinetic energy of the load as the position of the load approaches a set
    point.
Figure 1.
Influence on Oil Flow Curve Due to Voltage

Flow vs Duty Cycle at One Temperature

Max Flow threshold in 'default' direction at Voltage Level 1

Max Flow threshold in 'max phase' direction at Voltage Level 2

Max Flow threshold in 'default' direction at Voltage Level 3

Max Flow threshold in 'max phase' direction at Voltage Level 2

Hold Position at Voltage Level 1

Hold Position at Voltage Level 2

Hold Position at Voltage Level 3

Flow (LPM)

% Duty Cycle

0% 5% 10% 15% 20% 25% 30% 35% 40% 45% 50% 55% 60% 65% 70% 75% 80% 85% 90% 95% 100%

Note: 'default' and 'max phase' direction is dependent on application, whether it is used on an exhaust or intake cam.

Figure 3
**Figure 5A**

1. **Start**
   - Stabilize the system (RPM > Specified Speed)
   - Learn Zero Offsets

2. **Read the feedback & retain the highest/lowest readings.**
   - Has *Noise Time interval* elapsed?
     - Yes: Back ground noise = highest reading - lowest reading
     - No: Learn the Integral Term (Increase the integral up until movement is seen - compensated for voltage)

3. **Ignore Set point change noises.**
   - Is the Set point change > user defined *Hysteresis or Background Noise?*
     - Yes: Use new Set point
     - No: Use old Set point

4. **Cap the Set point to reasonable Limits.**
   - Did Set point exceed *Limits?*
     - Yes: Set point = *High/Low Limit*
     - No: Determine Oil Temperature using voltage compensated Integral term

5. **Calculate and regulate.**
   - Determine Prop gain using calculated oil temp
   - Determine maximum flow DC's for the current Integral term and voltage
   - Calculate current system velocity. Calculate Avg. velocity from at rest & while at full output.
   - Velocity Compensation = (Avg. velocity / Reference velocity) * Compensation gain
   - Adjust servo gains & bands by a factor of Velocity Compensation
   - Error = Set point - Feedback
Figure 5B
Duty Cycle Bands to Determine the 'Lower Limits' for 100% Oil Flow

Algorithm Limits to 100% DC

Figure 6
**INTERNATIONAL SEARCH REPORT**

**A. CLASSIFICATION OF SUBJECT MATTER**
- IPC(7) : F15B 9/09; FOIL 1/34
- US CL : 91/363R; 123/90.15

According to International Patent Classification (IPC) or to both national classification and IPC.

**B. FIELDS SEARCHED**

Minimum documentation searched (classification system followed by classification symbols)
- U.S. : 60/329; 91/361

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)
- EAST: pulse width modulation, cam phaser, crankshaft.

**C. DOCUMENTS CONSIDERED TO BE RELEVANT**

<table>
<thead>
<tr>
<th>Category</th>
<th>Citation of document, with indication, where appropriate, of the relevant passages</th>
<th>Relevant to claim No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>X</td>
<td>US 5,937,806 A (LYKO et al.) 17 August 1999 (17.08.1999), column 1, line 53 - column 2, line 29.</td>
<td>1-10</td>
</tr>
</tbody>
</table>

[ ] Further documents are listed in the continuation of Box C.  
[ ] See patent family annex.

Date of the actual completion of the international search  
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Form PCT/ISA/210 (second sheet) (July 1998)